Model-Based Design and Virtual Testing of Steer-by-Wire Systems

Marcus Irmer
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Abstract

Driven by the need for automation and autonomy as well as the need to reduce resources and emissions, the automotive industry is currently undergoing a major transformation. Technologically, this transformation is addressing a wide range of challenges and opportunities. The optimal control of all components is significant for the sustainable development and eco-friendly operation of vehicles. Additionally, robust control of the actuators forms the basis for the development of driver assistance systems and functions for autonomous driving. The actuators of the steering system are particularly important, as they enable safe and comfortable lateral vehicle control. Therefore, the model-based development and virtual simulation of an innovative highly robust control approach for modern Steer-by-Wire systems were conducted in this thesis. The approaches and algorithms described in this thesis allow the design of robust Steer-by-Wire systems and offer the opportunity to conduct many investigations in a computer-aided virtual environment at an early stage in the development process. This reduces time- and cost-intensive testing on prototypes, avoids unnecessary iterations in the design and significantly increases the efficiency and quality of the development. The desired high degree of robustness of the steering control also ensures that the parameterization of the steering feel generator can be freely selected for the individual application. This enables safe and comfortable vehicle lateral control. In summary, the research results described in this thesis accelerate the development of new, modern Steer-by-Wire systems whose robust design forms the basis for the realization of functions for highly automated and autonomous driving.

Keywords: mechatronic systems, vehicle dynamic systems, steer-by-wire systems, modeling, model reduction, optimal control theory, robust controller synthesis, robustness analysis

Marcus Irmer, Department of Electrical Engineering, Electricity, Box 65, Uppsala University, SE-751 03 Uppsala, Sweden.

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This thesis is based on the following papers, which are referred to in the text by their Roman numerals.


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Abbreviations

AU  Axle Unit
C   control
D   disturbance
drv  driver
DAS Driver-Assistance Systems
DoF Degrees of Freedom
EPS Electromechanical Power Steering
FU  Feedback Unit
HPS Hydraulic Power Steering
LQE Linear-Quadratic Estimator
LQG Linear-Quadratic-Gaussian
LQR Linear-Quadratic Regulator
M   measurement
MA  Axle Motor
MF  Feedback Motor
O   objective
P   plant
R   reference
R   Rack
req requested
S   Steering wheel
SbW Steer-by-Wire
TB  Torsion Bar
WL  Left Wheel
WR  Right Wheel
zoh zero-order-hold
Nomenclature

$A_{AU}$  
System matrix of the AU model

$A_D$  
System matrix of the discrete augmented plant model for the LQR resp. LQE design

$A_{FU}$  
System matrix of the FU model

$A_{SbW}$  
System matrix of the detailed SbW model

$A_{\xi SbW}$  
System matrix of the reduced SbW model

$A_{\xi SbW,D}$  
System matrix of the discrete reduced SbW model

$B_{AU}$  
Input matrix of the AU model

$B_D$  
Input matrix of the discrete augmented plant model for the LQR resp. LQE design

$B_{FU}$  
Input matrix of the FU model

$B_{SbW}$  
Input matrix of the detailed SbW model

$B_{\xi SbW}$  
Input matrix of the reduced SbW model

$B_{\xi SbW,D}$  
Input matrix of the discrete reduced SbW model

$b_{MF}$  
Viscous friction of the feedback motor

$b_S$  
Viscous friction of the steering wheel

$b_{TB}$  
Damping constant of the torsion bar

$c_{AU}$  
Output matrix of the AU model

$c_D$  
Output matrix of the discrete augmented plant model for the LQR resp. LQE design

$c_{FU}$  
Output matrix of the FU model

$c_{SbW}$  
Output matrix of the detailed SbW model

$c_{\xi SbW}$  
Output matrix of the reduced SbW model

$c_{\xi SbW,D}$  
Output matrix of the discrete reduced SbW model

$c_{MR}$  
Equivalent stiffness between AU motor and rack

$c_{res}$  
Resulting stiffness

$c_{RWL}$  
Stiffness of the left wheel attachment

$c_{RWR}$  
Stiffness of the right wheel attachment

$c_{TB}$  
Stiffness of the torsion bar

$D_D$  
Feedthrough matrix of the discrete augmented plant model for the LQR resp. LQE design

$F$  
Noise filter matrix

$F_R$  
Rack force

$i_{RWL}$  
Gear ratio of the left wheel attachment

$i_{RWR}$  
Gear ratio of the right wheel attachment
$\dot{J}_{MA}$ equivalent moment of inertia of the AU motor and the nut transformed to rack coordinates

$J_{MF}$ moment of inertia of the feedback actuator

$J_{res}$ resulting moment of inertia

$J_S$ moment of inertia of the steering wheel

$J_{WL}$ moment of inertia of the left front wheel

$J_{WR}$ moment of inertia of the right front wheel

$k$ index for characterizing the point in time

$K$ optimal controller gain matrix

$\hat{K}$ optimal observer gain matrix

$n$ number of state variables of the augmented plant model for the LQR resp. LQE design

$n_p$ number of state variables of the plant model

$p$ number of input variables of the augmented plant model for the LQR resp. LQE design

$p_c$ number of control input variables of the plant model

$p_d$ number of disturbance input variables of the plant model

$P_e$ stationary covariance matrix of the a-priori estimation error

$q$ number of output variables of the augmented plant model for the LQR resp. LQE design

$q_m$ number of measurement output variables of the plant model

$q_o$ number of objective output variables of the plant model

$Q$ positive semidefinite weighting matrix for the LQR design to penalize the control errors

$R$ positive definite weighting matrix for the LQR design to penalize the use of the control variables

$S$ matrix representing the positive definite solution of the algebraic matrix Riccati equation for the LQR design

$s_R$ deflection of the rack

$s_{R_rq}$ requested deflection of the rack

$t$ time

$T$ sample time

$T_{TB}$ torsion bar torque

$T_{TB_rq}$ requested torsion bar torque

$T_{MA}$ AU motor torque

$T_{MArq}$ requested AU motor torque

$T_{MF}$ FU motor torque

$T_{MFrq}$ requested FU motor torque

$T_{S}$ steering torque

$T_{WL}$ torque about the steering axis of the left front wheel

$T_{WR}$ torque about the steering axis of the right front wheel

$u_{AU}(t)$ input vector of the AU model

$u_{pd,k}$ disturbance input vector of the discrete plant model
\( u_{FU}(t) \) input vector of the FU model
\( u_k \) input vector of the discrete augmented plant model for the LQR resp. LQE design
\( u_{pc,k} \) control input vector of the discrete plant model
\( u_{SbW}(t) \) input vector of the SbW model
\( u_{SbW,k} \) input vector of the discrete SbW model
\( V \) positive semidefinite intensity matrix of the process noise
\( W \) positive definite intensity matrix of the measurement noise
\( x_0 \) initial state vector of the discrete augmented plant model for the LQR resp. LQE design
\( \dot{x}_{AU}(t) \) state vector of the AU model
\( x_{AU,0} \) initial state vector of the AU model
\( \hat{x}_{d,k} \) a-posteriori estimate of the disturbance variables for the LQE design
\( \dot{x}_{FU}(t) \) state vector of the FU model
\( x_{FU,0} \) initial state vector of the FU model
\( \dot{x}_k \) state vector of the discrete augmented plant model for the LQR resp. LQE design
\( \hat{x}_0 \) initial a-priori estimate of the state vector of the discrete augmented plant model for the LQE design
\( \hat{x}_k \) a-priori estimate of the state vector of the discrete augmented plant model for the LQE design
\( \hat{x}_k \) a-posteriori estimate of the state vector of the discrete augmented plant model for the LQE design
\( \hat{x}_{p,k} \) a-posteriori estimate of the state vector of the reduced SbW model
\( \dot{x}_{r,k} \) reference vector
\( \dot{x}_{SbW}(t) \) state vector of the detailed SbW model
\( \dot{x}_{SbW,0} \) initial state vector of the detailed SbW model
\( \hat{x}_{SbW}(t) \) state vector of the reduced SbW model
\( \hat{x}_{SbW,0} \) initial state vector of the reduced SbW model
\( \hat{x}_{SbW,k} \) state vector of the discrete reduced SbW model
\( y_{AU}(t) \) output vector of the AU model
\( y_{FU}(t) \) output vector of the FU model
\( y_k \) output vector of the discrete augmented plant model for the LQR resp. LQE design
\( \hat{y}_k \) estimate of the measurement output vector of the discrete plant model
\( y_{pm,k} \) measurement output vector of the discrete plant model
\( y_{po,k} \) objective output vector of the discrete plant model
\( y_{SbW}(t) \) output vector of the SbW model
\( y_{SbW,k} \) output vector of the discrete SbW model
\( \phi_{drv} \) steering angle requested by the driver
\( \phi_{MA} \) rotational degree of freedom of the AU motor
$\varphi_{MF}$ rotational degree of freedom of the FU motor
$\varphi_S$ rotational degree of freedom of the steering wheel
$\varphi_{WL}$ angle of the left front wheel
$\varphi_{WR}$ angle of the right front wheel
$\tau_{MA}$ time constant of the AU motor control
$\tau_{MF}$ time constant of the FU motor control
$\omega_0$ eigenfrequency
$\Omega_{MF}$ angular velocity of the FU motor
$\Omega_S$ angular velocity of the steering wheel
$\Omega_{WL}$ angular velocity of the left front wheel
$\Omega_{WR}$ angular velocity of the right front wheel
1 Introduction

This thesis provides a comprehensive summary of the appended papers which are results of the author’s research in the domain of model-based design and virtual testing of Steer-by-Wire (SbW) systems and driver-assistance systems (DAS) for automated vehicles.

In this chapter, the motivation for the topic is explained. Based on this, the research questions are derived. Furthermore, an outline of the thesis is presented.

1.1 Motivation and Research Questions

Highly automated and autonomous driving will be the future. SbW systems represent a key technology for this. In this context, robust steering control is a fundamental precondition for automated vehicle lateral control. [1][2] However, the design models for current steering controls fail to account for various challenges such as degrees of freedom, signal delays and nonlinear characteristics of the steering system. Therefore, the objective of this thesis is to develop new methods for SbW systems and associated functions to increase their quality significantly. So, the following main research question is addressed within this thesis:

*How can highly dynamic control approaches for modern SbW systems be designed, which simultaneously guarantee an extremely high robustness?*

This main question can be divided into two more specific research questions:

1. What is the minimal and optimal level of detail for modeling the characteristics of a SbW system for this purpose?
2. How can control algorithms and components for SbW systems already be comprehensively virtual tested in simulation under realistic conditions at an early stage of the development?

