CHALMERS

Development of the control system for an electric vehicle

Master of Science Thesis

MINAS ROUKAS

Department of Signals and Systems
Division of Automatic Control, Automation and Mechatronics
CHALMERS UNIVERSITY OF TECHNOLOGY
Göteborg, Sweden, 2013
Report No. EX050/2013

Chalmers University of Technology

Master Thesis

Development of the Control System for an Electric Vehicle

Author:
Minas Roukas

Supervisor:
Nicolas Soulier

Examiner:
Prof. Maben Rabi

*A thesis submitted in fulfilment of the requirements
for the degree of Master of Science at
'Systems, Control and Mechatronics'*

in the

Signals and Systems Department

October 2013
Report number: EX050/ 2013

Abstract

Development of the Control System for an Electric Vehicle

by Minas Roukas

In Volvo Group Trucks Technology, an in-house project started in order to develop an electric one-seater demonstration vehicle. The final product, apart from demonstration reasons, will also be used as a testing platform. The present report is a part of this Volvos project. Its scope is first to structure the control system for the one-seater vehicle, then to implement a subset of the requested functionalities, to validate the designed control strategy and finally to document the design using the EAST- ADL architecture description language.

The focused functionality for this thesis is the control of the propulsion system. For a road vehicle the three main motions, which have to be controlled, are the longitudinal and lateral speed as well as the yaw motion. Furthermore, the actuators, which can be controlled and contribute to the movement of the specific vehicle are two electric motors for the rear wheels and four brakes, one in each wheel. On the other hand, the steering actuation is not included in the control system and the driver can directly influence the front wheels' steering angle. Having more actuators, which act as controlled inputs of the system, than controlled motions, which are the control outputs, allows the writer to use a control method, which can be used for over-actuated systems, called control allocation. Moreover, a conceptual model, developed by Volvo, was used in order to structure the control architecture.

The implementation of the control system was made in Matlab/Simulink and several simulation tests are made in order to validate the functionality of the designed system. An already made vehicle model was adjusted and used during the simulations.

The results from the simulation tests validated that the designed control system could handle the propulsion of the vehicle. Thus, a first version of the control system is delivered and could be enriched with other functionalities in order to accomplish Volvo's project.

Key words: *electric vehicle, control system, EAST- ADL language, over-actuated systems, control allocation, CVC architecture*

Acknowledgements

This master thesis has been conducted at Volvo Group Trucks Technology and it is a part of the examination of the master program "Systems, control and mechatronics" at Chalmers University.

First of all, I would like to thank my supervisor in Volvo, Nicolas Soulier, for giving me the opportunity to work on such an interesting topic and for his valuable help and guidance throughout this project. Henrik Hallberg and Dhasarathy Parthasarathy for the interesting discussions and suggestions, as well as all the other members participated in the EDV project. Many thanks to Henrik Lönn for his assistance with the EAST-ADL and Enterprise Architecture.

I would also like to thank my Chalmers' examiner Maben Rabi. He has been supportive and his advices and feedback proved to be very helpful. Additionally, Kim Viggedal for his great job as opponent.

Moreover I would like to thank all my friends, who have been encouraging and supporting me during my stay in Sweden.

Last, but not least, I would like to specially thank my parents for giving me the chance to study abroad. Without them I would not be able to fulfil my master studies in Chalmers.

October 2013
Minas Roukas

Contents

List of Figures

List of Tables

Abbreviations

ADL	**A**rchitectural **D**escription **L**anguage
BLDCM	**B**rushless **DC** **M**otor
CoG	**C**enter **o**f **G**ravity
CVC	**C**omplete **V**ehicle **C**ontrol
DM	**D**riving **M**ode
DOF	**D**egrees **O**f **F**reedom
BP	**B**rake **P**edal
EA	**E**nterprise **A**rchitecture
EDV	**E**lectric **D**emonstration **V**ehicle
EV	**E**lectric **V**ehicle
FAA	**F**unctional **A**nalysis **A**rchitecture
GP	**G**as **P**edal
HEV	**H**ybrid **E**lectric **V**ehicle
HMI	**H**uman **M**ahine **I**nterface
HW	**H**ard**w**are
ICE	**I**nternal **C**ombustion **E**ngine
OOA	**O**bject-**O**riented **A**nalysis
RA	**R**eference **A**rchitecture
RCS	**R**eal-time **C**ontrol **S**ystem
SAE	**S**ociety of **A**utomotive **E**ngineers
SW	**S**oft**w**are
SWA	**S**teering **W**heel **A**ngle
UML	**U**nified **M**odeling **L**anguage

Chapter 1

Introduction

This is the introduction chapter of the present thesis report. It gives some general information about the background and the motivation for the current project. The working framework as well as the expected outcome are also mentioned.

1.1 Background

In general, as the population on earth increases and the life-standards are improved, the density of vehicles increases as well [1]. This increase in the number of vehicles, along with the capability of constructing more and more powerful vehicles, leads to a higher demand for energy consumption. Using fossil fuel to cover the energy demands provokes an increase of environmental pollution, as well as a decrease of the available natural resources. While the available quantity of fossil fuel declines significantly, there is no doubt that resources like petroleum, if the present rate of energy consumption is considered, will vanish in some decades. As a result, the transportation costs will become very high.

For the reasons mentioned above, the usage of sustainable energy resources becomes a vital factor for automotive industry. One suggested solution is the development of electric vehicles (EV). Contrary to conventional vehicles, an electric one uses electric motors for propulsion. EV is not a recent discovery, it was first introduced in the mid of 19th century. Despite its long existence, it never became as popular as vehicles with internal combustion engine (ICE). Some main reasons for that are the limitations in travelling range without the need to recharge and in velocity. Nevertheless, during recent decades, since sustainable solutions become more and more important and technological improvements assist in overcoming the original drawbacks, the interest for this kind of vehicles sharply increases.

There are three main types of EVs [2]. The first uses electric power given directly to it by an external power station, the second uses rechargeable batteries to store and provide electric power. The third type uses rechargeable batteries, but also includes an ICE. This type is called hybrid electric vehicle (HEV) and uses the ICE in order to charge the battery or increase the available power for the propulsion system.

1.2 Motivation

The Volvo Technology company started an in-house project for developing an electric demonstration vehicle (EDV). The main goals for this project are two:

- Use the constructed EDV as a demonstration vehicle and as a platform for testing new technologies.
- Increase the employees' competences in designing an HEV from scratch.

The EDV will be approximately at the size of a go-kart vehicle. The propulsion system will consist of two electric motors, one at each rear wheel. Although the motors will be powered primarily from a battery, an ICE electrical generator will also be included. The output of ICE will be converted to electrical power and then applied to the motors of the battery. The braking system will be electro-hydraulic and will consist of four brakes, one for each wheel. Regarding the steering system, it will be steer-by-wire.

The present thesis report is only a small part of the Volvo's project for the development of the EDV. It will focus on designing the control system of the vehicle and more precisely the propulsion system. Considering that Volvo's project is at the beginning, the thesis work will include the following tasks:

- Identify and characterize vehicle features and break them down into requirements.
- Structure the control system according to a specific reference architecture for matching Volvo standards.
- Adapt and enhance existing control functions or develop new in order to match the requirements.
- Validate and verify functionalities of the control system using simulations.

1.3 Limitations

In order to overcome some predicted obstacles and difficulties during the thesis, the following limitations help to delimit the working framework.

- The design of the control system will focus on the propulsion system of the EDV.
- Steering will not be included in the control system, nevertheless driver's steering commands will be directly applied to the wheels, through a fully mechanical steering system.
- Estimated values for vehicle's components, that are not yet decided, will be used.
- Neither sensors models nor actuator models will be included.

1.4 Expected Outcome

After the completion of the thesis work, the expected outcome should consist of:

- A first version of a functional control system for the propulsion system of the EDV.
- Documented design of the control system using EAST-ADL architecture description language.

1.5 Thesis Outline

This report is structured into 8 chapters. After this introduction, Chapter 2 presents the theory for the designing reference architecture and the architecture description language called EAST- ADL. In Chapter 3, the vehicle's configuration is presented, along with theory for modelling different parts of the car. In the next chapter, the theory for the control system is explained and formed based on the previous chapter's results. In Chapter 5, the control system and the different models are implemented with the EAST-ADL language, whereas in Chapter 6 are implemented as a Simulink model. The results of the simulation tests are found in Chapter 7. Finally, some discussion about the results and suggestions for future work are included in the last chapter.

Chapter 2

Designing Architecture and Documentation

In this chapter the theoretical background about reference architecture is presented. Different definitions are given, in order for the reader to be able to follow the control designing methodology.

2.1 Concept of Reference Architecture

2.1.1 Background about Reference Architecture

In this section, the necessary information for understanding the concept of the Reference Architecture (RA) is presented. The RA will be implemented based on object-oriented programming. Thus some basic definitions are presented below:

An **Object** is an instance of a class, which has the variables of the class uniquely defined [3].

Object oriented programming is a programming approach that interprets different concepts as "objects" [3].

An **Object-oriented system** is composed of different objects which collaborate with each other.The collaboration between the objects means that they can send messages each other. After a message is received, it is up to the receiver to decide what function to use, depending of its state, to service that message.The collaboration of the objects of a systems defines the behaviour of the system [3].

Object-oriented analysis is the process when a task is analysed in order to develop a conceptual model, which can be used to complete that task. During this phase, the problem is usually decomposed and modelled in subtasks, also called domains, focusing on "what the system does" [3].

Object-oriented design is the phase when constraints to the conceptual model produced during OOA are applied. The focus now is on "how the system does it" [3].

2.1.2 Concept of Reference Architecture

In general, the idea of RA is not completely solidified. For that reason, this section gives the definition for some terminology and explains RA's usefulness.

Architecture: The fundamental organization of a system embodied in its components, their relationships to each other, and to the environment, and the principles guiding its design and evolution [4].

An **Architecture Framework** provides guidance and rules for structuring, classifying, and organizing architectures [4].

A **Design Pattern** systematically names, motivates, and explains a general design that addresses a recurring design problem in a system. It describes the problem, the solution, when to apply the solution, and its consequences. It also gives implementation hints and examples. The solution is customized and implemented to solve the problem in a particular context [4].

View: A representation of a whole system from the perspective of a related set of concerns [4].

Viewpoint: A specification of the conventions for constructing and using a view. A pattern or template from which to develop individual views by establishing the purposes and audience for a view and the techniques for its creation and analysis [4].

As mentioned earlier, although RA lacks solidity, there are some trends that imposes the usage of RA. The first trend is the increasing complexity and size of a system and the organization that is creating it. The second is the increasing demands in integration. Shorter time to market, rapid changes and adaptation are required. A Reference Architecture can provide to a small group of people up to different departments or even different companies, the following features:

- A common lexicon and taxonomy.
- A common architectural vision.
- Modularization.

With the common lexicon and taxonomy, communication between multiple people and departments is getting easier. The common vision aligns efforts of different people or teams and the modularization helps in the division of the work [4]. A strong tool that makes RA more effective is patterns. A pattern helps in documenting the knowledge of an individual in a way that, it will be reusable for other designs than the original and by more developers than the creator.

2.1.3 Designing Pattern

For designing the architecture of the EDV, a pattern designed by Volvo was given. This pattern is called Complete Vehicle Control (CVC), is based on the [5] and provides a conceptual model for control architecture. A basic overview is shown in figure 2.1, where it can be observed that the architecture is separated in different hierarchical layers. Each layer can be seen as a functionality domain that encapsulates several systems. The reason for partitioning the functionalities is the complexity of vehicles and the different demands in execution times. The lowest level has a very narrow horizon of some milliseconds, thus it has to be prioritized in order to be executed on time. On the contrary, higher levels can have an horizon of over sixty minutes. One more reason for having separated functionality systems is that, in this way the re-usability of the different functionalities is facilitated, leading in reducing the time spent in designing. Moreover, based on the CVC pattern, there are some principle rules that the designer should respect and they are mentioned below:

1. Set points or requests are not allowed between objects of functionality within the same layer.
2. Set points or requests shall be given from a layer with longer horizon to a shorter horizon layer and not vice versa.
3. Each layer shall provide its aggregated nominal and dynamic capabilities and limitations only to the next layer with higher horizon.
4. Status signals, model parameters, states and limitations may be shared within different functional areas or objects within the same layer.
5. The HMI should provide information of the driver's intentions rather than set points to all the different levels.

This set of rules ensures that the CVC pattern will have a standardised form, which will facilitate the modularity of the design and the separation of the functionalities based on the time frames. Furthermore, neither disadvantages nor limitations in the capabilities of the controller were found, despite the use of the previous rules in the design.

The levels in which this thesis was focused were primarily the "Vehicle Motion Management" and secondary the "Motion Support Devices".

Vehicle Motion Management

The most important characteristic for this domain is that it deals with continuous dynamic behaviour of the vehicle as a whole system. It is a domain where a failure can result in significant physical damage or economics losses or even threat to human life. No mechanical parts like actuators or sensors are included in this level, only software that controls and defines set points for the lower level' s actuators. This is the level where most of the controller for the propulsion system will be designed.

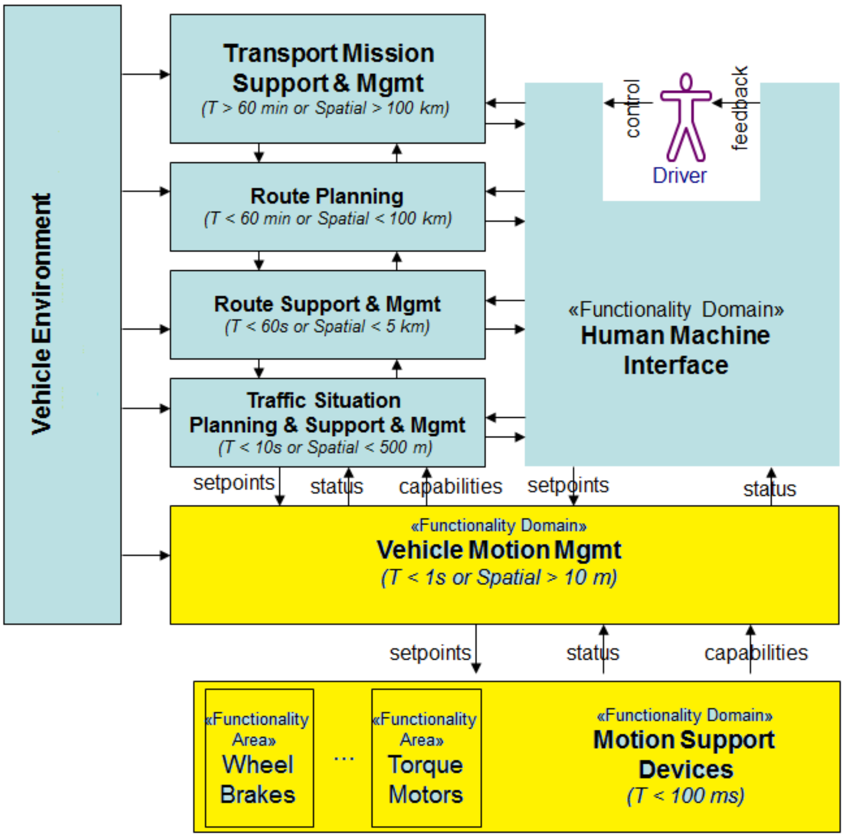


FIGURE 2.1: Visual representation of the different functionality levels of the CVC reference architecture. The functionality domains, in which this thesis focuses on, are highlighted with yellow colour.