Fundamental for answering these questions are optimal SbW models. Then, the structural design of the research according to Figure 1.1 results.
1.2 Outline of the Thesis

The outline of the thesis is derived from the structural design from Figure 1.1 and follows the mechatronic development circle from Figure 1.2. In this thesis, the steps 1 to 4 are described. Steps 5 and 6 are subject of future work.

First, a theoretical background on the topic is given in Chapter 2. Then, Chapter 3 describes the steps of the mechatronic development cycle to answer the research questions from Chapter 1.1. The mechatronic development cycle addresses the development of technical systems with mechanical, electrical and information-processing components. The result of this development yields a product that shows optimal behavior for the specified requirements and objectives. Central for the development are suitable models. Therefore, a novel detailed model of a SbW system is developed in step 1. In step 2, the dominant behavior of the SbW system is identified. Based on this, appropriate reduced
models of a SbW system are derived to answer the first research question. By using these innovative models, the best possible results of the individual steps are achieved. The steps of modeling and model analysis are followed by the model-based synthesis of a new control algorithm in step 3, which leads to an extraordinary robustness of the corresponding control system. The excellent characteristics of the control algorithm are then validated through virtual simulation in step 4 to answer the second research question. The result of this step is a validated control algorithm whose good characteristics are guaranteed by mathematical methods and which does not have to be subsequently tuned using further cost- and time-consuming tests on prototype vehicles. Furthermore, a summary and conclusion is given in Chapter 4. In Chapter 5, future work is discussed. Chapter 6 provides a brief summary of each paper included in the thesis. Finally, a Swedish summary of the thesis is presented in Chapter 7.
2 Theoretical Background

The aim of this chapter is to introduce the reader to general definitions and concepts of steering systems. First, the function of steering systems and the resulting task of vehicle lateral control are described. Then, a definition of the central term steering feel is given. Finally, a conventional electromechanical power steering (EPS) system, which currently represents the state of the art, is outlined before Chapter 3 discusses the development of modern SbW systems.

2.1 Function and Importance of a Steering System

A steering system enables a driver to steer a vehicle. Its function is to convert the angle at the steering wheel into an angle at the wheels. In addition, the steering system should provide the driver with relevant information by giving feedback of the current driving situation via the steering wheel. This feedback sensed by the driver is called steering feel. The steering feel is a major factor in the overall driving experience. [3] Therefore, a large number of requirements are specified for the steering feel and the steering system [4]:

- Adequate steering torque and small required steering wheel angle for parking (steering power assistance of up to 80 %)
- Good handling, reliability, sensitivity, accuracy and directness (e.g. precise steering without delay)
- Feedback of contact between tire and road (e.g. friction coefficient)
- Automatic return to center position, good center feel and stabilizing behavior during all driving situations (e.g. no overshoot)
- Disturbance rejection and sufficient damping (e.g. shock suppression)
- Fulfillment of crash requirements
- Low energy consumption, noise and vibration level as well as wear and maintenance

The individual requirements are evaluated in a highly subjective manner. In addition, the requirements are partly in conflict with each other. This makes the development of new steering systems more difficult.
Since the steering system is crucial for the safety and control of a vehicle, the development of robust steering systems forms the basis for highly automated and autonomous driving in the future.

2.2 Vehicle Lateral Control

Driving a vehicle can be divided into two tasks: controlling the vehicle’s longitudinal and lateral motion. The longitudinal motion is controlled by the driver through the throttle and brake pedal. Conversely, the lateral motion (translation in lateral direction and rotation around the vertical axis) is controlled by the driver using the steering wheel and the steering system. [5] A simplified block diagram of the resulting vehicle lateral control is presented in Figure 2.1. In this context, the driver acts as the controller while the vehicle serves as the plant.

![Figure 2.1. Simplified block diagram of the vehicle lateral control.](image)

The driver visually and kinesthetically perceives the vehicle lateral motion. If this observed motion is not equal to the desired motion, the driver applies a torque at the steering wheel to correct the vehicle lateral motion by deflecting the steering wheel and so the front wheels of the vehicle. This correction is in turn noticed by the driver, resulting in a closed-loop system. The steering system represents the actuator in this closed-loop system. To ensure safe vehicle lateral control, a robust steering system, which will be developed in Chapter 3, is therefore essential.

2.3 Definition of Steering Feel

The steering feel is the subjective experience of a driver of a vehicle based on the interaction between the driver and the vehicle while steering. The steering feel is the sum of the driver’s visual, kinesthetic and haptic perceptions during steering. [3][6][7] However, the torque $T_S$ that the driver induces resp. perceives at the steering wheel via the hands is the most significant component affecting the steering feel. This torque $T_S$ is called steering torque. Thus, a desired steering feel directly results in a corresponding steering torque $T_S$. The two terms steering feel and steering torque are therefore closely connected.
In modern steering systems, the characteristics of the steering torque $T_S$ can be designed almost arbitrarily. However, there are some requirements. The steering torque $T_S$ should provide the driver with feedback on useful information about the road surface, such as changes in friction coefficients, as well as the driving situation, including aspects like understeer or oversteer. Disturbance information, on the other hand, such as steering unsteadiness, bumps and vibrations, should ideally not be perceived by the driver. [8] Figure 2.2 shows an example of how the steering feel is composed by various driving situations.

![Figure 2.2. Structure of the steering feel.](image)

The tires are the only connection between the vehicle and the road. They transmit the forces and torques that lead to the vehicle movement. When driving, the point of application of these forces and torques is shifted by the dynamic tire offset. This shift induces torques $T_{WL}$ and $T_{WR}$ about the steering axis of the left and right front wheel due to the lateral forces at the tires. The torques $T_{WL}$ and $T_{WR}$ are a nonlinear function of the current driving situation, the condition of the road and the contact between the tire and the road. [9] For conventional rack-and-pinion steering systems, the torques $T_{WL}$ and $T_{WR}$ about the steering axis of the left and right front wheel can be combined and transformed into an equivalent force $F_R$ at the rack. This force $F_R$ is proportional to the steering torque $T_S$ experienced by the driver at the steering wheel. Thus, the driver can receive feedback on the road surface and the current driving situation via the equivalent force $F_R$ at the rack. [8] The force $F_R$ is used for the feedback because the torques $T_{WL}$ and $T_{WR}$ about the steering axis of the left and right front wheel cannot be estimated individually with the given measured variables. However, disturbances remain present in the resulting force $F_R$
at the rack. To allow a comfortable steering while suppressing the disturbances, only a part of the rack force $F_R$ should be transmitted to the driver. The rest has to be compensated by the steering control. Therefore, the task is to design a steering feel generator and a corresponding control of the driver’s steering torque which are able to transmit the useful information to a sufficient level and at the same time suppress disturbances.

An exemplary structure of a steering feel generator can be found in [10]. The approach to control the driver’s steering torque is based on [11].

### 2.4 Electromechanical Power Steering Systems

The most common steering technology in vehicles is rack-and-pinion steering. Its advantage lies in its simple and effective structure, which ensures precise and reliable steering. Since the 1990s, rack-and-pinion steering systems equipped with steering power assistance have been the standard, initially in the form of hydraulic power steering (HPS) systems. Since the 2010s, rack-and-pinion steering systems exist almost exclusively in the form of EPS systems, in which the hydraulic components are replaced by electric motors to provide steering assistance. The advantages of EPS systems, such as improved energy efficiency, the possibility of integration into driver assistance systems and lower maintenance compared to HPS systems, have contributed to their increasing popularity. Further advantages of EPS systems can be found, for example, in [6]. Figure 2.3 (left) shows the steering mechanism of a conventional EPS.

![Figure 2.3. Steering mechanism of an EPS (left) and SbW system (right).](image)

Today’s vehicles are predominantly equipped with EPS systems. However, SbW systems are expected to become standard in new vehicles in the future as they offer several advantages: They can be dynamically adapted to different driving situations. SbW systems also make new steering modes possible, such
as autonomous driving, in which the steering wheel moves automatically without the driver physically interacting with it. Further advantages can be found in [3][4][12]. The steering mechanism of SbW systems is similar to that of an EPS. The structure of a SbW system can be seen in Figure 2.3 on the right. As depicted, the only difference is that SbW systems no longer have a mechanical connection between the steering wheel and the front wheels. Thus, before the robust development of modern SbW systems is described in the next chapter, EPS systems will first be discussed in the next sections.

In the EPS considered here, the EPS motor for steering assistance (resp. the AU motor in the SbW system) is located in an axis-parallel configuration parallel to the front axle resp. the rack, as shown in Figure 2.3, because this configuration can transfer large rack forces and meets the highest dynamic requirements [6]. Nevertheless, the results presented in this thesis are also transferable to other configurations. In the axis-parallel configuration, the steering assistance generated by the EPS motor is transmitted to the rack via a combination of a ball screw drive and a toothed belt drive. The ball screw drive converts the rotational movement of the motor into a translational movement of the rack. A translational movement of the rack, in turn, results via the tie rods and levers in a rotational movement of the left and right front wheel about their steering axis. Via a pinion, the rack is additionally connected to the steering column and thus to the steering wheel. Therefore, a translational movement of the rack leads to a rotational movement of the steering wheel and vice versa. [3]

In previous developments, many of the dominant characteristics of a real EPS were neglected. For instance, in the design and validation of steering controls, models as in [13]-[18] with two degrees of freedom were often applied, which only represent the stiffness of the torsion bar and assume that all other connections between the individual bodies are rigid. In other models as in [19][20], in addition to the torsion bar, the connection of the EPS motor to the rack was also assumed to be elastic, but all other viscoelastic characteristics were still neglected. In more recent publications, such as [21]-[24], similar simplified steering models were used for both the design and the validation of the controls. However, these models do not reflect all dominant characteristics of a real steering mechanism. Consequently, the resulting steering controls cannot guarantee sufficient robustness in all driving situations. At certain operating points, existing steering controls even lead to an unstable control system. This can be confirmed by an analysis in the frequency domain of the control systems that represent the state of the art. Here, a conventional EPS is the plant. Figure 2.4 then depicts the Nyquist plots of the open-loop system when individual parameters are varied within the plant model. Information regarding Nyquist plots can be found in [25]. The open-loop system is created by cutting the closed-loop system at the control input of the plant model. The closed-loop system, in turn, consists of a linear detailed model of the EPS as
the plant model and a controller with an observer. The controller and the observer were designed with a simple model of the EPS, as is common in the state of the art.

Figure 2.4. Nyquist plots of the open-loop system with parameter variations in the plant model.

The curve $J_{\text{max}}$ illustrates the result for the case where the moments of inertia $J_{WL}$ and $J_{WR}$ of the left and right front wheel in the plant model are increased. This corresponds, for example, to the case when other wheels are mounted to the vehicle. On the other hand, the curve $i_{\text{max}}$ characterizes the case where the rack is maximally deflected. A deflection $s_R$ of the rack results in different values for the gear ratios $i_{RWL}$ and $i_{RWR}$ of the left and right wheel attachment due to the kinematics of the tie rod and lever linkages.

According to the simplified Nyquist criterion, a closed-loop system is stable if the Nyquist plot of the corresponding open-loop system does not orbit the critical point on the negative real axis at -1. The distance from the critical point to the Nyquist plot characterizes the robustness of the resulting control system. For the nominal case ($\text{nom}$), in which the parameters are not varied within the plant model, the control system appears to have sufficient robustness in the form of a satisfactory distance to the critical point. However, the Nyquist plots with varied parameters within the plant model ($J_{\text{max}}/i_{\text{max}}$) show that the control system actually does not have good robustness characteristics, since these plots orbit the critical point. Thus, the respective closed-loop system is unstable. Hence, the steering control becomes unusable in a wide range
around such operating points. Other driving situations and operating points can be found where this is the case.

Consequently, the steering control must be deactivated in the area surrounding such operating points. For conventional EPS systems, the driver then no longer receives any steering assistance from the EPS motor. Nevertheless, the vehicle can still be steered because of the mechanical connection between the steering wheel and the front wheels. However, this causes a significant loss of comfort for the driver of the vehicle. In [26]-[28], the results of a systematic model and system analysis by the author are utilized to develop approaches that solve these stability and robustness problems of EPS systems. This knowledge is further used to develop robust SbW systems. For modern SbW systems, the requirements for robustness are even stricter as there is no longer a mechanical connection between the steering wheel and the front wheels. As a result, it is not possible to control the vehicle’s lateral position if the steering control fails. This can lead to a considerably dangerous situation. Therefore, a guaranteed high robustness of the steering control in any driving situation is essential for SbW systems which will be integrated in modern vehicles. How this can be achieved is the subject of the following chapters.
3 Methodical Approach

SbW systems have become a key technology on the path to highly automated and autonomous driving and will therefore be used in modern vehicles. The advantages of SBW systems are outlined in [3][4][12]. The fundamental difference between them and conventional EPS systems is that SbW systems no longer have a mechanical connection between the steering wheel and the front wheels (see Figure 2.3). Due to this lack of a mechanical fallback, it is impossible to control the vehicle’s lateral position if the steering control of a SbW system malfunctions. This could lead to a considerably dangerous situation. Thus, ensuring high robustness of the steering control in any driving situation is essential for SbW systems. However, current control approaches cannot always guarantee such a high level of robustness. These approaches often show only limited robustness to degrees of freedom (DoF) and nonlinear characteristics of the plant that are neglected in the control design. To address this issue, this chapter describes the methodical approach for the development of SbW systems so that the resulting control systems have a guaranteed high robustness.

For this, a detailed model of a SbW system is developed that considers all relevant degrees of freedom and nonlinear characteristics that may occur in a real SbW system. This detailed model describes the real plant with a high degree of accuracy and is thus considered as a valid plant model. The detailed plant model is presented in Chapter 3.1. The analysis of the detailed model is afterwards described in Chapter 3.2. The model analysis is the basis for deriving optimal design models from the detailed model. The corresponding control design is then performed in Chapter 3.3. Subsequently, the control system consisting of the designed optimal controller and the detailed plant model is analyzed in Chapter 3.4.

3.1 Development of a Detailed Steer-by-Wire Model

The mechatronic development cycle begins with modeling. The aim of this step is to provide a mathematical description of a SbW system by converting the different phenomena (mechanical, electrical, algorithmic, ...) of the SbW system into a uniform representation based on equations. Therefore, an innovative nonlinear detailed model of a SbW system is developed in this chapter.
The individual process steps from the requirements to the model equations are shown for the mechanical components as an example in Figure 3.1. They follow the steps of the Newton-Euler method [29][30].

Figure 3.1. Process steps of the modeling.

The corresponding developed physical substitute model of the SbW system with nine degrees of freedom is shown in Figure 3.2. It contains all relevant characteristics of a real SbW system. In Figure 3.2, the bodies of the physical substitute model are shown in black, the viscoelastic elements in red and the gear ratios in gray. The model includes a steering wheel and a feedback actuator which are connected to each other via a torsion bar. Moreover, the model contains a front axle actuator. It is connected via a belt drive to a nut, which, in turn, is connected via a ball screw drive to a rack and finally via the tie rods and levers to the left and right front wheel.

The resulting detailed SbW model can be divided into a submodel for the feedback unit (FU) and a submodel for the axle unit (AU). Both units are mechanically decoupled (see the green dashed line in Figure 3.2). The FU and the AU are described in the following sections.
3.1.1 Development of a Feedback Unit Model

The FU consists of the steering wheel and a current-controlled feedback actuator. The feedback actuator is also called FU motor. It is used to generate a desired steering feel for the driver. The FU motor is a permanent-magnet synchronous motor with field-oriented control and can be divided into a mechanical part and an electrical part. The electrical part generates a torque $T_{MF}$ that acts on the mechanical part of the FU motor. The dynamics of the corresponding torque control can be modeled by a first order lag system with the differential equation

$$\tau_{MF} \hat{T}_{MF} + T_{MF} = T_{MF_rq}$$  \hspace{1cm} (3.1)$$

as shown in [31]. Here, $\tau_{MF}$ is the time constant of the FU motor control and $T_{MF_rq}$ is the requested FU motor torque. The mechanical part of the FU motor is modeled by a rotational mass with the moment of inertia $J_{MF}$, the viscous friction $b_{MF}$ and the angle $\phi_{MF}$. The steering wheel is also modeled as a rotational mass with the moment of inertia $J_S$, the viscous friction $b_S$ and the angle $\phi_S$. At the steering wheel, the steering torque $T_S$ is applied. Both the steering wheel and the FU motor are connected by a torsion bar [32]. The torsion bar

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$^1 \dot{x} = \frac{dx}{dt}$ symbolizes the time derivative of a variable $x$
is modeled by a viscoelastic element with the stiffness $c_{TB}$ and the damping constant $b_{TB}$. The resulting developed mechanical model of the FU is illustrated in Figure 3.3.

![Figure 3.3. Mechanical model of the feedback unit with two degrees of freedom.](image)

The bodies of the mechanical model are indicated in Figure 3.3 by the labels S (steering) and MF (feedback motor). The degrees of freedom of the mechanical model are represented by blue arrows, while the applied forces and torques are indicated by orange arrows.

The equations of motion of this mechanical model are derived using the Newton-Euler method (see Figure 3.1). For this, the free-body system, which exposes the forces and torques acting on the individual bodies of the model, is built from Figure 3.3. Based on this, the principle of linear and angular momentum for all bodies is formulated, which finally leads to the differential equations

$$J_S \dot{\Omega}_S = T_S - T_{TB} - b_S \Omega_S$$
$$J_{MF} \dot{\Omega}_{MF} = -T_{MF} + T_{TB} - b_{MF} \Omega_{MF}$$

for the mechanical part of the FU model. Here, $\Omega_S$ describes the angular velocity of the steering wheel and $\Omega_{MF}$ the angular velocity of the FU motor. Furthermore,

$$T_{TB} = c_{TB}(\phi_S - \phi_{MF}) + b_{TB}(\Omega_S - \Omega_{MF})$$

defines the torsion bar torque for $\phi_S > \phi_{MF}$ and $\Omega_S > \Omega_{MF}$. Hence, the mechanical part is modeled by a two-mass oscillator.
Equations (3.1) to (3.3) form the FU model. They can be combined to the state-space representation\(^2\)

\[
\begin{align*}
    \dot{x}_{FU}(t) &= A_{FU}x_{FU}(t) + B_{FU}u_{FU}(t), \quad x_{FU}(0) = x_{FU,0} \\
    y_{FU}(t) &= C_{FU}x_{FU}(t) 
\end{align*}
\]

(3.4)

of the FU model with

\[
A_{FU} = \begin{bmatrix}
0 & 1 & 0 & 0 & 0 \\
-c_{TB} & -b_{TB} + b_S & c_{TB} & b_{TB} & 0 \\
J_S & -J_S & J_S & J_S & 0 \\
0 & 0 & 0 & 1 & 0 \\
-c_{TB} & b_{TB} & -c_{TB} & -b_{TB} + b_{MF} & -\frac{1}{J_{MF}} \\
J_{MF} & J_{MF} & J_{MF} & J_{MF} & -\frac{1}{J_{MF}} \\
0 & 0 & 0 & 0 & -\frac{1}{\tau_{MF}} 
\end{bmatrix}, \quad (3.5)
\]

\[
B_{FU} = \begin{bmatrix}
0 & 0 \\
0 & 1 \\
0 & 0 \\
0 & 0 \\
1 & 0 \\
\frac{1}{\tau_{MF}} & 0 
\end{bmatrix}, \quad (3.6)
\]

\[
C_{FU} = \begin{bmatrix}
c_{TB} & b_{TB} & -c_{TB} & -b_{TB} & 0 \\
1 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 1 
\end{bmatrix}, \quad (3.7)
\]

and

\(^2\) Matrices are symbolized by upper-case letters with an underline. Vectors are symbolized by lower-case letters with an underline.
\[
\begin{bmatrix}
\varphi_{S}(t) \\
\Omega_{S}(t) \\
\varphi_{MF}(t) \\
\Omega_{MF}(t) \\
T_{MF}(t)
\end{bmatrix}, \quad
u_{FU}(t) =
\begin{bmatrix}
T_{MF_{rq}}(t)
\end{bmatrix}, \quad
y_{FU}(t) =
\begin{bmatrix}
T_{TB}(t) \\
\varphi_{S}(t) \\
T_{MF}(t)
\end{bmatrix}.
\] (3.8)

Here, \(A_{FU} \in \mathbb{R}^{5 \times 5}\) represents the system matrix, \(B_{FU} \in \mathbb{R}^{5 \times 2}\) the input matrix and \(C_{FU} \in \mathbb{R}^{3 \times 5}\) the output matrix of the FU model. Furthermore, \(\chi_{FU}(t) \in \mathbb{R}^{5}\) is the state vector, \(\chi_{FU, 0} \in \mathbb{R}^{5}\) is the initial state vector, \(u_{FU}(t) \in \mathbb{R}^{2}\) is the input vector and \(y_{FU}(t) \in \mathbb{R}^{3}\) is the output vector, which contains, for example, the control and measurement variables of the FU\(^3\).