Motion Support Devices

This is the lowest level that interacts directly with the vehicle. The functionalities included in this level are related to the vehicle's motion. A major characteristic of these devices is that they cannot handle the vehicle's motion but rather contribute and facilitate higher levels of CVC to control vehicles' s motion. This domain can include devices that consist of only actuators, only sensors or both along with some software. Its functionalities are to organize monitoring, measurement and control different devices of the vehicle. In order to be more clear of what is included in this domain an example will follow.

One device that can be included in this domain is the "Steering Management". This can consist only from the steering actuators, which receive a set point from a higher level and just actuate to the wheels. Another device can be the "Body Management", which consists only from sensors that report to higher levels the longitudinal, lateral velocities or the angular velocity of the vehicle. An example where sensors, actuators and software are working together can be the "Motors Management". In this device, the requested torque is sent from the higher levels. A software can evaluate if the wheels will slip when this torque will be applied and maybe decide to reduce the value. Then the actuators apply the given command, some sensors can measure the wheel angular velocity and the software can report back if the command was fulfilled and what are the current capabilities of the motors.

2.2 EAST-ADL Concept

The complexity of the electrical and electronic systems in automotive industry increases continuously. This trend leads to the need of developing a tool that will make the documentation, presentation, designing and analysis of the embedded systems of a car easier. EAST-ADL is an architecture description language used for automotive embedded systems and it was developed in different European projects, where automotive industries like Volvo participated. EAST-ADL provides a framework for describing features, functions, requirements, SW and HW components and communication for automotive industry in a standardized form. It contains abstraction levels, which are the core system model, depicted in the figure 2.2. Each level is describing the system from a different view and with different amount of details.

Vehicle level: Contains the Technical Feature Model element, which represents the features of a vehicle from a top-level perspective without giving any information about realization. This level states "what" the vehicle should do.

Analysis level: Contains the Functional Analysis Architecture (FAA) element, which presents a complete representation of the functionalities in an abstract form. The FAA captures the basic interfaces and behaviour of the different subsystems. Critical issues for understanding and analysing can be considered without the implementation details. This and the lower levels state "how" the vehicle does what is being stated in the Vehicle level.

Design Level: Contains two main elements. The Functional Design Architecture , where the implementation aspects are introduced. The Hardware Design Architecture , which constraints the development, thus they are implemented at the same time. There is also an effect on higher levels. Control strategies or the entire functionality may have to be revised to be implemented on a realistic hardware architecture.

Implementation Level: The implementation of the represented embedded system using AUTOSAR[1] elements.

Along with the core system model, an environmental model should exist to assist in verification and validation of the vehicle's features. This model captures the behaviour of elements that interact with the core model. Vehicle dynamics, mechanical or hydraulical systems of the vehicle, road surface and traffic environment are included in this model. Furthemore, EAST-ADL is enriched with extensions that allow the definition of models focusing in different spects. For example a model can be analysed based on time modelling, requirements modelling, functional safety and variability modelling [7]. In figure A.1 of the Appendix A, a notation guideline is presented.

[1]AUTOSAR is an open and standardized software architecture used in automotive industry [6].

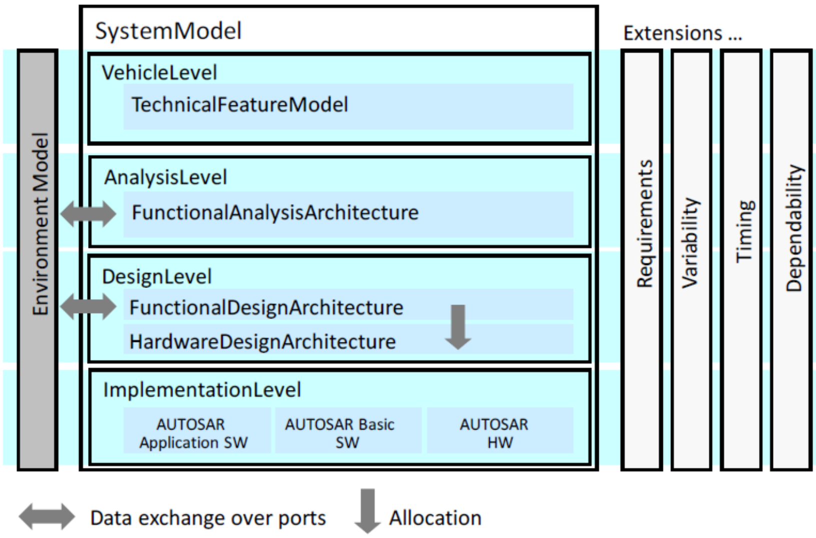


FIGURE 2.2: The EAST-ADL' s breakdown in abstraction levels (vertically) and in core system model, environment and extensions (horizontally). Originally illustrated in [7]

Chapter 3

Vehicle Configuration and Modelling

This chapter presents the vehicle's configuration and provides theoretical information about the modelling of different parts of the EDV, based on a dynamic approach.

3.1 Background on HEVs and their propulsion system

From the beginning of the EDV project, it was decided that the constructed vehicle should be a hybrid electric small car. There are three main configuration types of HEVs.

Parallel hybrid: Both ICE and electric motor operate on the same drive shaft, so they can power the vehicle individually or simultaneously.

Series hybrid: The electric motor only drives the vehicle. The electricity can be supplied either from battery or by a generator.

Combined hybrid: This configuration, as its name shows, has both mechanical link as parallel and an electrical link as series hybrids.

The propulsion system of a series HEV utilizes the ICE as an auxiliary power unit to extend the driving range of the vehicle. Using a generator, the ICE's output is converted to electricity which can be used either for recharging the battery or for supply the electric motor. An advantage of series hybrids is that regenerative braking can be used. Regenerative braking is when the electric motor is used in order to decelerate the vehicle. Then the motor acts as a generator and the produced electricity can be stored to the battery. Another advantage is that there is not need for a clutch for the transmission [8]. The basic series HEV configuration can be seen in the figure 3.1.

At the beginning of the EDV project, the desired configuration of the car was formulated. Based on this the car should have the following attributes:

- The propulsion system of the car should be a type of series hybrid.

- There should be one motor for each rear wheel (2- inputs).
- The motors should be brushless DC motors (BLDCM).
- Each wheel should be equipped with brake (4- inputs).
- The two front axle wheels should have steering capability.

3.2 System Modelling

Modelling a complete HEV is split into a number of different sub-models. Thus, it is useful to have these sub-models autonomous in order to model a modular system description with clear inputs and outputs. There are two different modelling approaches, quasistatic and dynamic [8]. The modelling approach method that will be used in this thesis is the dynamic modelling, which is based on mathematical equations [8]. The flows of the physical exchanged data are shown in figure 3.1. Furthermore, the modelling of a vehicle is divided into two different parts, the chassis dynamics and the actuators dynamics. This thesis focuses in the control design and not in the modelling. Hence, only some basic modelling, which was needed for the control design is presented in this section.

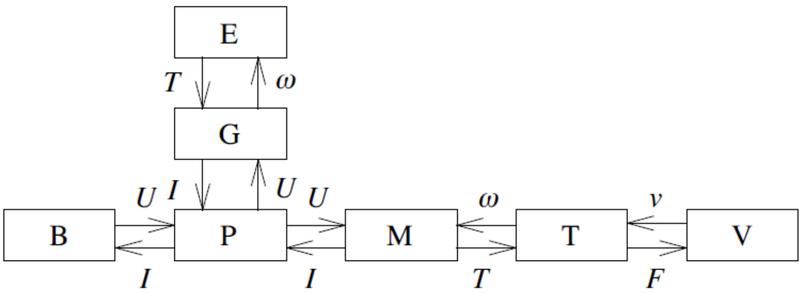


FIGURE 3.1: Basic series hybrid configuration. **B**: battery, **E**: engine, **G**: generator, **M**: motor, **P**: power converter, **T**: transmission, **V**: axles and vehicle. Flow of the physical exchanged data for a series hybrid configuration with dynamic approach. **F**: force, **I**: current, **P**: power, **T**: torque, $\boldsymbol{v}$: speed, $\boldsymbol{U}$: voltage, $\boldsymbol{\omega}$: rotational speed. Illustrated at [8].

3.2.1 Chassis Modelling

For the definition of the axis orientation, the SAE standard [9] provided the main guidance. The chassis will be seen as a system of 3 DOF; longitudinal, lateral and yaw (around $z - axis$) motion. The reference frame for the system will be at the CoG of the chassis. Additionally in figure 3.2, a top view of the vehicle model is depicted, where the axis orientation can be observed. The explanation of the different variables is given in the text or in the figures, while the values of the parameters are presented in Appendix B. Based on Newton's motion laws, the equations of the acting forces for the longitudinal

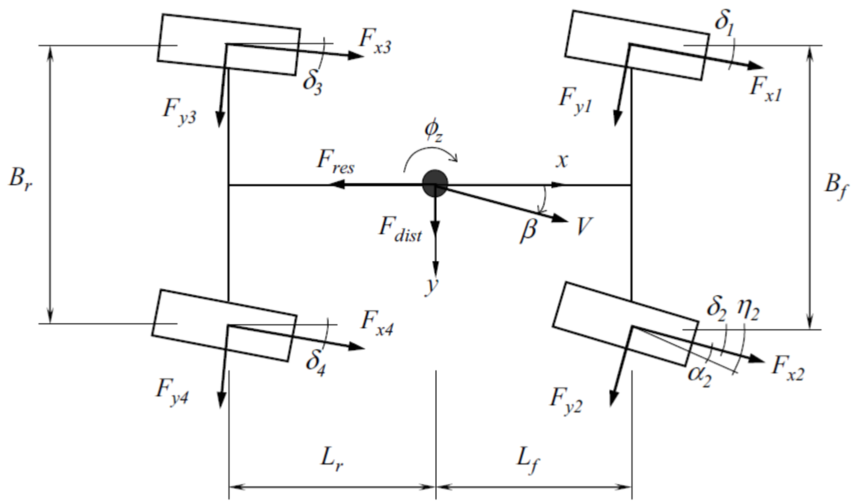


FIGURE 3.2: Chassis model, $x - y$ view, from top. Illustration from [10].

(x) and the lateral (y) direction are presented below:

$$ma_x = \sum_{i=1}^{4} F_{x,i}\cos(\delta_i) - \sum_{i=1}^{4} F_{y,i}\sin(\delta_i) - F_{res} \tag{3.1}$$

$$ma_y = \sum_{i=1}^{4} F_{x,i}\sin(\delta_i) - \sum_{i=1}^{4} F_{y,i}\cos(\delta_i) \tag{3.2}$$

The accelerations are approximated as $a_x \approx \ddot{x} - \dot{y}\dot{\phi}_z$ and $a_y \approx \ddot{y} - \dot{x}\dot{\phi}_z$. The F_{res} is the sum of the forces which act in the opposite direction from the vehicle's movement [8].

$$F_{res} = F_{drag} + F_{roll} + F_{grav} \tag{3.3}$$

where F_{drag} is the aerodynamical drag force, F_{roll} is the roll resistance and the F_{grav} is the loss because of gravity when the road has a slope and vehicle is moving uphill. It was assumed that the resistance forces act on the CoG of the vehicle body and some further analysis for these forces is presented in appendix B. Moreover, for simplification reasons, no disturbances were considered.

The sum of moments around the CoG for the yaw-direction is described by the following equation:

$$
\begin{aligned}
I_{zz}\ddot{\phi}_z = L_f &\left(\sum_{i=1}^{2} F_{y,i}\cos(\delta_i) + \sum_{i=1}^{2} F_{x,i}\sin(\delta_i)\right) \\
- L_r &\left(\sum_{i=3}^{4} F_{y,i}\cos(\delta_i) + \sum_{i=3}^{4} F_{x,i}\sin(\delta_i)\right) \\
+ \frac{b_f}{2} &\left(\sum_{i=1}^{2}(-1)^{1+i} F_{x,i}\cos(\delta_i) + \sum_{i=1}^{2}(-1)^{i} F_{y,i}\sin(\delta_i)\right) \\
+ \frac{b_r}{2} &\left(\sum_{i=3}^{4}(-1)^{1+i} F_{x,i}\cos(\delta_i) + \sum_{i=3}^{4}(-1)^{i} F_{y,i}\sin(\delta_i)\right).
\end{aligned}
\tag{3.4}
$$

3.2.2 Brushless DC Motor

A simple model of a DC motor consists of two parts, one describing the electrical and the other the mechanical part of the motor. For the electrical part, dynamic equations are derived based on Kirchhoff' s voltage law, while for the mechanical part, dynamic equations are derived based on Newton's second law. A brushless DC motor is a synchronous machine, which approximates the behaviour of a brush-type DC motor. In order to stimulate the permanent magnets of the rotor, a three phase AC voltage is applied on the stator's armature windings. Their difference is that power electronics are taking the place of the brushes [8]. A BLDC motor has some advantages, when it is used as a motion actuator in a vehicle, compared to ICE or AC motor and these are listed below:

1. A DC motor is easier controllable than an AC motor, by just controlling the current provided to the electrical part of the motor.
2. There is lower need of using a gearbox when using a DC motor. This is because the maximum torque of a DC motor is available when motor speed is zero and remains available for a wider range compared to what happens to ICE.

The available torque of the motor depends on the rotational speed of the motor. After a critical rotational speed ω_c, the torque starts declining. The equation describing the behaviour of the torque due to rotational speed is given by the following equation:

$$
T_{avail} = \begin{cases} T_{max} & \forall\omega \in 0 \le \omega \le \omega_c, \\ \\ T_0 - K_t\omega & \forall\omega \in \omega_c < \omega \le \omega_{max}. \end{cases}
\tag{3.5}
$$

Because of the advantages mentioned earlier, for the EDV project, the use of BLDC motors was decided. The basic motor candidate was the HPM5000B72V from Golden Motor. Although for this thesis, no motor model was created, the necessary values needed in the control system were calculated based on the information and data values at the company's web site [11]. The characteristic curve showing the relation between torque and rotational speed was derived and is depicted in figure 3.3. An approximated value was assigned to T_0, after linearising the characteristic curve for $\omega > \omega_c$. All the values for the motor's parameters are presented in the table B.3.

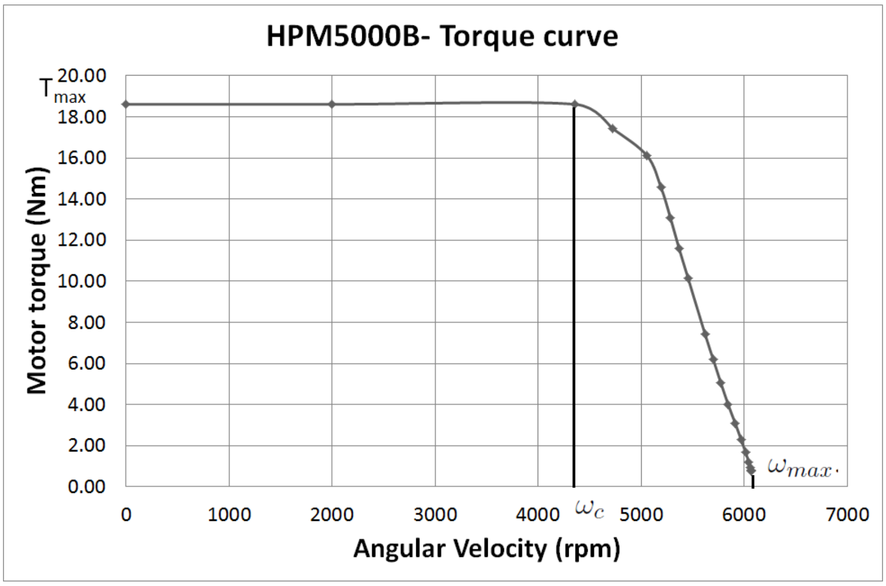


FIGURE 3.3: Torque curve for motor model HPM500B72V6000rpm of Golden Motor.

3.2.3 Electro-Hydraulic Disc Brake System

The braking system is used in order to convert the kinetic energy of a vehicle into thermal energy. Ideally the kinetic energy is completely absorbed by the braking system during a maximum deceleration event, when going from any speed to zero speed [12]. The follow equation shows this basic conversion:

$$E_{kinetic} \Rightarrow E_{thermal} \Leftrightarrow \frac{1}{2}m\dot{x}^2 \Rightarrow m_b C_p \Delta T_b \tag{3.6}$$

where m_b is the mass of the braking system components, C_p a constant based on material properties and ΔT_b is the temperature rise of the braking system. In a simple model of electro-hydraulic braking system, the signal created by the driver in the *brake pedal* is multiplied and transmitted to the *master cylinder*. There, the force is translated to hydraulic pressure and then transferred to the *caliper*. It is the responsibility of caliper to translate the hydraulic pressure into mechanical force. The force at the caliper is given by the equation :

$$F_{cal} = 2P_{cal}A_{cal} \tag{3.7}$$

where P_{cal} is the hydraulic pressure transmitted to the caliper and A_{cal} the effective area of the caliper pistons. Then, this force is applied to the rotor through the *braking pads* with just multiplying that force by a brake pads friction coefficient.

In the present thesis, a brake model was not implemented. Nevertheless, it was taken in consideration that the braking system will be electro-hydraulic. Based on the equations presented above an approximated but realistic braking force value for a small car was used. The value for the rate of change was chosen arbitrary and was chosen to be high since the applied braking force can change fast. The assigned values are presented in the table B.3.

3.2.4 Vehicle Model

As mentioned earlier, it was not in the thesis' responsibilities to develop a vehicle model. For this reason, when a model was needed during the simulation tests, one was provided by Volvo and it was a mechanical model of a vehicle. The vehicle body consisted of 6 DOF for describing longitudinal, lateral, vertical, roll, pitch and yaw motion. Furthermore each one of the wheels had 3 DOF , for describing vertical, pitch and yaw motion, leading to a model of 18 DOF in total. The vehicle model was implemented utilizing SimMechanics because of the following benefits of the program [13]:

- It handles 3D mechanical modelling.
- It is modular.
- It has bi-derectional connections.
- It is completed integrated with Simulink.

In order for the vehicle model to be compatible with this project, a modification to the model's parameters was done, therefore they were adjusted to the size of the EDV.

Chapter 4

Theory for Control Design

This chapter presents the design of the control system. Theoretical parts about the control methods that will be implemented are explained in order to prove the correctness of the choices. At the end of this chapter, the reader should have understood the "why" specific choices were made and the "how" the control system was formulated.

4.1 Introduction to Control Allocation

The main DOF that are controlled in a road vehicle are the longitudinal, lateral and yaw motions and they are generated by different types of motion actuators. Previous decades these actuators were controlled directly from the driver through mechanical links. Today, these actuators are more and more controlled as embedded software. This leads to an over actuated system where there are more controlled actuators than the three main motions of a vehicle. For such systems, control allocation can be implemented. This method is attractive because it can be implemented not only in linear systems, but also in non-linear. Furthermore, control allocation can handle actuators constraints and saturations. When using control allocation, the control design is separated in two different steps. At the first step a control law is implemented for regulating the total control effort, which is also called virtual inputs, $v(t) \in \mathbb{R}^k$. At the second step, control allocator maps the virtual inputs v to set points for the different actuators, $u(t) \in \mathbb{R}^m$, where $m > k$ [14]. In figure 4.1 the general overview of the control system is presented.

The general affine form for a non-linear system is the following:

$$\begin{aligned} \dot{x} &= f(x) + g(x)u \\ y &= h(x) \end{aligned} \tag{4.1}$$

If $g(x)$ can be expressed linearly, then the mapping of $g(x)u \mapsto v$ can be expressed as

$$Bu(t) = v(t) \tag{4.2}$$

substituting $g(x)$ by a constant matrix B multiplied by the real control inputs, u. However this approximation is not always true. For this reason the control allocation problem ends up in a constraint minimization problem as it will be shown further on. Matrix B is called *control effectiveness matrix*, it is a $k \times m$ matrix with rank k and its defination

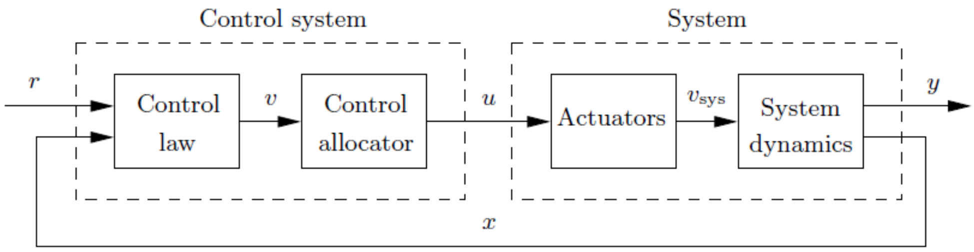


FIGURE 4.1: Control system structure when control allocation is used. The control system is made up by a control law, specifying which total control effect v should be produced and a cotrol allocator, which distributes this control demand among the individual actuators, u. In the system, the actuators generate a total control effect. v_{sys}, which determines the system behaviour. If the control allocation is successful then $u_{sys} = v$. Originally illustrated in [14].

is given in equation (4.20). In order to incorporate the limitations of the actuators, the actuators' position constraints are expressed as:

$$u_{min} \leq u(t) \leq u_{max} \tag{4.3}$$

for every actuator separately. If actuators rate constraints exist, they are expressed as:

$$\rho_{min} \leq \dot{u}(t) \leq \rho_{max} \tag{4.4}$$

Because control allocator is part of digital control, an approximation of the derivative is made:

$$\dot{u}(t) \approx \frac{u(t) - u(t - T_s)}{T_s} \tag{4.5}$$

where T_s is the sampling time. In order to obtain the overall position constraints, the equations (4.3) - (4.5) are combined giving:

$$\underline{u}(t) \leq u(t) \leq \overline{u}(t) \tag{4.6}$$

where

$$\begin{aligned} \underline{u}(t) &= max\{u_{min}, u(t - T_s) + T_s\rho_{min}\} \\ \overline{u}(t) &= min\{u_{max}, u(t - T_s) + T_s\rho_{max}\} \end{aligned} \tag{4.7}$$

Equation (4.2) constrained by (4.6) forms the standard linear control allocation problem. In Härkegård's thesis [14] different methods for solving the control allocation problem are discussed. The solution of the optimal control input can be separated to two steps optimization problem as:

$$\begin{aligned} u &= \arg\min_{u \in \Omega} \|W_u(u - u_d)\|_p \\ \Omega &= \arg\min_{\underline{u} \leq u \leq \overline{u}} \|W_v(Bu - v)\|_p \end{aligned} \tag{4.8}$$

where u_d is the desired control input, W_u and W_v are weighting matrices and Ω is the set of feasible control inputs. Additionally p is the type of norm. l_2 norm is preferable because it distributes the virtual control demand to all the available control inputs, while l_1 only to a few. Moreover l_2 solution varies continuously with the solution parameters [14].

One suggested branch of solutions is the *active set methods*, which are the most efficient methods when a good estimation of the active set is available. In [14], two active set methods are proposed. One is for handling the control allocation problem as a *weighted least squares* optimization problem, reformulating equation (4.8) into the following:

$$u = \arg \min_{\underline{u} \leq u \leq \overline{u}} \|W_u(u - u_d)\|_2^2 + \gamma \|W_v(Bu - v)\|_2^2 \tag{4.9}$$

where the matrices W_u and W_v are designing parameters and $\gamma \gg 1$ to emphasize that the term $Bu - v$ should be primarily minimized.

4.2 State Eqations for the Vehicle System

In order to represent the vehicle system as a non-linear system, the state variables are chosen as $x = \begin{bmatrix} V_x & V_y & \omega_z \end{bmatrix}^T$.

The lateral slip, a_i for each wheel, is separated into one part due to steering angle and into a second part due to vehicle states as in [10]:

$$\begin{aligned} a_1 &= \delta_1 - \frac{x_2 + L_f x_3}{x_1 + \frac{b_f}{2} x_3} \\ a_2 &= \delta_2 - \frac{x_2 + L_f x_3}{x_1 - \frac{b_f}{2} x_3} \\ a_3 &= \delta_3 - \frac{x_2 - L_r x_3}{x_1 + \frac{b_r}{2} x_3} \\ a_4 &= \delta_4 - \frac{x_2 - L_r x_3}{x_1 - \frac{b_f}{2} x_3} \end{aligned} \tag{4.10}$$

For small angles the lateral force can be calculated as

$$F_{y,i} = C_{a,i} a_i \tag{4.11}$$

where $C_{a,i}$ is the conrnering stiffness for each one of the tyres. If equations (4.10) are combined with equation (4.11), the lateral force can be written as the sum of the force during steering angle, $F_{y,i}(\delta)$ and vehicle states, $F_{y,i}(x)$

$$F_{y,i} = F_{y,i}(\delta) + F_{y,i}(x) \tag{4.12}$$

It is assumed that all the wheels are the same and the track width for front and rear axles is equal with $b_f = b_r = b_t$, as well as the lengths $L_f = L_r = L/2$. Using the equations (4.10) - (4.12), the equations (3.1) -(3.4) can be expressed as a non-linear system with the below equations [10]:

$$f(x) = \begin{bmatrix} mx_2x_3 - D_1 sign(x_1)x_1^2 - D_2 sign(x_1) \\ -mx_1x_3 - C_a \dfrac{16x_1x_2}{4x_1^2 - (b_t x_3)^2} \\ -L^2 C_a \dfrac{16x_1x_3}{16x_1^2 - 4b_t x_3^2} \end{bmatrix} \tag{4.13}$$

$$g(x)u = \begin{bmatrix} \sum_{i=1}^{4} F_{x,i} \\ C_a \sum_{i=1}^{4} \delta_i \\ \dfrac{L}{2} C_a \sum_{i=1}^{2} \delta_i - \dfrac{L}{2} C_a \sum_{i=3}^{4} \delta_i + \dfrac{b_t}{2} \sum_{i=1}^{4} (-1)^{1+i} F_{x,i} \end{bmatrix} \tag{4.14}$$

$$h(x) = \begin{bmatrix} x_1 & x_2 & x_3 \end{bmatrix}^T \tag{4.15}$$

where D_1 is a constant parameter for aerodynamic force and D_2 a constant parameter for rolling resistance based on Appendix B.2. The system can be rewritten as:

$$\begin{aligned} M\dot{x} &= f(x) + g(x)u \\ y &= h(x) \end{aligned} \tag{4.16}$$

where, M is the mass matrix

$$M = \begin{bmatrix} m & 0 & 0 \\ 0 & m & 0 \\ 0 & 0 & I_z \end{bmatrix}$$

Matrix M is invertible and thus the system can be written in the affine form of a non-linear system. From the equations (4.13) - (4.15) can be observed that $f(x)$ is the non-linear, while the inputs of the system are the forces created by the motion actuators and the steering angle of the wheels. Furthermore, the virtual inputs are selected as $v = \begin{bmatrix} F_x & F_y & M_z \end{bmatrix}^T$

4.3 Designed Control System

4.3.1 Designed Control Law and Feedback Linearisation

As mentioned in section 4.1, the responsibility of the control law was to regulate the virtual inputs. The chosen regulator was a PI-controller. This type of controller was selected because it was simple enough, but still sufficient as it has been shown in similar implementations as [10, 13]. Moreover, an anti-windup method was added to the controller. All actuators have some limitations and when an actuator's limit is reached, then the system runs as an open loop because the actuator remains at its limit independently of the process. In such a circumstance, if the controller has an integrating part, the value of this term continues to increase. There are different methods in order to avoid this windup phenomenon as described in [15]. The one chosen in this thesis is called back-calculation. In this method, when the output saturates, the integral is recomputed so that its new value gives an output at the saturation limit. The reset of integrator part is done dynamically with a time constant T_t. With the anti-windup mechanism, the controller has one more feedback that calculates the error, e_s, between the controller output and the actuator output. For the implementation of this project, the error e_s was defined as the difference between the outputs of the control law and the control allocator, as it can be seen in figure 4.2.

Along with the PI controller, a feedback linearisation was added to the control law. With the feedback linearisation, the non-linear term $f(x)$ is subtracted in order to cancel the non-linearities, transforming the non-linear system into a linear. The designed control law is described by the equation:

$$v = -a(x) + K_p e + \int_0^{T_s} K_i e + \frac{1}{T_t} e_s \, \mathrm{d}t \tag{4.17}$$

where $a(x)$ is the cancellation of the non-linearities and T_s the sampling time. Figure 4.2 shows the scheme of the control law.

The designed parameters for the controller were chosen similar to [10] as:

$$\begin{aligned} K_p &= 4m\sqrt{K_i/m} \\ K_i &= 5\begin{bmatrix} m & 0 & 0 \\ 0 & 0.6m & 0 \\ 0 & 0 & 1.5I_z \end{bmatrix} \\ T_t &= \begin{bmatrix} 0.2 & 0 & 0 \\ 0 & 0.2 & 0 \\ 0 & 0 & 0.2 \end{bmatrix} \end{aligned} \tag{4.18}$$

4.3.2 Designed Control Allocator

After designing the control law, the second step, in order to complete the control system, was the design of the control allocator. The described solution in section 4.1 was followed. The vector of the real control inputs consists of the available motion actuators:

$$u = \begin{bmatrix} T_{wm3} & T_{wm4} & T_{wb1} & T_{wb2} & T_{wb3} & T_{wb4} \end{bmatrix}^T \tag{4.19}$$

where T_{wmi} the torque generated by the motors to wheels 3-4 and T_{wbi} the torque generated by the brakes to wheels 1-4.