### 3.1.2 Development of an Axle Unit Model

The structure of the AU is similar to the structure of the lower part of an EPS (see Chapter 2.4). It consists of the steering mechanism and a current-controlled front axle actuator. The front axle actuator is also called AU motor. Its task is to deflect the rack and thus the front wheels according to the steering wheel angle \(\varphi_{S}\). The AU motor is a permanent-magnet synchronous motor with field-oriented control and can be divided into a mechanical part and an electrical part. The electrical part generates a torque \(T_{MA}\) that acts on the mechanical part of the AU motor. The dynamics of the corresponding torque control can be modeled analogously to Chapter 3.1.1 using a first order lag system with the time constant \(\tau_{MA}\) of the AU motor control and the requested AU motor torque \(T_{MA_{rq}}\).

The mechanical part of the AU motor is modeled as a rotational mass with the angle \(\varphi_{MA}\). It is one component of the steering mechanism. A real steering mechanism is a complex mechatronic system with many degrees of freedom (see Figure 3.2). For instance, it contains rubber bearings and dampers to reduce impacts in the steering torque caused by road unevenness [8]. So, the nut is mounted in axial direction relative to the steering casing using rubber bearings [3]. The steering casing itself is also mounted viscoelastically. In addition, numerous components of a steering mechanism, such as the toothed belt, the balls within the ball screw drive or the tie rods, also have non-negligible viscoelastic characteristics. Moreover, the viscoelastic elements and gears possess partly nonlinear characteristics. In contrast to the simplified models for example from [16][17][24], the mechanical part of the developed AU model considers all these relevant characteristics of a real steering mechanism and has seven degrees of freedom. The equations of motion of this mechanical

---

\(^3\) The time dependence of the vectors is explicitly denoted the first time they are mentioned. For every further mention, the time dependence is omitted for better readability.
model are derived using the Newton-Euler method analogously to Chapter 3.1.1 resp. Figure 3.1. Alternatively, the equations of motion can be determined using the d’Alembert principle [33]. The large number of degrees of freedom leads to a mathematical model with numerous coupled differential equations and additional ordinary equations. Due to the size, the equations are not illustrated here.

By coupling the mechanical model with seven degrees of freedom and the model of the AU motor control, the detailed AU model, also referred to as the 7DoF model, is generated. It can be linearized and transformed to the state-space representation

\[ \dot{x}_{AU}(t) = A_{AU}x_{AU}(t) + B_{AU}u_{AU}(t) , \quad x_{AU}(0) = x_{AU,0} \]
\[ y_{AU}(t) = C_{AU}x_{AU}(t) \]  \hspace{1cm} (3.9)

with

\[ u_{AU}(t) = \begin{bmatrix} T_{MA,q}(t) \\ T_{WL}(t) \\ T_{WR}(t) \end{bmatrix} , \quad y_{AU}(t) = \begin{bmatrix} s_{R}(t) \\ \varphi_{MA}(t) \\ T_{MA}(t) \end{bmatrix}. \]  \hspace{1cm} (3.10)

Here, \( A_{AU} \in \mathbb{R}^{15 \times 15} \) represents the system matrix, \( B_{AU} \in \mathbb{R}^{15 \times 3} \) the input matrix and \( C_{AU} \in \mathbb{R}^{3 \times 15} \) the output matrix of the AU model. Moreover, \( x_{AU}(t) \in \mathbb{R}^{15} \) is the state vector and \( x_{AU,0} \in \mathbb{R}^{15} \) the initial state vector. The input vector of the AU model is denoted by \( u_{AU}(t) \in \mathbb{R}^{3} \) and the output vector by \( y_{AU}(t) \in \mathbb{R}^{3} \).

If the linearized detailed AU model from Equation (3.9) is combined with the FU model from Equation (3.4), the state-space representation

\[ \dot{x}_{SbW}(t) = A_{SbW}x_{SbW}(t) + B_{SbW}u_{SbW}(t) , \quad x_{SbW}(0) = x_{SbW,0} \]
\[ y_{SbW}(t) = C_{SbW}x_{SbW}(t) \]  \hspace{1cm} (3.11)

of the linearized detailed SbW model with

\[ A_{SbW} = \begin{bmatrix} A_{FU} & 0 \\ 0 & A_{AU} \end{bmatrix} , \]  \hspace{1cm} (3.12)

\[ B_{SbW} = \begin{bmatrix} B_{FU} & 0 \\ 0 & B_{AU} \end{bmatrix} , \]  \hspace{1cm} (3.13)
\[ C_{\text{SbW}} = \begin{bmatrix} C_{FU} & 0 \\ 0 & C_{AU} \end{bmatrix} \] (3.14)

and

\[ x_{\text{SbW}}(t) = \begin{bmatrix} x_{FU}(t) \\ x_{AU}(t) \end{bmatrix}, \quad u_{\text{SbW}}(t) = \begin{bmatrix} u_{FU}(t) \\ u_{AU}(t) \end{bmatrix}, \quad y_{\text{SbW}}(t) = \begin{bmatrix} y_{FU}(t) \\ y_{AU}(t) \end{bmatrix} \] (3.15)

follows. Here, \( A_{\text{SbW}} \in \mathbb{R}^{20 \times 20} \) is the system matrix, \( B_{\text{SbW}} \in \mathbb{R}^{20 \times 5} \) is the input matrix, \( C_{\text{SbW}} \in \mathbb{R}^{6 \times 20} \) is the output matrix, \( u_{\text{SbW}}(t) \in \mathbb{R}^{5} \) is the input vector and \( y_{\text{SbW}}(t) \in \mathbb{R}^{6} \) is the output vector of the detailed SbW model. Additionally, \( x_{\text{SbW}}(t) \in \mathbb{R}^{20} \) is the state vector and \( x_{\text{SbW},0} \in \mathbb{R}^{20} \) the initial state vector of the detailed SbW model. Further information regarding the development of this novel model can be found in Paper I. The methodology for modeling is also described in Papers IV and V. For the detailed SbW model, the results of a comprehensive model analysis are shown in the following section. These results form the basis for answering the first research question from Chapter 1.1.

### 3.2 Analysis of the Detailed Steer-by-Wire Model

A steering system has to meet specific requirements. To guarantee safe driving, good reference behavior must be ensured. For example, the rise time\(^4\) should be less than 0.1 s and the settling time\(^5\) should be less than 0.2 s with a maximum transient overshoot of 10 % in the case of step-shaped reference excitation. A small residual control error is acceptable, since steady-state accuracy is not mandatory. Furthermore, the system response to sinusoidal excitations up to 100 rad/s (16 Hz) should not deviate by more than 20 % from the amplitude of the excitation, since the driver is particularly sensitive in this frequency range. To allow comfortable driving, good disturbance behavior should also be ensured. For example, the settling time should be less than 0.2 s with correspondingly small amplitudes in the case of step-shaped disturbance excitation.\([3]\)

To investigate whether an uncontrolled SbW system already meets these requirements or how it behaves in general, a model analysis is performed as the second step of the mechatronic development cycle. The aim of the model analysis is to get a deep insight into the characteristics of a SbW system, which

---

\(^4\) The rise time is defined as the time until 90 % of the reference value is reached for the first time.

\(^5\) The settling time is defined as the time until the actual value is continuously within a tolerance band of 5 % around the reference value.
includes structure, stability, dominant behavior in the time and frequency domain, deficiencies, parameter dependencies and sensitivities. This insight allows the development of specific approaches for the improvement of the system characteristics. Therefore, the results of the conducted analysis of the detailed SbW model are shown in the following sections. The parameterization of the model is based on a representative steering system. The individual process steps from the model equations to the model characteristics are shown in Figure 3.4.

Figure 3.4. Process steps of the model analysis.

### 3.2.1 Analysis of the Feedback Unit Model

According to Chapter 3.1, the developed SbW model is divided into a sub-model for the FU and a submodel for the AU. In this section, the FU model is analyzed first.

Based on the approach of controlling the driver’s steering torque [34], the FU should give the driver a desired steering feel depending on the current driving situation [10][35]. According to Chapter 2.3, the steering torque \( T_S \) induced at the steering wheel corresponds to the steering feel experienced by the driver. At a constant steering angle \( \phi_S \), the steering torque \( T_S \) is equal to the torsion bar torque \( T_{TB} \). Since the steering torque \( T_S \) cannot be measured, the torsion bar torque \( T_{TB} \) is the controlled variable. The steering torque \( T_S \) itself represents a disturbance variable for a corresponding control. The control variable is the requested FU motor torque \( T_{MFreq} \). Figure 3.5 then shows the results of the model analysis of the control transfer path of the FU model from
Equation (3.4) in form of its step response (top left), its pole-zero map (bottom left) and its frequency response (right). The control transfer path is defined as the path from the control variable $T_{MFrq}$ to the controlled variable $T_{TB}$.

![Diagram of step response, pole-zero map, and frequency response.](image)

It is evident that the control transfer path is dominated by a second order lag behavior. This is also the case for the disturbance transfer path of the FU model, which is depicted in Figure 3.6. The figure shows the results of the model analysis of the disturbance transfer path in form of the step response (top left), the pole-zero map (bottom left) and the frequency response (right). The disturbance transfer path is defined as the path from the disturbance variable $T_S$ to the controlled variable $T_{TB}$. In both step responses, the second order lag behavior is visible by a weakly damped oscillation. The eigenfrequency of this oscillation at about 150 rad/s (24 Hz) can be described by the equation

$$\omega_0 = \sqrt{\frac{C_{TB}}{J_{res}}}$$  \hspace{1cm} (3.16)

with
\[ J_{\text{res}} = \frac{J_S J_{MF}}{J_S + J_{MF}}. \] (3.17)

Hence, the eigenfrequency depends on the stiffness \( c_{TB} \) of the torsion bar as well as on the moments of inertia \( J_S \) and \( J_{MF} \) of the steering wheel and the FU motor.

The low damping of the oscillation is also noticeable in the frequency responses by a resonance peak in the amplitude response and a steep phase drop in the phase response near the eigenfrequency. In the pole-zero map the second order lag behavior is indicated by a conjugate complex pole pair that is not compensated by zeros. In the disturbance transfer path, there is an additional uncompensated zero.