The dimensions of the control effectiveness matrix will be $k \times m = 3 \times 6$ and has to be constant. For that reason, it is assumed that there are no losses and time delays or non-linearities in the developing tyre forces. In this way the control effectiveness matrix

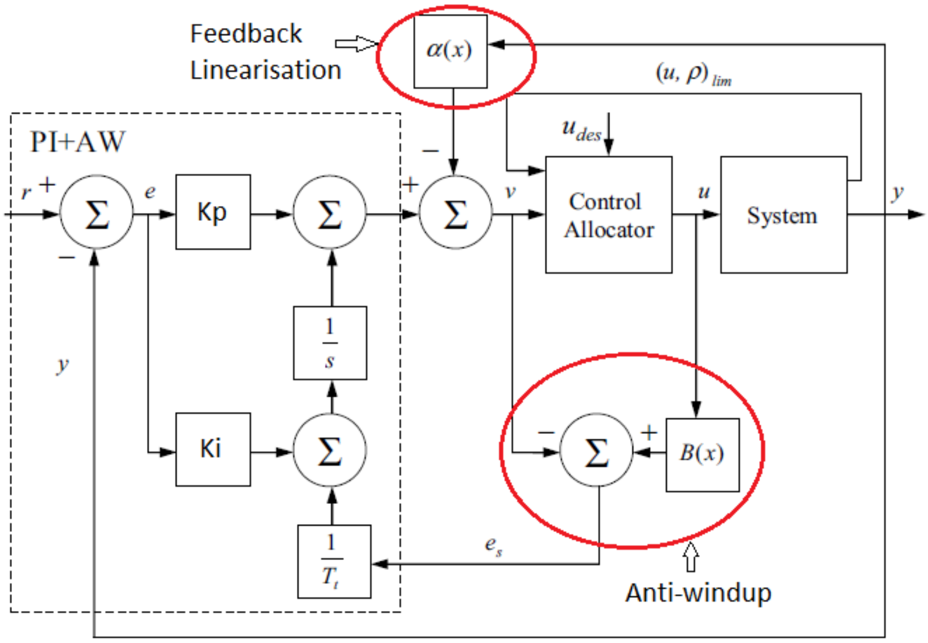


FIGURE 4.2: Layout of the designed Control Law, including the PI- controller, anti-windup and feedback linearisation. The term $a(x)$ is a matrix similar to theoretical matrix $f(x)$, but it includes the measured longitudinal, lateral and yaw speeds.

is formulated as :

$$B = \begin{bmatrix} \frac{r_{fg}}{R_w} & \frac{r_{fg}}{R_w} & \frac{1}{R_w} & \frac{1}{R_w} & \frac{1}{R_w} & \frac{1}{R_w} \\ 0 & 0 & 0 & 0 & 0 & 0 \\ \frac{r_{fg}b_t}{2R_w} & \frac{-r_{fg}b_t}{2R_w} & \frac{b_t}{2R_w} & \frac{-b_t}{2R_w} & \frac{b_t}{2R_w} & \frac{-b_t}{2R_w} \end{bmatrix} \tag{4.20}$$

The real control inputs consist only from the motion actuators. It can be observed that, if no steering actuators are included in the controller, the lateral force F_y cannot be influenced. The designed parameters for the solution of control allocation problem, based on equation (4.9), are chosen similar to [10] as:

$$\begin{aligned} W_v &= diag \begin{bmatrix} 1 & 1 & 1 \end{bmatrix} \\ W_u &= diag \begin{bmatrix} 1 & 1 & 0.25 & 0.25 & 0.25 & 0.25 \end{bmatrix} \\ u_d &= \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}^T \\ \gamma &= 1000 \end{aligned} \tag{4.21}$$

The above selection is based on the desired behaviour of the controller. As this is the first version, it is preferred that all the virtual inputs have the same importance. As far as the real inputs are concerned, since regenerative braking is a requirement to be fulfilled in the future, the use of the motors as actuators is prioritized over the brakes. Moreover, the desired real inputs' matrix, u_d, is set to zero, because no investigation was made for potentially preferable values. Since $u_d = 0$, parameter γ is selected very high in order to prioritize the second term of the optimization problem.

4.4 Interaction between Driver and Vehicle

Until now, the control design for the control allocation is explained, but nothing is mention about how the driver interacts with the vehicle. The three main inputs from the driver to the vehicle are the *gas pedal* (GP), the *braking pedal* (BP), which can be seen as position sensors and the *steering wheel angle* (SWA). Besides these three inputs, for this project, one more input is included, called *driving mode* (DM). It is common logic, that the desired trajectory for the vehicle is derived based on the signals from the driver. Hence, apart from the control allocation, another control system should exist and being responsible for creating the desired trajectory, $r = \begin{bmatrix} V_x & V_y & \omega_z \end{bmatrix}$. This control system is named as *Driver Interpreter*.The overall control system design is depicted in figure 4.3.

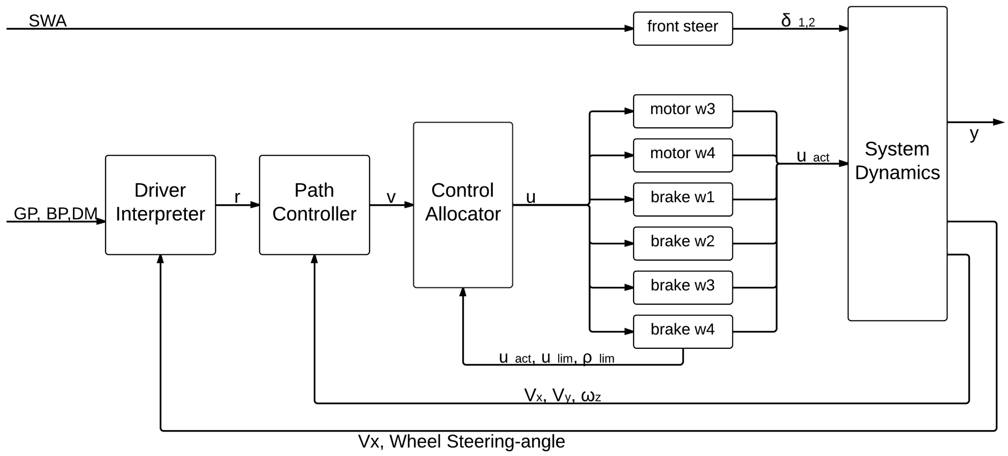


FIGURE 4.3: Overall control system's block diagram. *Driver Interpreter* is responsible for creating the desired trajectory $r = \begin{bmatrix} V_{xdes} & V_{ydes} & \omega_{zdes} \end{bmatrix}^T$. *Path Controller* is the PI controller for regulating the virtual inputs, $v = \begin{bmatrix} F_x & F_y & M_z \end{bmatrix}^T$. Then, *control allocator* follows, where the virtual inputs are mapped to the real inputs $u = \begin{bmatrix} T_{wm3} & T_{wm4} & T_{wb1} & T_{wb2} & T_{wb3} & T_{wb4} \end{bmatrix}^T$. The SWA is not included in the control system, nevertheless the driver can influence the vehicle's dynamics by steering.

4.4.1 Gas Pedal Mapping

Before the use of embedded software in vehicles, the GP was mechanically controlling the fuel injection to the ICE. Nowadays, the GP is software-linked to the engine. In order to link the GP with the engine, its different positions have to be mapped to a corresponding acceleration or torque request. Based on the article [16], the most common model is to map the GP positions to some torque request as a percentage from the total available torque, $GP_{posit} \mapsto T_{request}$. The available torque is considered to be constant and is the maximum value of the motor's capacity, T_{max}. The one-dimension map should obey the following principles [16]:

- When the driver is not pressing the GP, the torque request should equate the torque needed to keep the engine idle.
- When the driver is pressing the GP at its full position, the torque request should equate to the maximum torque.
- The rest of GP's positions should be mapped to some torque request between 0 and 100 percent of maximum torque.

4.4.2 Driving Modes

One of the requirements of the EDV project, is that the driver should be able to choose different driving modes. This can be integrated through different mappings of GP. Three driving modes are implemented. The choice of the different modes is based on the project's requirements, while the mapping for each mode is based on the previously stated principles:

Normal mode: The mapping between the pedal position and the requested torque is linear.

Sport mode: The mapping between the pedal position and the requested torque is more aggressive" than the normal mode. Higher levels of torque are demanded during early pedal positions. This mode is designed to give to the driver the sense that the vehicle is racier.

Eco mode: The mapping is designed so that less torque than the normal mode is demanded in earlier pedal positions. This mode is designed for a more comfortable and energy saving driving behaviour.

4.4.3 Formulating the reference trajectory

The responsiblity of *Driver's interpreter* is to create the reference trajectory, as mentioned earlier. In order to estimate a desired V_{xdes}, the driver's requests for accelerating or braking should taken into consideration. For the present thesis, only the GP is taken in consideration and the desired speed is calculated by the following equation:

$$V_{xdes} = T_{request} * \int_0^{T_s} a_{max}\,\mathrm{d}t \tag{4.22}$$

where a_{max} is the maximum value of a_x of equation (3.1).

In order to calculate the desired yaw speed, ω_{zdes}, the formula below, based on [13], is used:

$$\omega_{zdes} = \frac{V_{xmes}\delta}{(L_f + L_r) + Kus\dfrac{V_x^2}{g}} \tag{4.23}$$

where V_{xmes} is the measured longitudinal speed, $L_f = L_r$ the distance from the front and rear axles to the CoG of the chassis in the x-axis, δ is the wheel's steering angle and Kus is the steering coefficient.

In vehicle dynamics, Kus coefficient determines whether a vehicle understeers or oversteers. It is a measurement of how the current steering differs from a steady steering and depending on its value, a given curve has to negotiate by increasing or decreasing the desired yaw rate [13]. The equation for calculating Kus coefficient is:

$$Kus = \frac{W_f}{C_f} - \frac{W_r}{C_r} \tag{4.24}$$

where, C_f and C_r is the cornering stiffness of front or rear tyres, while W_f and W_r is the load on front or rear tyres. There are three different occasions where:

- When $Kus = 0$, vehicle has a neutral steer.
- When $Kus > 0$, vehicle is understeered.
- When $Kus < 0$, vehicle is oversteered.

During the current thesis, the ω_{zdes} is calculated always in the neutral steer occasion. This happens because the cornering stiffness of rear tyres is considered the same as the front tyres and because the load is considered to be equally distributed in all the tyres. A useful notice is that although the steering is not included in the control system and is mechanical, the desired rotational speed is influenced by the wheels' steering angle.

The last desired motion of the trajectory is the lateral speed V_y. Since the control system does not handle the steering actuators, no matter the value of the desired lateral speed, V_{ydes}, the control's system results cannot be influenced.

Chapter 5

Implementation of the Control System Architecture

In this chapter, the steps followed for the realisation of the project are mentioned. Furthermore, the designed EAST- ADL levels are presented.

5.1 Thesis' Procedure

The steps followed for the completion of this project can be separated into three different periods. The first was the initialization period. During that time, the requirements for this project were given and documented in Enterprise Architect (EA) as a draft scheme. Most of the time spent during this period was for literature search and study about the theories of Reference Architecture, EAST- ADL, vehicle dynamics and control for overactuated systems were made during these early weeks. Furthermore, the Vehicle level from the core model of EAST- ADL was created.

The control implementation was done in the next period. During that phase, the Functional Analysis Architecture for the Analysis level was made as a draft in EAST- ADL . The control system's functionalities, based on the EAST- ADL draft, were implemented in a Simulink model. During that period, the vehicle model was integrated as a SimMechanics model. The models created in Simulink software were used for testing and validating that the designed control system had the desired behaviour. That was the longest and most important period of the thesis work.

The third and last period included the revision and finalization of the EAST- ADL model. The final scheme of the FAA was completed by updating the draft with the changes that had been made in the Simulink model. The documentation of the requirements for this project was also made at the end of the thesis work.

5.2 Structure Definition and Documentation with EAST-ADL

For designing the EAST- ADL model in EA, a pattern designed from Volvo was given. This pattern had classes already defined and ready to use. For the Vehicle level, a Technical Feature Model was sketched. Based on the theoretical framework of EAST-ADL , the features of a vehicle were documented in this model, without showing any deeper details of realisation. Nine basic group features were identified. Each one of them included numerous other features that can be implemented separately. Since this thesis was done at the beginning of the EDV project, only the basic Propulsion feature was taken in consideration. A property called "cardinality" was assigned to each feature class, indicating the necessity of the feature for this project. A "cardinality" value equals to one was representing a feature that was necesary for the EDV project. On the contrary, a "cardinality" value equals to zero was representing a feature that was optional for the project. The scheme of the Technical Feature Model can be seen in figure 5.1.

The other level that was completed during this thesis work was the Analysis level, with the FAA element. As mentioned earlier, two FAA schemes were made, one as a draft in the beginning and the final one. The final FAA had to be the same as the Simulink model, but representing only the most important signal flows. The CVC architecture was followed for the definition of the different functionality domains and more details about the creation of the different systems and subsystems is given in chapter 6. In figure 5.2, the final FAA is depicted.

During the present thesis, no further design was made for lower levels of EAST- ADL, but the requirements extension was made. A diagram was designed, associating each requirement with those functionality blocks from the FAA that fulfilled the specific requirement. Figure 5.3 shows the completed diagram.

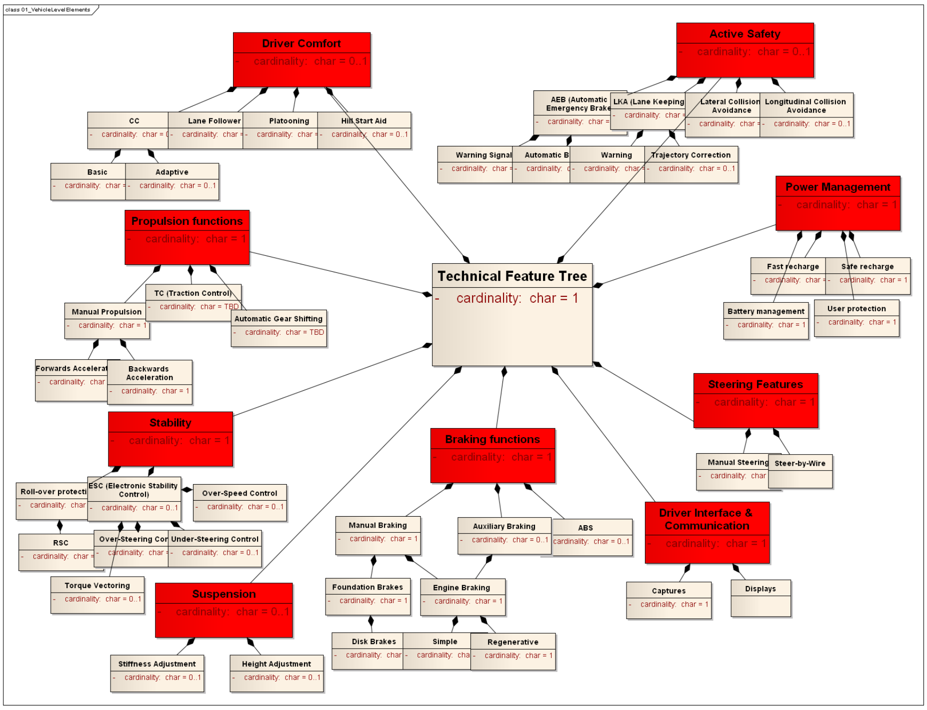


FIGURE 5.1: Technical Feature Model for the Vehicle Level of EAST- ADL. The nine basic group features are shown with red colour.