If the behavior of the FU model is compared with the requirements from the beginning of Chapter 3.2, it can be seen that the reference behavior with a rise time of 12 ms and a settling time of 0.22 s almost fulfills these requirements. However, the overshoot of 50 % is too high. Great amplitudes are also evident in the disturbance behavior and the settling time for disturbance excitation is too high as well. Consequently, the FU model has the major deficiency that the transient behavior is not sufficiently damped and particularly the disturbance behavior is unsatisfactory.
3.2.2 Analysis of the Axle Unit Model

In this section, the submodel for the AU of the detailed SbW model is analyzed. The AU converts a driver’s steering request into a corresponding deflection $s_R$ of the rack. Therefore, the deflection $s_R$ is the controlled variable and the requested AU motor torque $T_{M_{Arq}}$ is the control variable. Disturbance variables are the torques $T_{WL}$ and $T_{WR}$ about the steering axes of the left and right front wheel. These torques correspond to the tire forces and torques based on the current driving situation (see Chapter 2.3). Figure 3.7 presents the results of the model analysis of the control transfer path of the AU model from Equation (3.9) in form of its step response (top left), its pole-zero map\(^6\) (bottom left) and its frequency response (right). Here, the control transfer path is defined as the path from the control variable $T_{M_{Arq}}$ to the controlled variable $s_R$.

![Graphs](image)

Figure 3.7. Step response (top left), pole-zero map (bottom left) and frequency response (right) of the control transfer path of the AU model from requested AU motor torque $T_{M_{Arq}}$ to the deflection $s_R$ of the rack.

It is evident that the control transfer path is dominated by the superposition of an integral behavior, a first order lag behavior and a second order lag behavior. The integral behavior including the first order lag behavior results from the rigid body motion with viscous friction. The second order lag behavior results

---

\(^6\) For better visualization, the two residual high-frequency zeros of the control transfer path at about -58 000 s\(^{-1}\) and -61 000 s\(^{-1}\) are not shown in the pole-zero map of Figure 3.7.
due to the elastic coupling of the bodies. This is also the case for the disturbance transfer path of the AU model. Figure 3.8 shows the results of the model analysis of the first disturbance transfer path of the AU model, also in form of the step response (top left), the pole-zero map (bottom left) and the frequency response (right). The first disturbance transfer path is defined as the path from the disturbance variable $T_{WL}$ to the controlled variable $s_R$.

Figure 3.8. Step response (top left), pole-zero map (bottom left) and frequency response (right) of the disturbance transfer path of the AU model from the disturbance torque $T_{WL}$ at the left front wheel to the deflection $s_R$ of the rack.

Due to the symmetry of the steering mechanism, the results of the model analysis of the second disturbance transfer path of the AU model from the disturbance torque $T_{WR}$ to the controlled variable $s_R$ are identical to those from Figure 3.8. The two disturbance torques $T_{WL}$ and $T_{WR}$ are consequently often combined to generate only one disturbance variable in form of the rack force $F_R$.

The integral behavior including the first order lag behavior is particularly visible in the step responses. In both frequency responses, the second order lag behavior is observable by the first resonance peak in the amplitude response and a phase drop in the phase response at 150 rad/s (24 Hz). The corresponding lowest eigenfrequency can be approximated by the equation

$$\omega_0 \approx \sqrt{\frac{c_{res}}{J_{res}}}$$  \hspace{1cm} (3.18)
with

$$c_{res} = \frac{(c_{RWL} + c_{RWR})c_{MR}}{c_{RWL} + c_{RWR} + c_{MR}}$$

(3.19)

and

$$J_{res} = \frac{\left(\frac{J_{WL}}{i_{RWL}^2} + \frac{J_{WR}}{i_{RWR}^2}\right)\bar{J}_{MA}}{\frac{J_{WL}}{i_{RWL}^2} + \frac{J_{WR}}{i_{RWR}^2} + \bar{J}_{MA}}.$$

(3.20)

Here, $c_{RWL}$ and $c_{RWR}$ describe the stiffness of the left and right wheel attachment, while $c_{MR}$ denotes the equivalent stiffness between AU motor and rack. The stiffness $c_{MR}$ mainly results from the series connection of the stiffnesses of the axial nut bearing, the casing attachment and the ball screw drive. Additionally, $J_{WL}$ and $J_{WR}$ describe the moment of inertia of the left and right wheel and $\bar{J}_{MA}$ is an equivalent moment of inertia of the AU motor and the nut, transformed to rack coordinates. Furthermore, $i_{RWL}$ and $i_{RWR}$ define the gear ratio between the rack and the left resp. right front wheel.

Since in general $c_{RWL} + c_{RWR} \ll c_{MR}$ and $J_{WL}/i_{RWL}^2 + J_{WR}/i_{RWR}^2 \ll \bar{J}_{MA}$, the location of the lowest eigenfrequency, and thus the dominant behavior, depends mainly on the parameters of the front wheels and the wheel attachments. A stiffer connection $c_{RWL}$ and $c_{RWR}$ of the front wheels to the rack increases the eigenfrequency. Conversely, larger moments of inertia $J_{WL}$ and $J_{WR}$ of the left and right front wheel decrease it. A change in the gear ratio $i_{RWL}$ or $i_{RWR}$ between the rack and the left or right front wheel also primarily affects this eigenfrequency.

In addition, other stiffnesses, such as those of the axial nut bearing, the casing attachment, the ball screw drive, the bellows and the belt drive between the AU motor and the nut, only have a significant influence on the location of the high-frequency eigenfrequencies and therefore the residual behavior. In this context, the eigenfrequencies increase with increasing stiffness and vice versa.

When comparing the behavior of the AU model with the requirements of a steering system, it can be observed that the transient behavior is in general unsatisfactory due to the large time constant of the first order lag behavior and the low damping of the second order lag behavior. Moreover, the disturbance behavior is particularly insufficient, as a disturbance leads to a considerable decrease of the deflection $s_R$ of the rack. To eliminate these deficiencies of the AU as well as the deficiencies of the FU identified in Chapter 3.2.1, a control
will be developed in the next section. Further results of the performed analysis of the SbW model can be found in Paper I.

### 3.3 Design of an Optimal Multivariable Control

The third step of the mechatronic development cycle is the control design. The aim of this step is to extend the SbW system by a control so that the corresponding control system exhibits optimal behavior and fulfills all requirements. To achieve this, the identified model characteristics from Chapter 3.2 are used. The individual process steps from the model characteristics to the control system are illustrated in Figure 3.9.

![Figure 3.9. Process steps of the control design.](image)

As described in Chapter 3.2, the controlled variables of the SbW system are the torsion bar torque $T_{TB}$ within the FU and the deflection $s_R$ of the rack within the AU. The corresponding reference variables depend on the respective driving situation. Thus, the requested torsion bar torque $T_{TBreq}$ is mainly a function of the rack force $F_R$ and is computed within the steering feel generator. Conversely, the requested deflection $s_{Rreq}$ of the rack is a function of the steering wheel angle $\varphi_S$ and is computed within the position generator. The parameterization of both the steering feel generator and the position generator result from the requirements of the SbW system.

Current control approaches perform separate single-input single-output control designs for the torsion bar torque $T_{TB}$ and the deflection $s_R$ of the rack. The corresponding separately designed control systems are combined via the steering feel generator and the position generator. However, this causes shifts of the eigenvalues and changes of the dynamic behavior (time and frequency responses), so that the supposed high robustness achieved by the respective
design can no longer be guaranteed. Therefore, a novel multivariable control approach is presented in this chapter which results in improved and guaranteed robustness of the controlled SbW system. For this, a state-space controller is used, since it yields better results than classical controllers such as PID controllers or cascade controllers, as demonstrated in [3]. Further information regarding robust control design can be found, for example, in [36][37].

3.3.1 Development of a Reduced Steer-by-Wire Model
The control system must satisfy the requirements of good control and disturbance behavior as well as it must have a high degree of robustness against unconsidered eigenmodes and parameter uncertainties in the plant model. A precondition for good dynamic behavior is active vibration damping of the oscillating modes of the mechanical system. Therefore, a linear-quadratic-Gaussian (LQG) compensator will be designed which considers the natural limitations of the real system, ensuring that no bounds are exceeded during normal operation. However, the use of a high-order model for the compensator (controller and observer) design causes problems because its parameters often cannot be identified or vary substantially during operation. This is detrimental to the control, since it does not match the respective eigenmodes of the plant model sufficiently. Consequently, a compensator of the lowest possible order should be implemented. Nevertheless, the design model should not be reduced too much so that all dominant characteristics of the plant are represented. Thus, an optimally reduced SbW model is derived in the following section which will be used for the subsequent compensator design. The starting point is the state-space representation of the linearized detailed SbW model from Equation (3.11).

Real system excitations can have frequencies up to 190 rad/s (30 Hz) [3]. Consequently, it is important that the reduced model matches the detailed model well up to this frequency. As identified in Chapter 3.2, the eigenfrequencies due to the elastic torsion bar within the FU and the elastic wheel attachment within the AU are both about 150 rad/s (24 Hz). Therefore, the reduced SbW model should consider these elastic elements, while the remaining elastic elements can be neglected, since the associated eigenfrequencies are much larger than 190 rad/s (30 Hz). Then, the reduced SbW model with the state-space representation

\[
\begin{align*}
\dot{\tilde{\mathbf{x}}}_{SbW}(t) &= \tilde{\mathbf{A}}_{SbW} \tilde{\mathbf{x}}_{SbW}(t) + \tilde{\mathbf{B}}_{SbW} \mathbf{u}_{SbW}(t), \\
\mathbf{y}_{SbW}(t) &\approx \tilde{\mathbf{C}}_{SbW} \tilde{\mathbf{x}}_{SbW}(t)
\end{align*}
\]

(3.21)

is obtained. Here, \(\tilde{\mathbf{A}}_{SbW} \in \mathbb{R}^{10\times10}\) describes the system matrix, \(\tilde{\mathbf{B}}_{SbW} \in \mathbb{R}^{10\times5}\) the input matrix and \(\tilde{\mathbf{C}}_{SbW} \in \mathbb{R}^{6\times10}\) the output matrix of the reduced SbW model.
Moreover, $\hat{x}_{\text{SbW}}(t) \in \mathbb{R}^{10}$ is the state vector and $\hat{x}_{\text{SbW},0} \in \mathbb{R}^{10}$ the initial state vector of the reduced model. This reduced model is developed in a way that it answers the first research question from Chapter 1.1 in the best possible way.

### 3.3.2 Discretization of the Reduced Steer-by-Wire Model

For the subsequent discrete compensator design, it is necessary to discretize the reduced SbW model. The discretization converts the continuous model with the state-space representation from Equation (3.21) into a discrete model with the state-space representation

$$
\hat{x}_{\text{SbW},k+1} = A_{\text{SbW},D} \hat{x}_{\text{SbW},k} + B_{\text{SbW},D} u_{\text{SbW},k},
$$

$$
y_{\text{SbW},k} = C_{\text{SbW},D} \hat{x}_{\text{SbW},k},
$$

(3.22)

where the system matrix of the discretized reduced SbW model is denoted by $A_{\text{SbW},D} \in \mathbb{R}^{10 \times 10}$, the input matrix by $B_{\text{SbW},D} \in \mathbb{R}^{10 \times 5}$, the output matrix by $C_{\text{SbW},D} \in \mathbb{R}^{6 \times 10}$, the state vector by $\hat{x}_{\text{SbW},k} \in \mathbb{R}^{10}$ and the initial state vector by $\hat{x}_{\text{SbW},0} \in \mathbb{R}^{10}$. Further information regarding discretization can be found, for example, in [38].

### 3.3.3 Direct Discrete Controller Design

The developed reduced SbW model is a suitable approximation of the detailed SbW model. It is used as the plant model for the discrete design of a novel dynamic LQG compensator. The compensator contains a discrete linear optimal static state-space controller (LQR) and a discrete linear optimal state-space observer (LQE). Figure 3.10 shows the block diagram of the discrete closed-loop system consisting of the resulting discrete compensator and a plant model with zero-order-hold (zoh) element. Here, $\hat{x}_{pk,k} \in \mathbb{R}^{np}$ and $\hat{x}_{dk,k} \in \mathbb{R}^{pd}$ describe the a-posteriori estimates of the state vector and the disturbance input vector of the plant model with the number $np$ of state variables and the number $pd$ of disturbance input variables of the plant model, whereas $u_{pc,k} \in \mathbb{R}^{pc}$ and $u_{pd,k} \in \mathbb{R}^{pd}$ describe the control and disturbance input vector of the plant model with the number $pc$ of control input variables of the plant model. The vectors $y_{pm,k} \in \mathbb{R}^{qm}$ and $y_{po,k} \in \mathbb{R}^{qo}$ denote the measurement and objective output vector of the plant model with the number $qm$ of measurement output variables and the number $qo$ of objective output variables of the plant model, and $x_{r,k} \in \mathbb{R}^{qo}$ describes the reference vector.

---

7 State-space matrices of a discrete model are symbolized by an index $D$. The corresponding sequence of vectors $\hat{x}_k := \hat{x}(kT)$ are symbolized by an index $k$ for characterizing the point in time $kT$ with $k \in \mathbb{N}_0$ and the sample time $T$. 

39
The task of the LQR within the compensator is to adjust the torsion bar torque $T_{TB}$ and the deflection $s_R$ of the rack to their respective requested values $T_{TB_{req}}$ and $s_{R_{req}}$. For this, the requested motor torques $T_{M_{Frq}}$ and $T_{M_{Arq}}$ are the control variables. The steering torque $T_S$ and the rack force $F_R$ represent the disturbance variables for the control system. The effect of these disturbance variables on the controlled variables $T_{TB}$ and $s_R$ is compensated by a disturbance feedforward.

The direct discrete LQR design is based on the discrete reduced SbW model from Chapter 3.3.2 and the assumption that all its state variables and disturbance variables are measurable. For this direct discrete LQR design, the reduced plant model is augmented by suitable linear models for reference and disturbance excitation as well as a weighting model, so that an augmented plant model in a mixed deterministic and stochastic environment is obtained as the design model. Then, the discrete augmented plant model with the state-space representation

$$
\begin{align*}
\dot{x}_{k+1} &= A_D x_k + B_D u_k \\
y_{k} &= C_D x_k + D_D u_k
\end{align*}
$$

results, using the system matrix $A_D \in \mathbb{R}^{nxn}$, the input matrix $B_D \in \mathbb{R}^{nxp}$, the output matrix $C_D \in \mathbb{R}^{qxn}$ and the feedthrough matrix $D_D \in \mathbb{R}^{qxp}$ of the augmented plant model for the LQR design with the number $n$ of state variables, the number $p$ of input variables and the number $q$ of output variables of the augmented plant model. Here, the input vector $u_k \in \mathbb{R}^p$ of the augmented plant model is equal to the control input vector $u_{pc,k}$ of the plant model ($p = p_c$) and the output vector $y_k = x_{k} - y_{po,k} \in \mathbb{R}^q$ summarizes the control errors ($q = q_o$). Moreover, $x_k \in \mathbb{R}^n$ describes the state vector and $x_0 \in \mathbb{R}^n$ the initial state vector of the augmented plant model.

For this discrete model, an optimal state-space controller with the control law
\[ u_k = -Kx_k \] (3.24)

and

\[ K = (R + D_D^TQD_D + B_D^TSD_D)^{-1}(B_D^TSA_D + D_D^TQC_D) \] (3.25)

is designed. Here, \( K \in \mathbb{R}^{pxn} \) is the optimal controller gain matrix and \( S \in \mathbb{R}^{nxn} \) is the matrix that represents the positive definite solution of the associated algebraic matrix Riccati equation

\[
A_D^TSA_D - (A_D^TSD_D + C_D^TQD_D)(R + D_D^TQD_D) + B_D^TSD_D)^{-1}(B_D^TSA_D + D_D^TQC_D) - S + C_D^TQC_D = 0 \quad ,
\]

where \( R \in \mathbb{R}^{pxp} \) and \( Q \in \mathbb{R}^{qaq} \) are the positive (semi-)definite weighting matrices. They are the design parameters. The weighting matrix \( R \) penalizes the use of the control variables, whereas the weighting matrix \( Q \) penalizes the control errors. A simple and physically appropriate choice is to set the elements of \( R \) and \( Q \) equal to the reciprocal values of the corresponding maximum allowed variances [39]. A detailed description of the continuous control design can be found in Paper II and one of the direct discrete design in Paper III. Furthermore, the methodology for the optimal control design is also described in Paper IV.

### 3.3.4 Direct Discrete Observer Design

Following the LQR design, the LQE is designed to provide optimal estimates for the state and disturbance variables of the plant, since these are often not measurable. The starting point for the direct discrete LQE design is the discrete reduced SbW model from Chapter 3.3.2. This model is augmented by a suitable disturbance model to enable the estimation of disturbances and account for a stochastic environment [39]. Then, the discrete augmented plant model with the state-space representation

\[
\begin{align*}
\underline{x}_{k+1} &= A_D \underline{x}_k + B_D \underline{u}_k \\
\underline{y}_k &= C_D \underline{x}_k + D_D \underline{u}_k
\end{align*}
\] (3.27)
results, with the state-space matrices \( A_D \in \mathbb{R}^{n \times n} \), \( B_D \in \mathbb{R}^{n \times p} \), \( C_D \in \mathbb{R}^{q \times n} \) and \( D_D \in \mathbb{R}^{q \times p} \) of the augmented plant model for the LQE design\(^8\). The input vector \( u_k = [u_{pc,k}, u_{pd,k}]^T \in \mathbb{R}^p \) of the augmented plant model now combines the input vectors of the plant model \((p = p_c + p_d)\) and the output vector \( y_k \in \mathbb{R}^q \) is equal to the measurement output vector \( y_{pm,k} \) of the plant model \((q = q_m)\).

The objective of the direct discrete LQE design is to develop an observer that provides optimal estimates for the state vector \( \hat{x}_k = [\hat{x}_{p,k}, \hat{x}_{d,k}]^T \in \mathbb{R}^n \) of the augmented plant model in the presence of process and measurement noise. For the discrete observer, the state-space representation\(^9\)

\[
\begin{align*}
\hat{x}_{k+1} &= A_D \hat{x}_k + B_D u_k, \quad \hat{x}_0 = 0 \\
\hat{y}_k &= C_D \hat{x}_k + D_D u_k \\
\hat{\hat{x}}_k &= \hat{x}_k + \hat{K} (y_k - \hat{y}_k)
\end{align*}
\]  

is defined, where \( \hat{x}_k = [\hat{x}_{p,k}, \hat{x}_{d,k}]^T \in \mathbb{R}^n \) and \( \hat{\hat{x}}_k = [\hat{x}_{p,k}, \hat{x}_{d,k}]^T \in \mathbb{R}^n \) describe the a-priori and a-posteriori estimates and \( \hat{x}_0 \in \mathbb{R}^n \) the initial a-priori estimate of the state vector \( \hat{x}_k \) of the augmented plant model as well as \( \hat{y}_k \in \mathbb{R}^q \) the estimate of the measurement output vector \( y_k = y_{pm,k} \) of the plant model. Moreover, \( \hat{K} \in \mathbb{R}^{n \times q} \) is the optimal observer gain matrix. Through reformulation, the state-space representation

\[
\begin{align*}
\check{x}_{k+1} &= A_D (I - \hat{K} C_D) \check{x}_k + [B_D - A_D \hat{K} D_D] u_k \\
\check{x}_k &= (I - \hat{K} C_D) \check{x}_k + [-\hat{K} D_D \hat{K}] u_k
\end{align*}
\]  

of the observer with

\[
\hat{K} = \check{P}_e C_D^T (C_D \check{P}_e C_D^T + W)^{-1}
\]  

is obtained. Here, \( \check{P}_e \in \mathbb{R}^{n \times n} \) is the stationary covariance matrix of the a-priori estimation error. It is the positive definite solution of the associated algebraic matrix Riccati equation.

---

\(^8\) The state-space matrices from Equation (3.23) are unequal to the state-space matrices from Equation (3.27). Nevertheless, the same identifiers are used here to simplify the subsequent equations.

\(^9\) \( 0 \) symbolizes a zero matrix and \( I \) an identity matrix of corresponding size.
Moreover, $\mathbf{F} \in \mathbb{R}^{n \times p}$ is the noise filter matrix. The positive (semi-)definite intensity matrices $\mathbf{V} \in \mathbb{R}^{p \times p}$ and $\mathbf{W} \in \mathbb{R}^{q \times q}$ of the process and measurement noise are the design parameters. They have a similar meaning as the weighting matrices $\mathbf{R}$ and $\mathbf{Q}$ in the LQR design. The intensity matrix $\mathbf{V}$ of the process noise penalizes the use of the control variables of the plant model, whereas the intensity matrix $\mathbf{W}$ of the measurement noise penalizes the use of the measurement output variables of the plant model to estimate the state vector $\mathbf{x}_k$ of the augmented plant model. Suitable elements of the intensity matrices $\mathbf{V}$ and $\mathbf{W}$ can be found in [39]. A detailed description of the performed direct discrete design is given in Paper III.