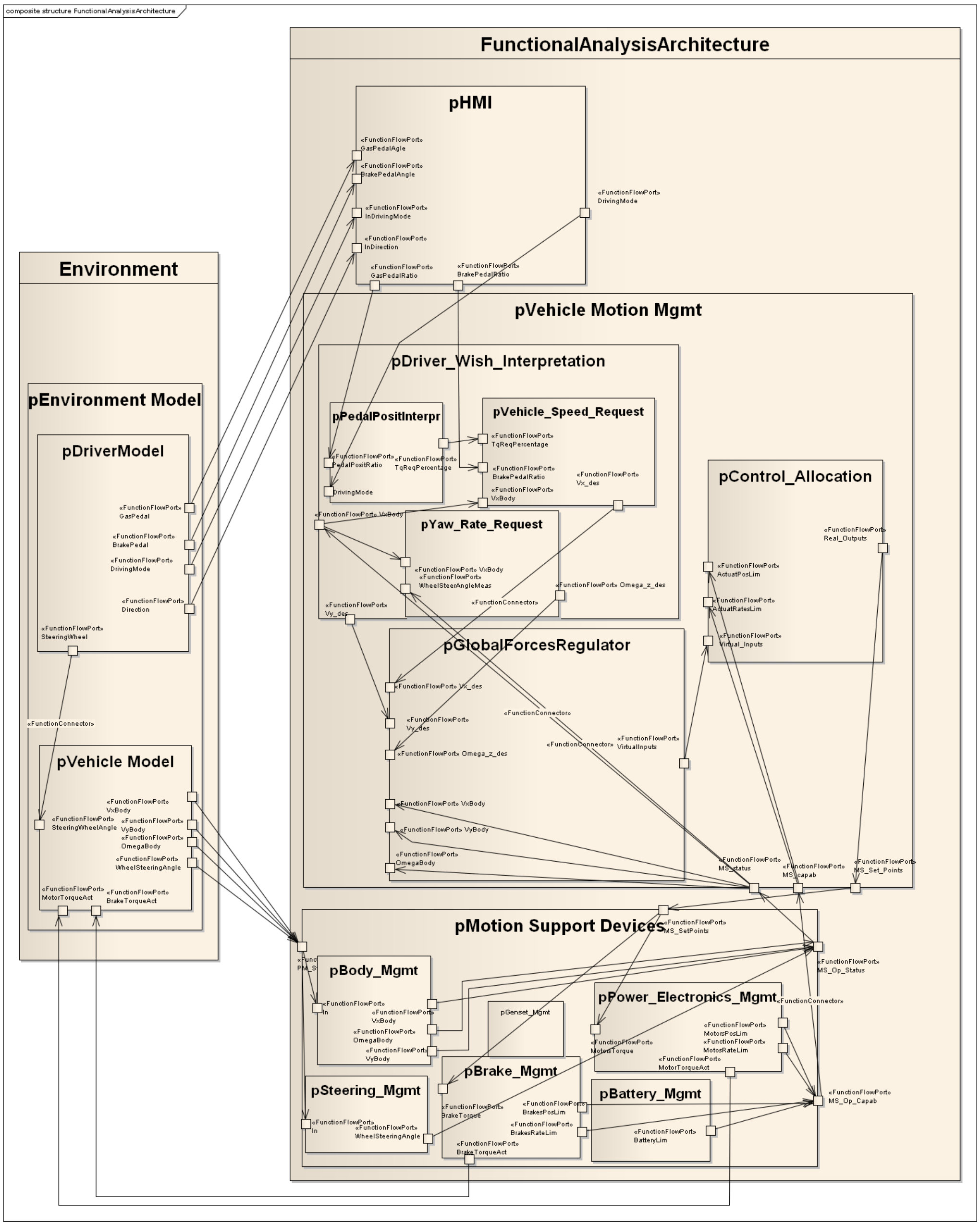


FIGURE 5.2: Functional Analysis Architecture for the Analysis Level of EAST- ADL.

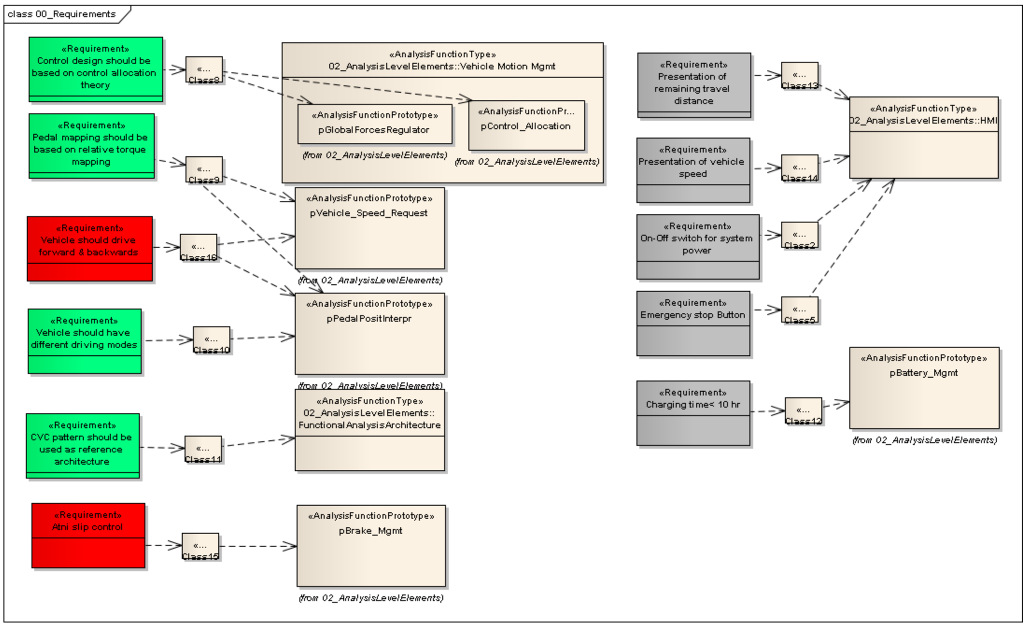


FIGURE 5.3: Presentation of thesis requirements in EAST- ADL. With green colour are the fulfilled requirements, while in red are the uncompleted ones. Grey colour shows requirements not relevant to this thesis, but formulated for the EDV project.

Chapter 6

Implementation in Simulink

This chapter shows how the Simulink model was designed, based on the FAA presented in the previous chapter. The functionality domains are explained in more details. Furthermore, the different systems, which were designed for the implementation of the control allocation are also presented.

For the implementation of the control system four layers of the CVC architecture were used and are shown in figure 6.1.

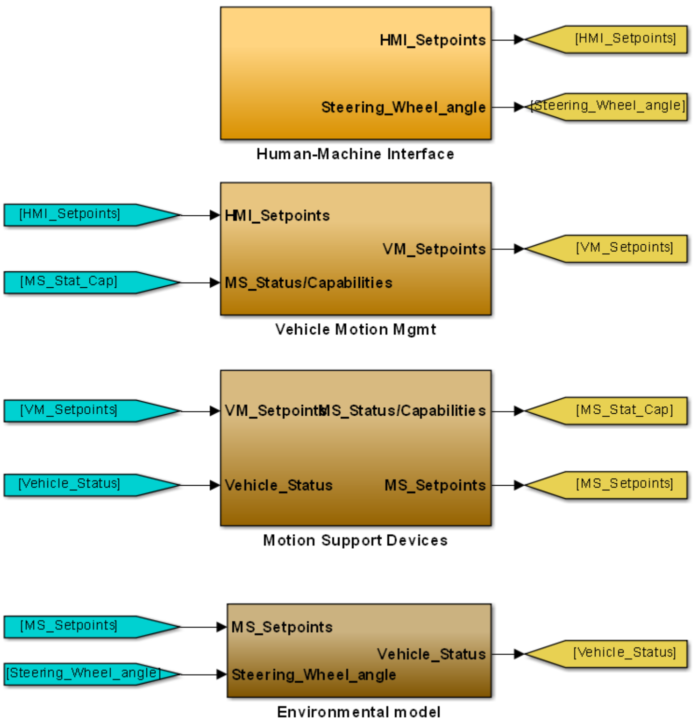


FIGURE 6.1: The four functionality domains of the CVC architecture as systems in the Simulink model.

6.1 Human-Machine Interface

This layer was the one that was directly communicating with the driver and capturing all the signals for the driver's intentions. The ones implemented for this thesis were the selection of driving mode, the gas pedal and the steering wheel angle. The output signals from this layer were used as inputs at the *Vehicle Motion Management* layer. From figure 6.1, it can be observed that the SWA was excluded from the control system and was applied directly to the vehicle model.

6.2 Vehicle Motion Management

This was the most important layer for this project, since the control system was implemented here. Three different systems were implemented, as it is shown in figure 6.2.

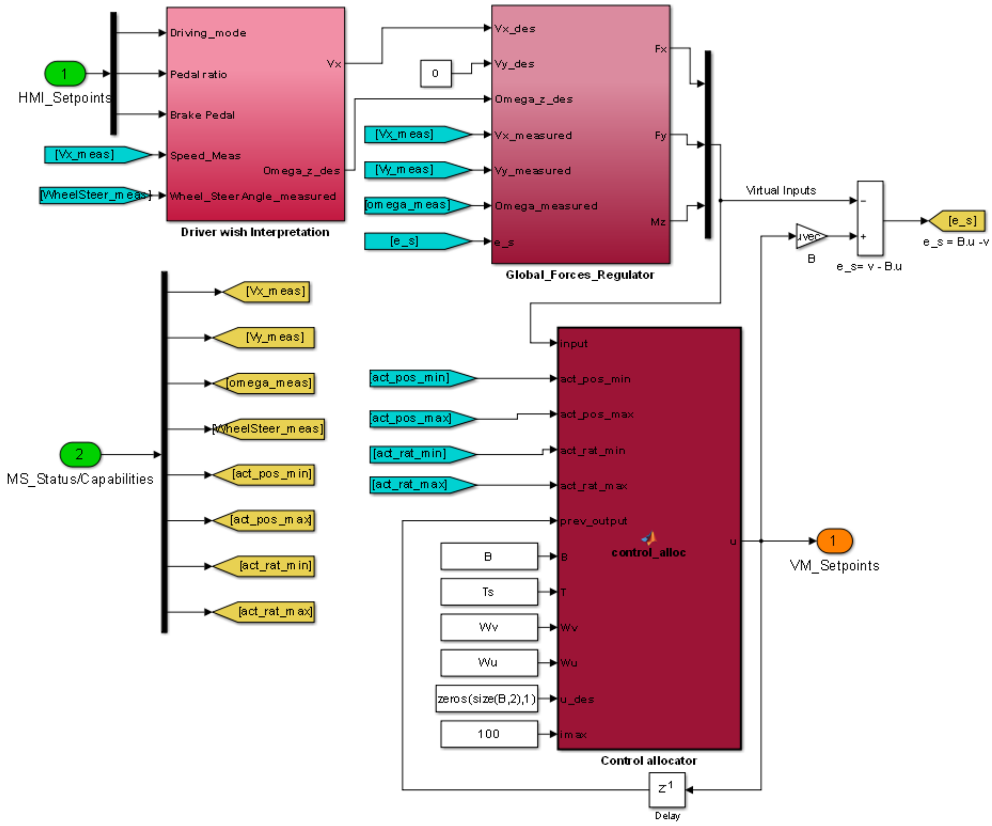


FIGURE 6.2: The three systems of the Vehicle Motion Management functionality domain in the Simulink model.

The first system, which was called *Driver Wish Interpretation*,was responsible for calculating the reference trajectory. It consisted of three different subsystems, each one assigned with a responsibility. The first subsystem was called *Pedal position interpretation* and its responsibility was to map the driver's signals from GP and the DM to some percentage of torque request, $T_{request}$. In the second subsystem, called *Vehicle Speed*

Request, equation (4.22) was implemented for the calculation of the desired speed. The *Yaw Rate Calculation* was the third subsystem and its responsibility was to calculate the desired yaw rate based on equation (4.23). Figures 6.3 - 6.5 show the implementation for these three different subsystems.

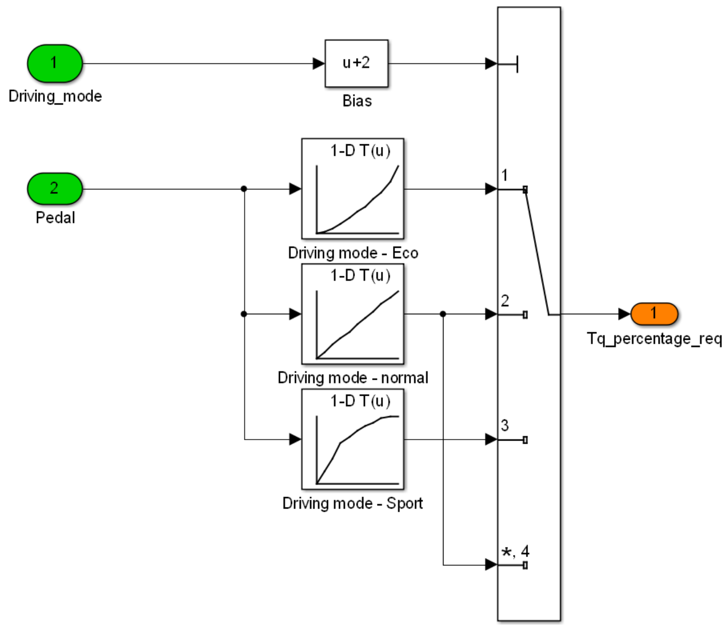


FIGURE 6.3: The Pedal Position Interpretation subsystem as implemented in the Simulink model.

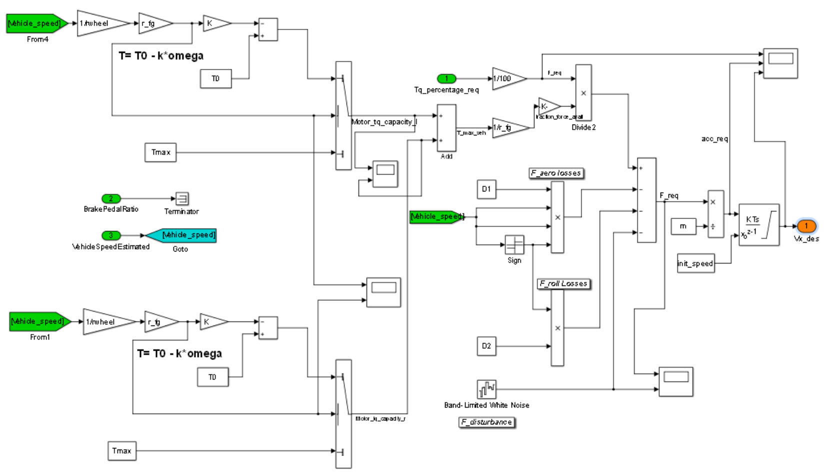


FIGURE 6.4: The Vehicle Speed Request subsystem as implemented in the Simulink model.

The second system was the *Global Forces Regualtor*. The antiwindup PI-controller with feedback linearisation was implemented in this system, evaluating the virtual inputs,

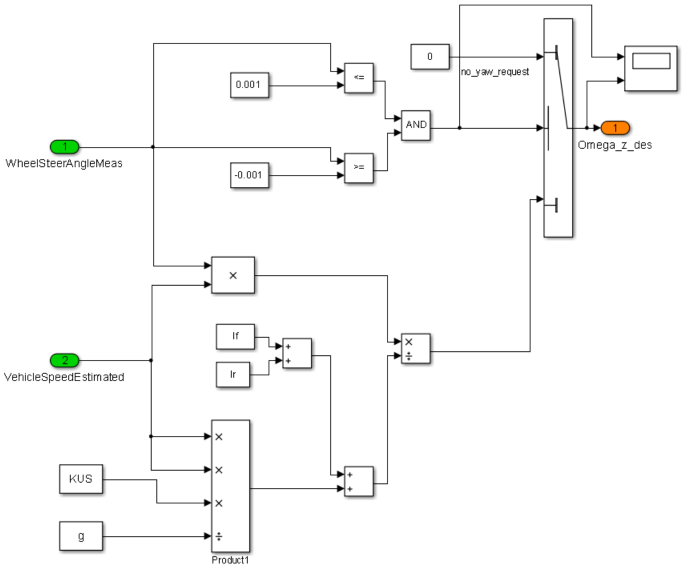


FIGURE 6.5: The Yaw Rate Calculation subsystem as implemented in the Simulink model.

v, which were used in control allocator. Figure 6.6 shows the implementation of the system.

The third system was the implemetation of the *Control Allocator*. It was implemented as a matlab function block, while the control allocation toolbox *QCAT*, downloaded from [17], was used for solving equation (4.9).

6.3 Motion Support Devices- Environmental Model

In the layer called *Motion Support Devices,* three different subsystems were implemented. The *Motors Management*, which was responsible for applying the requested torque from the allocators to the motors and to provide the motors' position and rate limits to the higher level. Another subsystem was the *Braking System Management*, which was responsible for applying the requested torque in the brakes and evaluate the brakes position and rate limits. A third subsystem, called *Body Management* was designed and was responsible for sensoring and calculating the vehicle's body trajectory $y = \begin{bmatrix} V_{x_{meas}} & V_{y_{meas}} & \omega_{x_{meas}} \end{bmatrix}^T$, as well as the angle of the wheels. As mentioned in the limitations section, no models of motors, brakes or sensors were designed. The values calculated from the higher level were directly transferred to the actuators, while the position and rate limits had been set as constant values.

As far as the last layer is concerned, it was called *Environmental Model* and included the model of the vehicle's body.

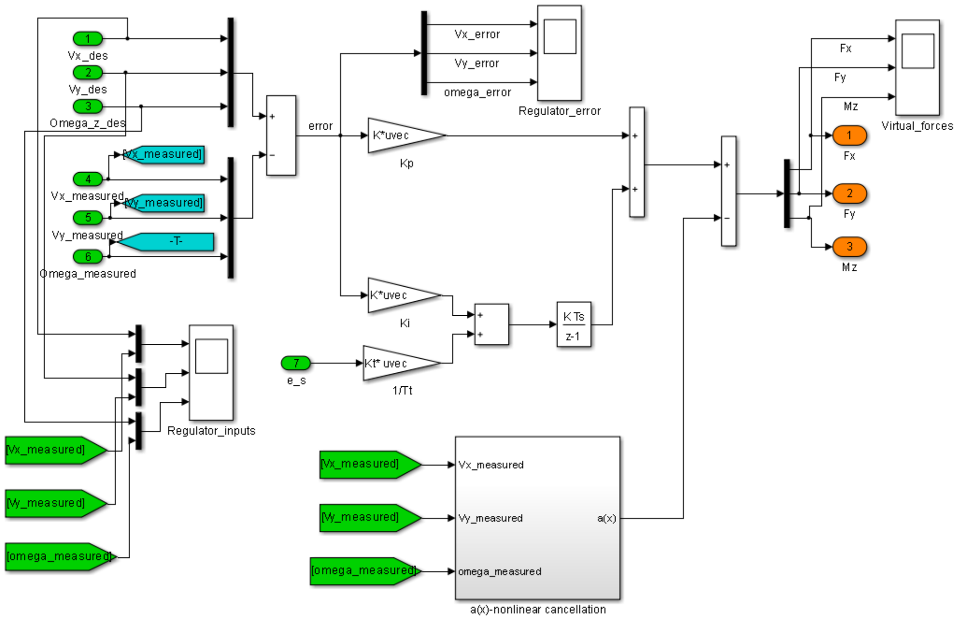


Figure 6.6: The Global Forces Regulator as implemented in the Simulink model.

Chapter 7

Simulation Results

In order to validate that the designed control system had the desired behaviour, different test cases were simulated. Although the test cases were chosen arbitrary, their purpose was to cover various different driving commands. Two groups of tests cases were done. The first was for validating the *Driver Wish Interpretation* system. The simulation tests were independent of the vehicle model. Furthermore the *Yaw Rate Calculation* subsystem was not validated, since similar system was tested in earlier Volvo project. The focus for these tests was the creation of the desired longitudinal speed based on GP ratio and the driving modes.

The second group of simulation tests was done with the completion of the current thesis for the validation of the control allocator. The ODE23 solver was chosen for the simulations. This is a variable-step implicit method solver, requiring fewer time steps than explicit methods and suggested when working with models using SimMechanics toolbox, as the *Environmental Model* was. Although in CVC architecture, different layers have different time spans, for this thesis one sampling time was assigned. Its value, $T_s = 0.2$ *sec* was selected arbitrary, but it was taken in consideration that has to fulfil Nyquist criterion. Thus, it was chosen to have at least the double frequency than what the hardware componets might have. Furthermore, the weighted least squares function of QCAT required as input the number of iterations, which the function was allowed to have before giving a result. The maximum number of iterations allowed by the designer of the function was $I = 100$, which was the selected value during the simulations.

7.1 Validation of Driver's Wish Interpretation

For the validation of the functionality of the system the following test cases were simulated:

1. Simulation using only the *Pedal Position Interpretation* subsystem.
2. Simulation using *Pedal Position Interpretation* and *Vehicle Speed Request* :
   (a) The GP ratio was increasing from zero to one starting at time 0.5 and with a step of 0.2 per two seconds for initial speed $V_0 = 0\ m/s$.
   (b) The pedal ratio was set to $GP = 0.4$ for initial speed $V_0 = 7\ m/s$.

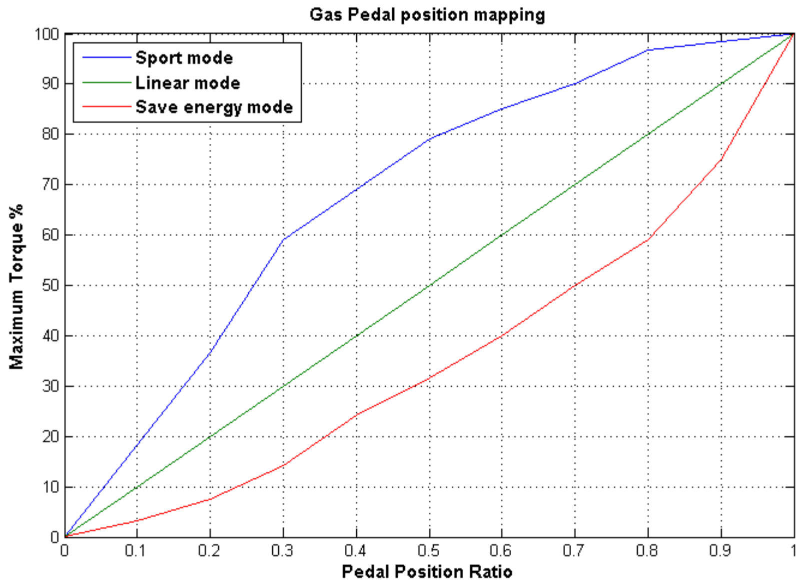


FIGURE 7.1: Test case 1. The GP ratio mapped to a percentage request of the total torque available for the different driving modes.

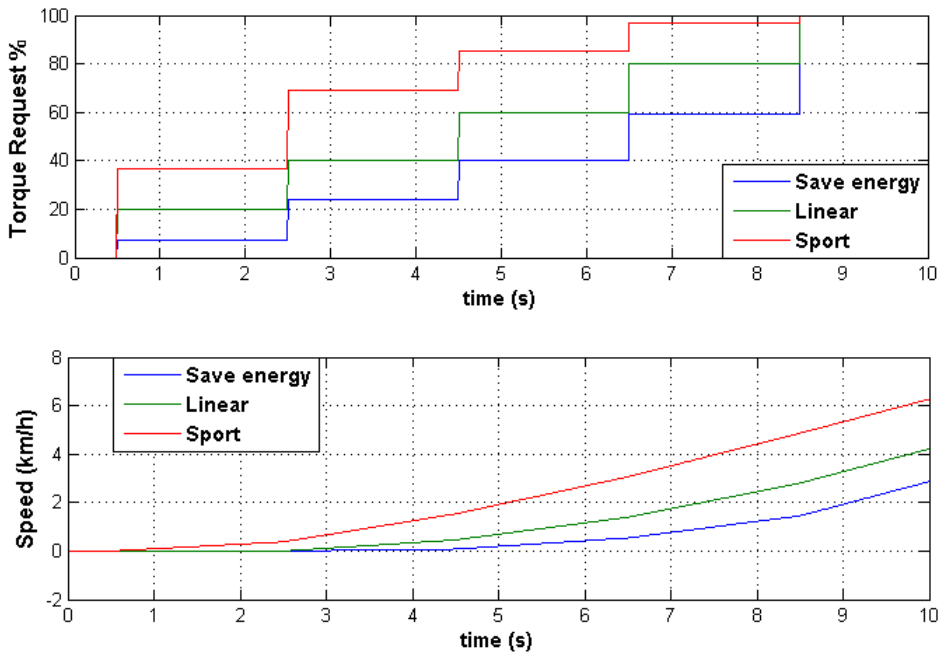


FIGURE 7.2: Test case 2a. The first plot shows the percentage of torque request for the different driving modes, while the second one shows the $V_{x_{des}}$ created during test case 2a.

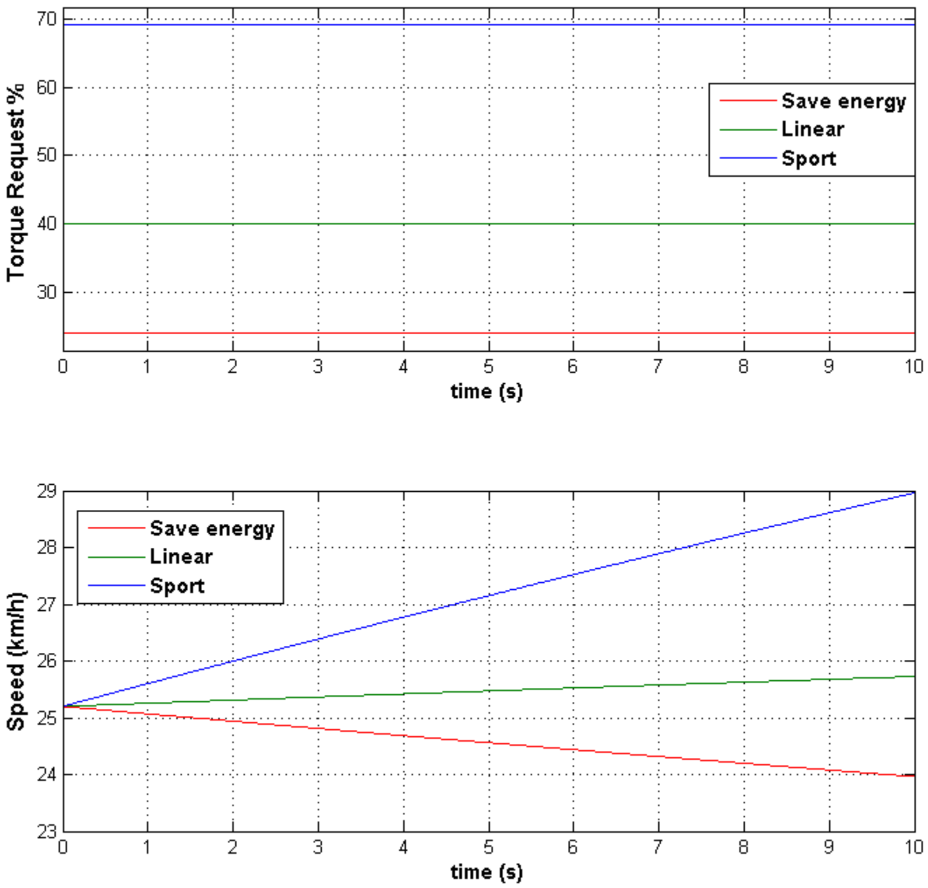


FIGURE 7.3: Test case 2b. The first plot shows the percentage of torque request for the different driving modes, while the second one shows the $V_{x_{des}}$ created during test case 2b.

7.2 Validation of Control System Design

For the validation of the functionality of the system the following test cases were simulated:

3. Acceleration demand for desired speed $V_{x_{des}} = 3\ m/s$ during the first 10 *sec* of simulation and desired speed $V_{x_{des}} = 10\ m/s$ for the rest of the simulation time.
   (a) Maximum torque assigned for motors' limits at $T_{m_{max}} = 18.61\ N.m$
   (b) Maximum torque assigned for motors' limits at $T_{m_{max}} = 10\ N.m$
4. Acceleration demand for desired speed $V_{x_{des}} = 10\ m/s$ during the first 10 *sec*. Then for time $10 < t < 30$, the desired speed was set at $V_{x_{des}} = 20\ m/s$ and finally deceleration demand to $V_{x_{des}} = 5\ m/s$ for time $30 \leq t$
   (a) Maximum torque assigned for motors' limits at $T_{m_{max}} = 18.61\ N.m$ and brake limits at $T_{b_{min}} = -100\ N.m$.

(b) Maximum torque assigned for motors' limits at $T_{m_{max}} = 10\ N.m$ and brake limits at $T_{b_{min}} = -200\ N.m$.

For the test cases 3-4 a direct set of desired speed was made. For the following test cases the *Driver Wish Interpretation* system was also utilised. Furthermore, steering manoeuvre was added to the driver's behaviour.