### 3.4 Analysis of the Controlled Steer-by-Wire System

The fourth step of the mechatronic development cycle is the system analysis. In this step, it is reviewed if all requirements are actually fulfilled by the designed control. For the system analysis, the control system consists of the detailed SbW model from Chapter 3.1, which describes the real plant with adequate accuracy, and the novel control designed in Chapter 3.3 using the reduced SbW model. This enables the analysis of the robustness of the control system against eigenmodes of the real steering mechanism that were not considered in the design model. The control system is analyzed comprehensively in the time and frequency domain. The corresponding procedure is analogous to the one shown in Figure 3.4.

To simulate real driving situations, the closed-loop system is augmented by a linearized vehicle and tire model as well as a linear driver actuation model. The vehicle and tire model computes the torques $T_{WL}$ and $T_{WR}$ about the steering axis of the left and right front wheel, based on the angles $\phi_{WL}$ and $\phi_{WR}$ as well as the angular velocities $\Omega_{WL}$ and $\Omega_{WR}$ of the wheels. In contrast, the developed driver actuation model determines the steering torque $T_S$ that the driver induces at the steering wheel. This calculation uses the steering angle $\phi_{drv}$ requested by the driver as well as the angle $\phi_S$ and the angular velocity $\Omega_S$ of the steering wheel as input variables. A detailed description of these models can be found in [40].

Figure 3.11 and Figure 3.12 depict the step responses of the discrete closed-loop system in this augmented simulation environment from a steering angle $\phi_{drv}$ of 1 rad ($180/\pi$°) requested by the driver to the two controlled variables $s_R$ and $T_{TB}$. The driver’s steering request results in a rapid increase of the steering wheel angle $\phi_S$ to over 1 rad (60°) within 50 ms and so to a corresponding
requested deflection $s_{\text{Req}}$ of the rack. The magnitude of the requested deflection $s_{\text{Req}}$ depends on the virtual gear ratio parameterized in the position generator. The controller then adjusts the actual deflection $s_{R}$ of the rack to the requested deflection $s_{\text{Req}}$, yielding the time histories shown in Figure 3.11. It is evident that the controller can realize such highly dynamic steering requests. Simultaneously, the system response is adequately damped, despite the large number of viscoelastic elements within the steering mechanism. Thus, the actual deflection $s_{R}$ of the rack lies within a tolerance band of $\pm 5\%$ around the requested deflection $s_{\text{Req}}$ after approximately 30 ms. Furthermore, no significant overshoot can be detected. In this way, the control also ensures that the requirements for autonomous driving are fulfilled, as it enables the instantaneous and robust control of the lateral motion of a vehicle.

![Figure 3.11. Step response of the discrete closed-loop systems in the augmented simulation environment from the steering angle $\phi_{\text{drv}}$ requested by the driver to the deflection $s_{R}$ of the rack (orange) and the requested deflection $s_{\text{Req}}$ of the rack (blue).](image)

A deflection $s_{R}$ of the rack resp. the wheels causes a change of the torques $T_{WL}$ and $T_{WR}$ about the steering axis of the left and right front wheel and so a change of the equivalent rack force $F_{R}$. This leads to a requested torsion bar torque $T_{\text{TBrq}}$, which should provide the driver with feedback on the current driving situation. The magnitude of the requested torsion bar torque $T_{\text{TBrq}}$ depends on the parametrization of the steering feel generator. The controller then adjusts the actual torsion bar torque $T_{TB}$ to the requested torque $T_{\text{TBrq}}$, so that the time histories shown in Figure 3.12 are obtained. Due to the highly dynamic characteristic of the steering request, the driver actuation model initially generates

44
a high steering torque $T_S$, resulting in a significant torsion bar torque $T_{TB}$. However, the controller compensates this after a short time and adjusts the actual torsion bar torque $T_{TB}$ to the requested torsion bar torque $T_{TB,req}$, ensuring that the driver promptly experiences the desired feedback. As a result, the actual torsion bar torque $T_{TB}$ is already within a tolerance band of ±5% around the requested torque $T_{TB,req}$ after about 40 ms. Moreover, no significant overshoot is visible.

Figure 3.12. Step response of the discrete closed-loop systems in the augmented simulation environment from the steering angle $\phi_{drv}$ requested by the driver to the torsion bar torque $T_{TB}$ (orange) and the requested torsion bar torque $T_{TB,req}$ (blue).

By comparing the characteristics of the control system with the requirements from Chapter 3.2, it can be identified that the requirements are fulfilled. Additionally, all deficiencies of the SbW model have been eliminated. Hence, the developed control approach ensures good characteristics and answers the second research question from Chapter 1.1. The next step is to confirm these characteristics in the nonlinear simulation before the control is realized. This is the content of future work.

Further results of the conducted analysis of the control system can be found in Paper II and Paper III.
4 Summary and Conclusion

The development of autonomous driving systems requires sophisticated control approaches, simulation techniques and agile development strategies. Therefore, this thesis addressed the author’s research in the development and simulation of advanced technologies and control systems to enhance the performance and sustainability of vehicles. The related papers discuss various approaches for optimizing vehicle systems and processes in the automotive industry. Particular focus is placed on the control engineering and optimization of steering systems. In this context, modern Steer-by-Wire systems represent a key technology, as they enable highly automated and autonomous driving. Thus, an innovative multivariable control for Steer-by-Wire systems was developed, and its performance was comprehensively analyzed in virtual simulation. This required optimal models that reflect the dominant characteristics of a real Steer-by-Wire system (Paper I). These models provided a deep insight into the system and enabled the design of a highly robust steering control. For the control design, the entire environment in which a Steer-by-Wire system is embedded was taken into account, including, for example, the steering feel generator, which determines the desired steering feel for the current driving situation (Paper II and Paper III).

The resulting control system was afterwards comprehensively analyzed in an augmented simulation environment. For this purpose, an innovative vehicle model with multivariable control was developed (Paper IV). The knowledge gained from the development of the corresponding multi-body models and optimal controls was also applied to light electric vehicles. As an example, the model-based development of a cargo bike was carried out and a prototype for data collection was built, which allowed the verification of the fundamental vehicle model (Paper V). In addition, the data obtained can be used to develop further models as well as driver assistance systems in the virtual simulation.

The virtual simulation environment can be used not only for the development of Steer-by-Wire systems and the investigation of automated driving functions but also for the evaluation of sensor and perception systems (Paper VIII and Paper IX). So, the virtual simulation also enabled the realistic investigation of vehicle behavior and traffic interactions (Paper VI and Paper VII).

The result of these developments is an increase in efficiency and process optimization in the automotive industry. The use of virtual simulations and intelligent methods aims to reduce costs, improve the quality of development
and apply existing knowledge specifically for future applications. This is re-
filed in the development of predictive models, intelligent analysis methods
and the utilization of AI for forecasting (Paper X and Paper XI).

The related papers illustrate the development of new technologies and con-
trol systems to increase the safety, efficiency and innovation of vehicles. Con-
trol engineering, driver assistance systems, virtual simulations and process op-
timization are key areas of current research to contribute to the advancement
of the automotive industry. The common aspects shared across the papers
highlight the interdisciplinary nature of the research and the necessity to inte-
grate different disciplines for creating forward-looking solutions in the auto-
motive industry. In conclusion, the author’s research contributes to the ad-
vancement of autonomous driving technologies through model-based design,
robust control, virtual simulation, the integration of human interaction and un-
certainty management. This multidisciplinary approach enables a multiobjec-
tive optimization at an early stage of the product development, allowing pa-
rameter variations based on circular economy requirements to be incorporated.
Additionally, it reduces time- and cost-intensive testing on prototypes, avoids
unnecessary iterations in the design and significantly increases the efficiency
and quality of the development. Moreover, it is essential for developing novel
autonomous driving systems in modern vehicles.
5 Future Work

This thesis presented a model-based design of a SbW system in accordance with the mechatronic development cycle from Figure 1.2. First, a model of the SbW system was developed and analyzed. Based on this, a control design was conducted. Finally, the resulting control system was analyzed. The next step is to perform a more realistic analysis of the system in nonlinear simulation. This requires the development of an augmented simulation environment containing a vehicle model with tire models as well as a driver actuation model similar to [40]. Preliminary research on the development of the vehicle model is published in Paper IV and Paper VIII. The driver actuation model will be published in a next paper. The augmented simulation environment is then used not only for a realistic simulation of real driving situations, but also for the development of HiL test benches and prototypes. This is also the subject of future work. The methodology for this is published in Paper V and [41].

In the realization step of the mechatronic development cycle, a prototype of the system will be built. The mechanical and electrical parts will be constructed, manufactured and assembled. This work is usually very time-consuming and cost-intensive. Therefore, it should be carried out only once if possible. Moreover, the signal processing part will be programmed and implemented on suitable powerful hardware. This step can be largely automated by the tools of computer science. Ideally, the execution of real experiments will be the last step of the mechatronic development cycle. Here, the same experiments will be performed on the real prototype as in the simulation. If the measured time histories match the time histories from the simulation, mechatronic the development cycle ends with an optimal prototype.
6 Summary of Papers

This chapter summarizes the content of the papers on which this thesis is based upon and describes the author’s contribution to each paper.

Paper I

Development and Analysis of a Detail Model for Steer-by-Wire Systems
This paper presents an innovative nonlinear detailed model of a Steer-by-Wire system. The detailed model represents all characteristics of a real Steer-by-Wire system. In the context of a dominance analysis of the detailed model, all dominant characteristics of a Steer-by-Wire system, including parameter dependencies, are identified. Through model reduction, a reduced model of the Steer-by-Wire system is then developed, which can be used for a subsequent robust control design. Furthermore, this paper compares the Steer-by-Wire system with a conventional electromechanical power steering and shows similarities as well as differences.

The author developed the detailed model of a Steer-by-Wire system, performed the analysis of the resulting model and identified the dominant characteristics of a Steer-by-Wire system as well as he developed optimal reduced models. In addition, he compared a Steer-by-Wire system with an electromechanical power steering. Moreover, the author wrote the paper.