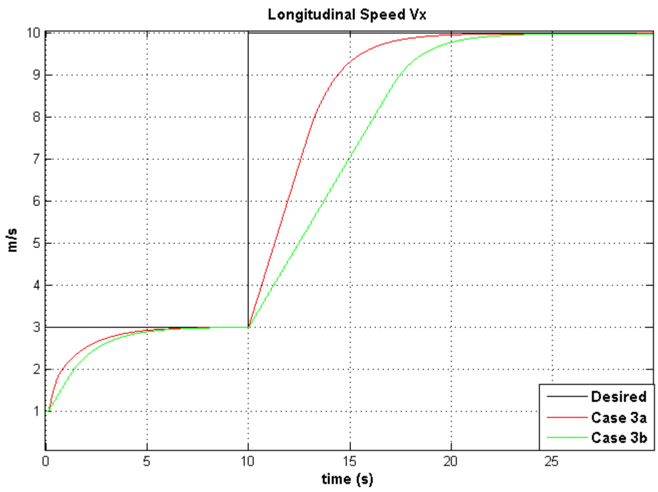
5. For the first 10 *sec* of the simulation test, the desired speed was set to $V_{x_{des}} = 10\ m/s$ and then was set to $V_{x_{des}} = 20\ m/s$. The steering manoeuvres during the simulation are shown in the following table:

Time interval	**SWA**
$5 \leq t < 10$	$1\ rad$
$10 \leq t < 15$	$0\ rad$
$15 \leq t < 20$	$-1\ rad$
$10 \leq t < 25$	$0\ rad$
$25 \leq t < 30$	$1\ rad$
$30 \leq t < 35$	$-1\ rad$

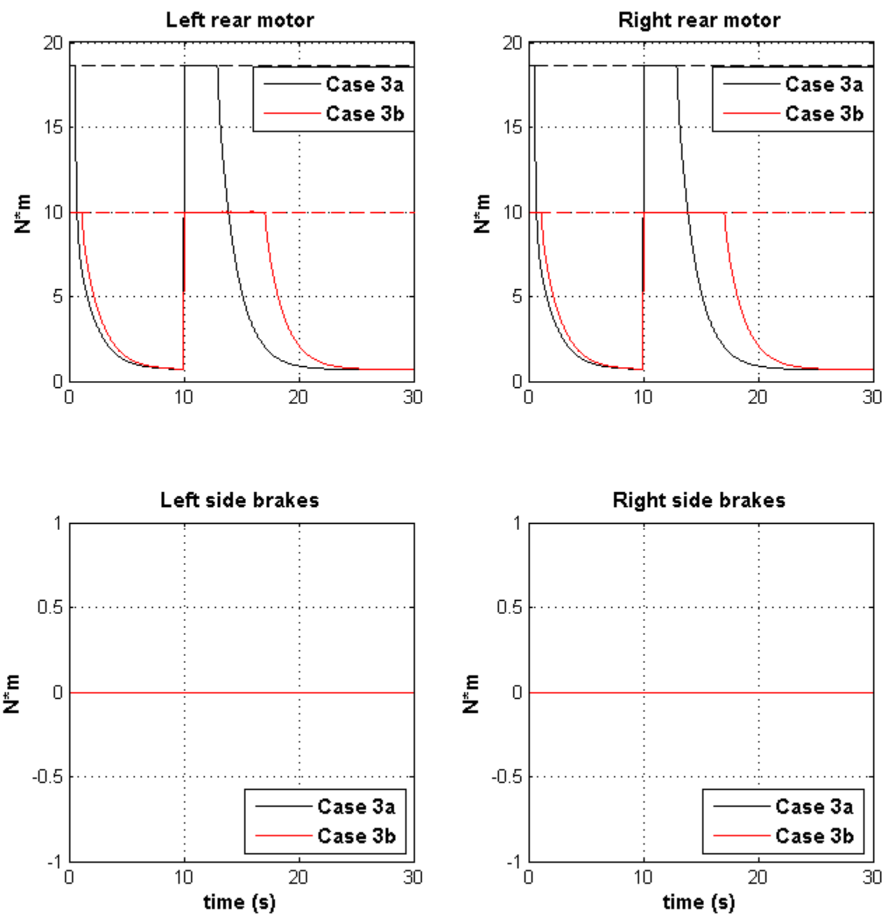
(a) The virtual weighting matrix was as usually used in the thesis' simulations, $W_v = diag \begin{bmatrix} 1 & 1 & 1 \end{bmatrix}$

(b) The virtual weighting matrix was set to $W_v = diag \begin{bmatrix} 1 & 1 & 10^6 \end{bmatrix}$, penalising the yaw moment of inertia M_z more than the other two virtual inputs.

6. As last test case simulation, all the three systems of Vehicle Motion Management were utilised. The driver's commands were given through the GP, while the control allocation was calculating the values of the real inputs based on the estimated desired trajectory. The GP ratio was increasing from zero to one at starting time $t = 1\ sec$, with an increasing step of 0.2 per five seconds.

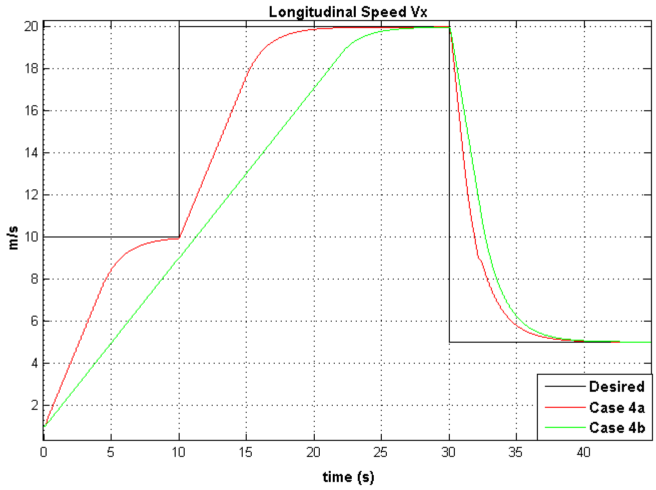


(a)

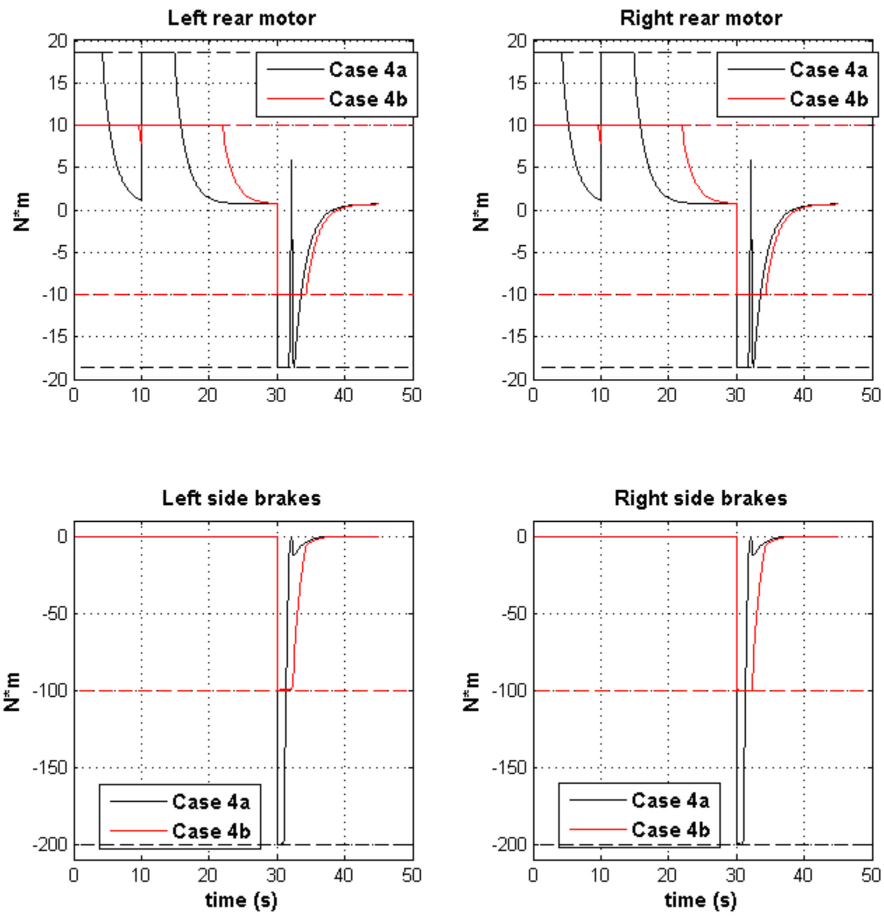


(b)

FIGURE 7.4: Test case 3. **(a):** The desired speed is the speed V_x given to the system, while case 3a and 3b are the measured speeds as outputs from the vehicle model for the two different configurations of motor limits.
(b): The real inputs are presented in this subfigure. The two electric motors and the brakes. The dashed lines show the limits of the motors for each case.



(a)



(b)

FIGURE 7.5: Test case 4. **(a):** The desired speed has been set to the system, while case 3a and 3b are the measured speeds as outputs from the vehicle model for the two different configurations of torque limits in all the actuators.
(b): The real inputs are presented in this subfigure. The two electric motors and the brakes. The dashed lines show the limits of the actuators for each case.

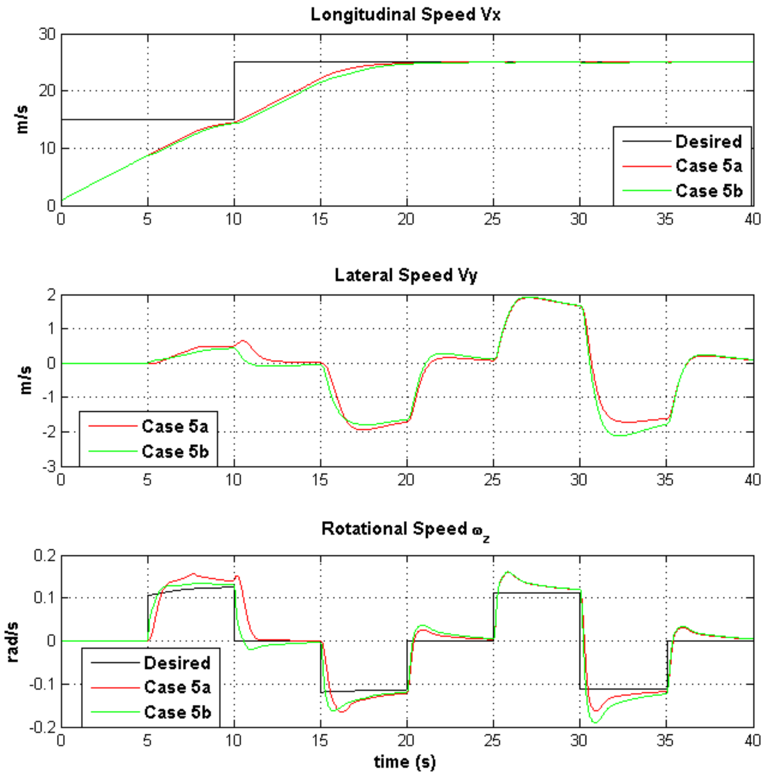


FIGURE 7.6: For the different values of the weighting matrix W_v, the vehicle's three main motions are presented. The values for the desired speed V_x and rotational speed ω_z have been set, but not the lateral speed, since the controller cannot influence this motion.

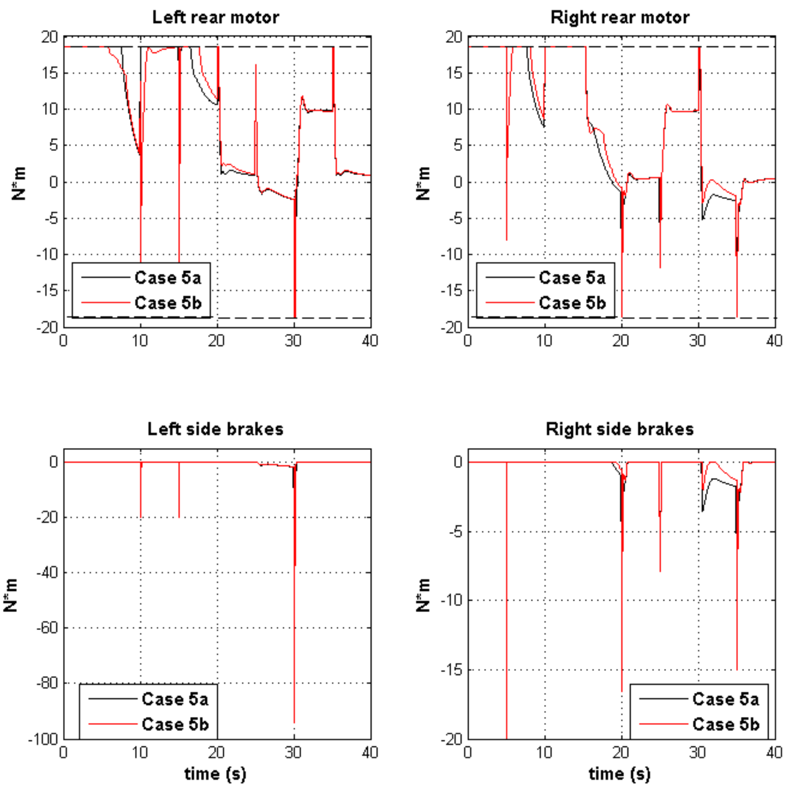
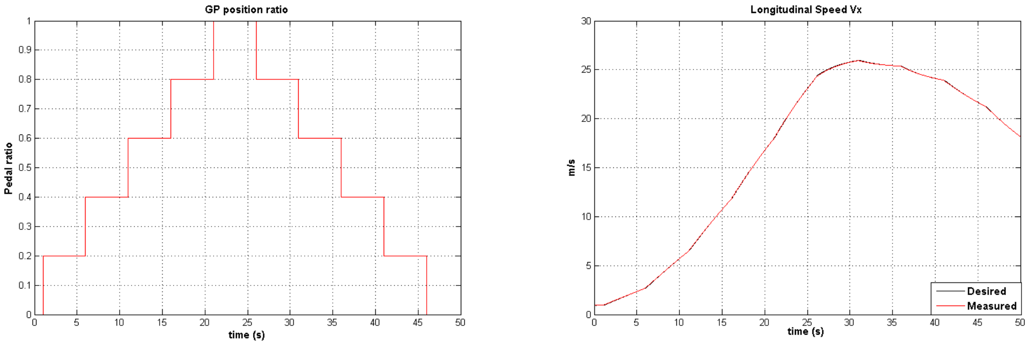
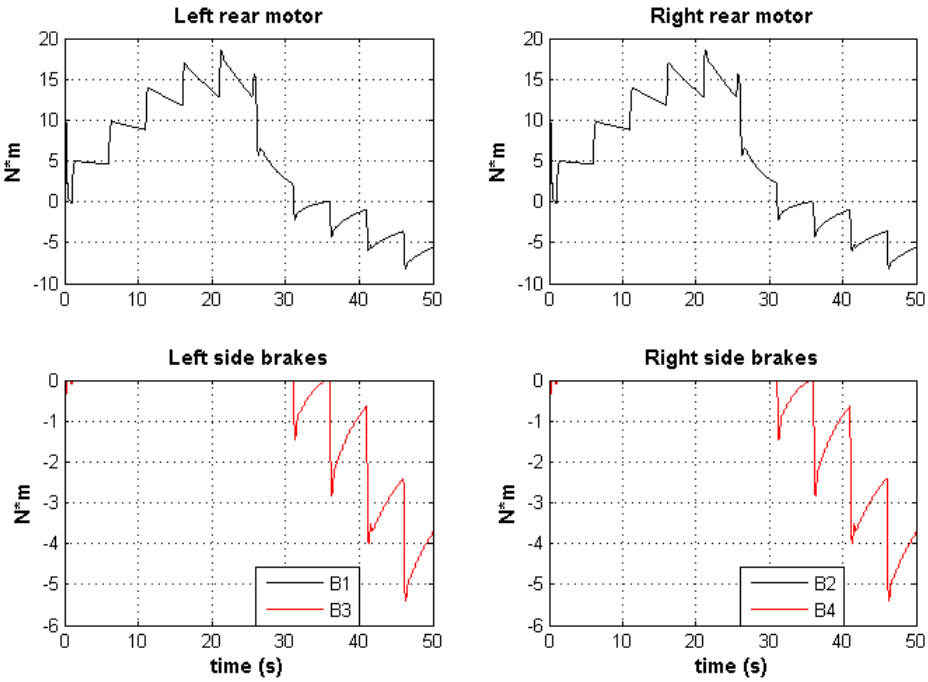


FIGURE 7.7: The real inputs for case 5a, 5b are presented, the two electric motors and the brakes. The dashed lines show the limits of the actuators.



(a)



(b)

FIGURE 7.8: Test case 6. **(a):** The changes of the GP over the time are presented in the left figure. On the right figure the desired speed calculated in *Vehicle Speed Request* subsystem is presented, along with the measured speed of the vehicle model.
(b): The values for the real inputs, which are calculated by the control allocation system in order to achieve the desired spead, are presented in this subfigure.

Chapter 8

Conclusion

8.1 Discussion about the designed control system

After the simulation tests, the general conclusion is that the designed control system functions are as it was expected. This means that the control system is capable of receiving signals as the GP, DM and SWA. Based on these signals a desired trajectory for the vehicle to follow is estimated. Then, the control allocator calculates the values to be assigned to the motion actuators of the system, while respecting the given values of their physical limits, based on the desired trajectory. It has been said that the number of iterations used in the control allocation was 100, but from the simulations, it was observed that the optimal solution was calculated at a maximum of 15 iterations. Each of the simulation tests validates different properties of the whole system.

Test case 1 shows the different maps of torque request depending on the selected DM. Test cases 2a and 2b show the differences in the estimated desired speed because of the selected DM, while using all the GP positions and a constant one, respectively.

It test case 3, it can be seen that the control system respects the motor constraints. It does not matter how much high the desired speed is, the controller will try to achieve that speed but always with respect to the given constraints. Similar observation is made in test case 4, not only about the motors, but for the brake constraints as well. One important issue that should be mentioned at this point is about the torque value that is needed to maintain a constant speed. As it can be seen from the plots, when the desired speed is reached, very low torque is needed to maintain that speed, no matter how much high it is. Of course this is not realistic, but it can be explained by the vehicle model configuration. The vehicle model only takes in consideration the rolling resistance force F_{roll} and not the aerodynamical drag force F_{drag}. The F_{roll} is always constant and has a much lower value compared to F_{drag} and that is why the required motor torque to maintain a constant speed is so low.

Another observation is that the weighting matrices configuration plays an important role and affects the control system. For the real inputs weighting matrix W_u, a priority to use the motors and not the brakes was given. Because of this configuration it can be explained why, when the controller wants to reduce speed it always uses negative motor torque along with the braking system. In test case 5, the importance of the configuration of the virtual weighting matrix, W_v, can be clearly seen. For the case 5a, the desired speed is reached a little bit faster than 5b and is maintained during the rest simulation time. On the contrary, case 5b, where the M_z is highly penalised, the deviation between the measured and the desired ω_z is lower than in 5a. At this point, it is reminded that no steering is included in the controller and that SWA is directly applied to the car, thus when the control system needs to steer the car, this is done using the available motion actuators, like motors and brakes. This can be observed for example at simulation time $t = 10$ *sec*. The controller wants to steer the vehicle to the left, so it applies less torque in the left motor as well as some torque to the left side brakes.

As far as the test case 6 is concerned, the simulation was made in order to test the overall behaviour of the control system. Only the GP was used and the results validate that the control system designed for this thesis, is able to estimate a desired speed based on the driver's commands and the control allocator assigns the appropriate values to the motion actuators so that the desired speed is followed really accurately.

A conclusion that is easily observed from all the test cases with the control allocation, is that the control system reaches the desired speed very smoothly, without any overshoots or oscillations in the steady state. When it comes to automotive industry, the driver's comfort is a key factor. Hence the absence of overshoots or oscillations in the vehicle's speed is a really important attribute that control allocator offers. Moreover, by testing the different systems alone and working all together, the modularity of the control system is validated. This is an advantage because parts of this thesis can be used in other projects as well.

In the last test case, no steering was included, but the GP was used in all of its position. The most important observation in these plots, is that the control system manages to assign the proper values to the actuators so accurately, that the desired speed and the measured speed are exactly the same. As mentioned in earlier chapters, the desired speed is calculated based on dynamic equation and is calculated smoother than in previous test cases when a value was directly assigned. The conclusion is that if the estimated speed is derived from real facts as the dynamic equations, the controller will be able to achieve it

8.2 Discussion about EAST- ADL

In the beginning of the project, there was the dilemma of what software should be used for implementing the EAST- ADL model. There were two candidates, the *Enterprise Architect* and the *Papyrus-Eclipse*. After trying both software the first candidate was chosen. Papyrus-Eclipse is an open-source software and problems in installation and usage were experienced. On the other hand Enterprise Architect was working as expected, while better support was provided by experienced users inside the company and company's manuals. Furthermore, in general it was more reliable, since it has been used for a longer period of time, than Papyrus-Eclipse.

In the EAST- ADL model only the two first levels were designed. The reason for this was because in order to go in the lower levels, more details were needed. Until the end of this thesis, details like the HW or the SW that will be installed in the EDV, were not decided.

As far as the utility of the EA is concerned, the software except for the obvious advantage of offering a way to create various documentations, it also has some drawbacks.Based on my personal experience, I would say that EA is a software easier to use than the other candidate, but still is not so much user friendly. When someone is designing something, it is only natural that the designer will need to erase and re-design several times. Doing this in EA is much time consuming. Apart from that, the program does not have any compatibility with other software but AUTOSAR that is used in the implementation level. This is the reason why until now, it is used mostly for documentation purposes. When EA will be made to be compatible with software like Simulink, then I strongly believe that it will be much more useful tool, not only for documentation but also for analysis.

8.3 Future Work

One of the basic tasks of this thesis was to complete a first version for the control system of the EDV car and validate its desired behaviour. A first version was accomplished but definitely, there are several different things yet to be done, in order for the control system to be implemented in the EDV.

Based on the writer's personal opinion, what could be easily done is to include steering in the control system. Moreover, it is suggested that more detailed models of the motion actuators can be designed. This will lead to better and more precise estimations about the achievable values for the actuators' constraints. A lumped model could be an option, in which the braking and accelerating capabilities will depend on the actuators' temperatures. As far as the braking system is concerned, when the HW components are decided, the real values can be used instead of the current estimations. Additionally, during this thesis the power provided by the battery was considered always constant. A suggested improvement is the design and integration of a battery model as a new functionality area in the *Motion Support Devices* layer.

In this first version of the designed control system, a simple PI controller was implemented. It would be interesting to explore the possibility of using a more advanced controller, as for example a model predictive controller, although this would increase the execution time of the control system. Moreover, for complexity reasons, no disturbances were used in the simulations. If disturbances, like wind speed or noise in sensors, are included in the modelling, more useful observations might be extracted from the simulation tests.

Appendix A

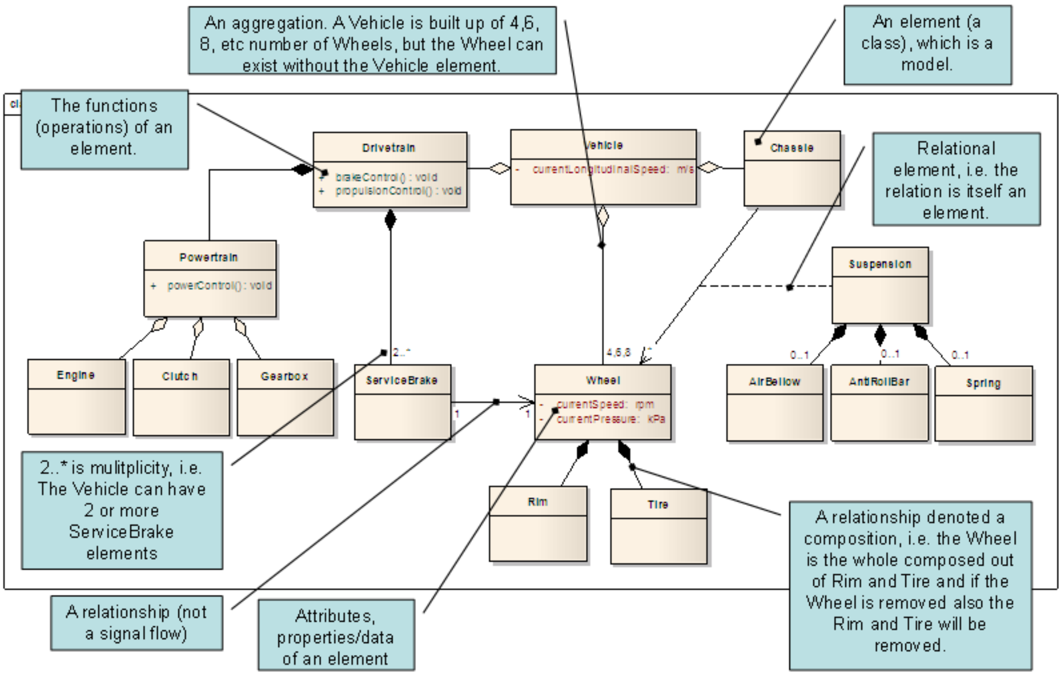


FIGURE A.1: Readers' guideline for EAST-ADL's graphical notation used in this report.

Appendix B

B.1 Vehicle Dimensions

Table B.1: Vehicle's constant parameters.

Parameter	**Symbol**	**Value**
Vehicle's mass	m_{veh}	300 kg
Driver's mass	m_{driv}	75 kg
Total mass	m	375 kg
EDV's length	L	2.5 m
EDV's width	W	1.3 m
EDV's height	H	1.4 m
Distance between rear wheel and CoG	L_r	1.25 m
Distance between front wheel and CoG	L_f	1.25 m
Front axle's width	b_f	1.3 m
Rear axle's width	b_r	1.3 m
Wheel's mass	m_{wheel}	12.7 kg
Wheel's radius	R_w	0.3107 m
Vehicle's body mass	m_{body}	324.2 kg
Wheel inertia around x-axis	$Iwheel_x$	0.6809 $kg * m^2$
Wheel inertia around y-axis	$Iwheel_y$	1.3075 $kg * m^2$
Wheel inertia around z-axis	$Iwheel_z$	0.6809 $kg * m^2$
Cornering stiffnes	Ca	918 $N/degrees$
Gear ratio	r_{fg}	6

B.2 Loss Forces

$$F_{drag} = 0.5 p_a C_d A_f \dot{x}^2 sign(\dot{x}) \tag{B.1}$$

where p_a is the is the density of the ambient air, c_d is the aerodynamic force coefficient and A_f is the frontal area of the vehicle body.

$$F_{roll} = c_r mg \cos(a) sign(\dot{x}) \tag{B.2}$$

where c_r is the rolling friction coefficient, g is the acceleration due to the gravity and a is the slope angle of the road. The rolling friction coefficient depends on many variables like vehicle speed, tire pressure and road surface conditions. During this thesis a typical value for a small car driving in a normal road was assigned to the parameter.

$$F_{grav} = mg\sin(a) \tag{B.3}$$

TABLE B.2: Constant parameters for the loss forces in longitudinal direction.

Parameter	**Symbol**	**Value**
Density of air	p_a	1.225 kg/m^3
Aerodynamic drag coefficient	c_d	0.6
Frontal area of vehicle	A_f	1.28 m^2
Rolling friction coefficent	c_r	0.007
Gravity acceleration	g	9.8 m/s^2
Slope of the road	a	0 $rads$

B.3 Motion Actuators

TABLE B.3: Parameters for the motion actuators.

Parameter	**Symbol**	**Value**
Brushless DC Motor		
Company	-	Golden Motor
Model	-	HPM5000B72V
Rated Voltage	V_{in}	72 $Volts$
Continuous Current	i	100 A
Critical Speed	ω_c	456.7 rad/sec
Maximum Torque	$T_{m_{max}}$	18.61 $N.m$
Minimum Torque	$T_{m_{min}}$	-18.61 $N.m$
Maximum Power	P_{max}	5 kW
Minimum Power	P_{min}	-5 kW
Rated Torque	T_0	75.6 $N.m$
Motor Rate of Change	ρ_m	± 2000
Hydraulic Disc Brake System		
Maximum Braking Force	$T_{b_{max}}$	0 $N.m$
Minimum Braking Force	$T_{b_{max}}$	-200 $N.m$
Braking Rate of Change	ρ_b	± 2000

Bibliography

[1] Andreas Schafer and David G. Victor. The future mobility of the world population. *Transportation Research Part A: Policy and Practice*, 34(3):171–205, April 2000.

[2] A. Faiz, C.S. Weaver, and M.P. Walsh. *Air Pollutation from Motor Vehicles: Standards and Technologies for Controlling Emissions*. Other World Bank Bks. The World bank, 1996. ISBN 9780821334447. URL http://books.google.se/books?id=Hqsyv_KD0lgC.

[3] Grady Booch, Robert A. Maksimchuk, Michael W. Engle, Bobbi J. Young, Jim Connallen, and Kelli A. Houston. Object-oriented analysis and design with applications, third edition. *SIGSOFT Softw. Eng. Notes*, 33(5):11:29–11:29, August 2008. ISSN 0163-5948. doi: 10.1145/1402521.1413138. URL http://doi.acm.org/10.1145/1402521.1413138.

[4] Robert Cloutier, Gerrit Muller, Dinesh Verma, Roshanak Nilchiani, Eirik Hole, and Mary Bone. The concept of reference architectures. *Systems Engineering*, 13(1): 14–27, 2010. ISSN 1520-6858. doi: 10.1002/sys.20129. URL http://dx.doi.org/10.1002/sys.20129.

[5] James Albus, HM Huang, E Messina, K Murphy, M Juberts, A Lacaze, S Balakirsky, MO Shneier, T Hong, H Scott, et al. *4D/RCS Version 2.0: A reference model architecture for unmanned vehicle systems*, volume 6912. 2002.

[6] Autosar. URL http://en.wikipedia.org/wiki/Autosar.

[7] Hans Blom, Henrik Lönn, Frank Hagl, Yiannis Papadopoulos, Mark-Oliver Reiser, Carl-Johan Sjöstedt, De-Jiu Chen, and Ramin Tavakoli Kolagari. EAST-ADL - An Architecture Description Languange for Automotive Software- Intensive Systems. URL http://www.maenad.eu/public_pw/conceptpresentations/EAST-ADL_WhitePaper_M2.1.10.pdf.

[8] L. Guzzella and A. Sciarretta. *Vehicle Propulsion Systems*. Springer-Verlag Berlin Heidelberg, 2007. ISBN 9783540746928. URL http://books.google.se/books?id=y3xN7jZTesUC.

[9] SAE. Surface vehicle recommended practice, vehicle dynamics terminology. J670E, 1976.

[10] L. Laine and J. Fredriksson. *Brake Blending for Hybrid Electric Vehicles using Control Allocation*. Göteborg, Sweden.

[11] Golden motor web site. URL http://www.goldenmotor.com/.

[12] Centric Parts. White paper: The physics of braking systems. URL `http://www.centricparts.com/files/White%20Paper%20Revisions%204-2012/Centric%20White%20Paper%20A1-2012-The%20Physics%20of%20Braking%20Systems.pdf`.

[13] Volvo Trucks Technology. Electrical architecture in future hybrid vehicles.

[14] Ola Härkegård. *Backstepping and control allocation with applications to flight control [Elektronisk resurs]*. Linköpings universitet, Linköping, 2003. ISBN 91-7373-647-3 (pr t).

[15] Karl Johan Åström and Tore Hägglund. *PID controllers : [theory, design, and tuning]*. ISA, Research Triangle Park, 2. ed. edition, 1995. ISBN 1-55617-516-7.

[16] Nicol Louw. Accelerator calibration: Performance feel. URL `http://www.carmag.co.za/technical/accelerator-calibration-performance-feel/`.

[17] Ola Härkegård. Control allocator toolbox for matlab. URL `http://www.mathworks.com/matlabcentral/fileexchange/4609`.