Published in IEEE Access Journal in January 2023.

Paper II

Design of a Robust Optimal Multivariable Control for a Steer-by-Wire System
This paper presents a new control approach for modern Steer-by-Wire systems. The approach consists of a multivariable control for the driver’s steering torque and the rack position simultaneously, using the requested torques of the downstream and upstream motors as control variables. The plant model used in this approach is a detailed model of a Steer-by-Wire system with nine degrees of freedom. For the control design, an optimal reduced model is derived. The reduced plant model is linearized, and it is augmented by linear models
for the reference and disturbance environment of the Steer-by-Wire system and by a linearized model for the feeling generator, which computes the requested steering torque. For this augmented model, a multivariable linear optimal static state-space controller is designed. Hence, the entire environment of the real steering system is considered in the control design. Due to the multivariable approach and the augmented model containing all subsystems and dominant characteristics of the real system, the resulting control system shows excellent robustness characteristics.

The author developed the model of the reference generator, performed the optimal multivariable control design and analyzed the resulting control system. Moreover, the author wrote the paper.

*Published in SAE Technical Paper, presented orally by the author in July 2023, Stuttgart, Germany.*

Paper III

**Direct Discrete Design of a Multivariable LQG Compensator with Combined Discretization Applied to a Steer-by-Wire System**

This paper presents a direct discrete control design for modern Steer-by-Wire systems. The novel approach consists of a true multivariable control for both the driver’s steering torque and the rack position simultaneously, using the requested torques of the downstream and upstream motors as control variables. For the control design, an optimal reduced plant model is used. It is derived from a detailed model of a Steer-by-Wire system with nine degrees of freedom. The reduced plant model is augmented by linear models for the reference and disturbance environment of the Steer-by-Wire system and discretized based on the characteristics of the input variables. For this augmented model, a direct discrete multivariable linear quadratic Gaussian compensator is designed. The proposed control design considers the entire environment of the real steering system. The direct discrete approach restores the good characteristics of the continuous control and ensures that discretization does not have any adverse effects. Thus, the resulting discrete control system shows the same favorable dynamic characteristics as the continuous system.

The author developed a novel discretization method, performed the direct discrete control design and analyzed the resulting control system. Moreover, the author wrote the paper.

*Published in Proceedings of Automotive meets Electronics, presented orally by the author in June 2023, Dortmund, Germany.*
Design of a Model-Based Optimal Multivariable Control for the Individual Wheel Slip of a Two-Track Vehicle

This paper presents a model-based optimal multivariable control for the wheel slip, which allows specifying the wheel slip and thus the tire force individually for each wheel. The plant model consists of a multibody two-track model of a vehicle, a tire model, an air resistance model and a motor model. In addition, the contact forces of the individual wheels are calculated dynamically. The resulting nonlinear model is linearized and used for the design of a linear optimal static state-space controller with reference and disturbance feedforward. The contact point velocities at the wheels are defined as the controlled variables, since they are proportional to the wheel slip and thus to the driving forces within the operating range of the controller. Furthermore, the rates of change of the contact point velocities are also chosen as controlled variables to set the damping of the closed-loop system. The four drive torques of the wheels represent the control variables. Therefore, a true multivariable control is developed. In the first step of the analysis, the linearized closed-loop system is investigated regarding stability, robustness and its dynamic behavior. The control system shows a high bandwidth, well-damped dynamic behavior and a large phase margin. In the second step of the analysis, various simulations of realistic experiments, such as an accelerated cornering maneuver or the Fishhook road test, are performed with the nonlinear closed-loop system. The results of these experiments confirm the high robustness and good dynamic behavior of the control system in most cases. Moreover, the results demonstrate how the control considers the dynamic contact forces of the wheels to achieve the requested wheel slip at any time. Lastly, dominant transfer paths are identified based on the gain matrix of the state-space controller, showing which input and state variables have a significant influence on the control variables.

The author developed the multibody model of the vehicle, an approach to approximate the contact forces and designed an optimal wheel slip control. Moreover, the author elaborated the majority of the paper.

Published in SAE Technical Paper, presented orally by Robert Rosenthal in July 2023, Stuttgart, Germany.
Paper V

**Methodical Data Collection for Light Electric Vehicles to Validate Simulation Models and Fit AI-based Driver Assistance Systems**

This paper presents an approach to collect vehicle dynamic parameters for the validation of simulation models. For this purpose, a measurement system is developed to capture and monitor driving dynamic information of the device under test in real time. This data is used to fit pre-developed simulation models and DAS. To investigate the vehicle dynamic behavior in critical driving situations, an extensive test study is conducted. Therefore, different ordinary driving situations in urban traffic scenarios are analyzed. Finally, the collected measured data is compared with the simulation results of a multi-body model for a multi-lane cargo vehicle.

The author developed the simulation model, the measurement setup and the verification study. Additionally, he supervised the realization of the study. Moreover, the author wrote most parts of the paper.

*Published in Proceedings of Kolloquium Future Mobility in June 2022, Ostfildern, Germany.*

Paper VI

**Integration of Vulnerable Road Users Behavior into a Virtual Test Environment for Highly Automated Mobility Systems**

This paper describes an approach to integrate real human traffic behavior into the approval and testing process of highly automated vehicle systems. It provides a safe and valid way to test critical traffic scenarios between vehicles and pedestrians. Basically, two different methodologies for the metrological detection of human movements are analyzed and experimentally examined for their suitability for this use case. Besides the general functionality, plausibility and realtime capability are further investigation criteria. The paper concludes with the integration of the proposed solution into a test bed for highly automated vehicle systems using a representative traffic scenario.

The author was involved in discussions, supported implementation and assisted in writing the paper.

*Published in Proceedings of Kolloquium Future Mobility in June 2022, Ostfildern, Germany.*
Methodical Approach to Integrate Human Movement Diversity in Real-Time into a Virtual Test Field for Highly Automated Vehicle Systems
This paper measures, processes and integrates real human movement behavior into a virtual test environment for highly automated vehicle functionalities. The overall system consists of a georeferenced virtual city model and a vehicle dynamics model, including probabilistic sensor descriptions. By using motion capture hardware, real humanoid behavior is applied to a virtual human avatar in the test environment. Through retargeting methods, the virtual avatar diversity is increased. To verify the biomechanical behavior of the virtual avatars, a qualitative study is performed, which is based on a representative movement sequence.

The author was involved in discussions, supported implementation and assisted in writing the paper.

Published in Journal of Transportation Technologies in July 2022.

Data Flow Management Requirements for Virtual Testing of Highly Automated Vehicles
This paper presents a virtual co-simulation approach for highly automated vehicle systems and uses it to demonstrate the data management requirements for a co-simulation platform such as AVL Model.CONNECT™. The basis for this is a real urban driving cycle for modern hybrid vehicles to investigate emissions, consumption and range as well as the effects of highly automated driving functions on these parameters.

The author was involved in discussions, supported conducting the study and assisted in writing the paper.

Published in Proceedings of AVL German Simulation Conference and presented orally by René Degen in September 2022, Regensburg, Germany.
Paper IX

**Stereoscopic Camera-Sensor Model for the Development of Highly Automated Driving Functions within a Virtual Test Environment**

This paper documents the development of a sensor model for depth estimation of virtual three-dimensional scenarios. For this purpose, the geometric and algorithmic principles of stereoscopic camera systems are recreated in a virtual form. The model is implemented as a subroutine in the Epic Games Unreal Engine. Its architecture consists of several independent procedures which enable a local depth estimation and a reconstruction of an entire three-dimensional scenery. In addition, a separate program for calibrating the model is presented.

The author was involved in discussions, assisted in writing the paper and supported implementation as well as evaluation.

*Published in Journal of Transportation Technologies in January 2023.*

Paper X

**Intelligent Analysis of Components with Regard to Significant Features for Subsequent Classification**

This paper develops an intelligent method to analyze existing data appropriately and, at the same time, prepare it ideally for further applications, such as forecast models based on Artificial Intelligence. To achieve this, several steps need to be taken. Firstly, a suitable segmentation of the component is performed. The aim is to detect areas in a component where features and form elements are found. Other regions are ignored after the inspection by segmentation and voxelization. Subsequently, the voxelization of the component takes place, which results in the three-dimensional component or Computer-Aided-Design file being mathematically readable. This is done by rasterizing the component based on a previously selected resolution and other upcoming steps. Finally, the segmented and relevant areas are analyzed accordingly.

The author was involved in discussions and assisted in writing the paper.

*Published in SAE Technical Paper, presented orally by Alexander Nüßgen in July 2023, Stuttgart, Germany.*
Robustness and Sensitivity of Artificial Neural Networks for Mechatronic Product Development

This paper aims to evaluate the performance characteristics of different uncertainty analysis methods and assess their applicability in agile automotive development processes. By considering the specific requirements and constraints of each method, a decision tree is proposed to recommend suitable and situation-appropriate methods for performing uncertainty analyses in network prediction. The goal is to enhance data exchange between departments, mitigate disruptions, and ensure informed decision-making throughout the development process.

The author was involved in discussions and assisted in writing the paper.

Published in Proceedings of Automotive meets Electronics and presented orally by Alexander Nüßgen in June 2023, Dortmund, Germany.
Bilindustrin genomgår just nu en stor förändring, vilket har sin grund i behovet av ökad automatisering och att resursanvändning och utsläpp måste minskas. Sett från ett teknologiskt perspektiv innebär den här förändringen många utmaningar och möjligheter. Optimal styrning av komponenter är nödvändig för en hållbar och miljömedveten användning av fordon. Därtill är stabil kontroll av ställdon fundamentalt för utvecklingen av förarstödsystem i fordon och funktioner för självkörande fordon. Styrsystemets ställdon är av särskild betydelse, eftersom de möjliggör för säker och komfortabel kontroll av fordon. Därför har i denna uppsats utvecklingen av modeller samt virtuell simulering av en innovativ och mycket stabil kontrollmetod för styrsystem i fordon där det inte finns någon mekanisk koppling mellan ratten och hjulen (så kallad Steer-by-Wire system) studerats.


Sammanfattningsvis, forskningsresultaten som är beskrivna i den här avhandlingen bidrar till utvecklingen av nya och moderna Steer-by-Wire system, designen av vilka utgör grunden för funktioner för högautomatiserade och självkörande fordon.
8 Acknowledgement

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Furthermore, I would like to thank my colleagues. Their shared experiences and collaborative spirit have been an invaluable source of strength. Finally, I would like to thank my family, friends and above all, my wife, for always standing by me.
9 References


