Exploiting individual wheel actuators to enhance vehicle dynamics and safety in electric vehicles

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Doctoral Thesis

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Abstract

This thesis is focused on individual wheel actuators[1] in road vehicles intended for vehicle motion control. Particular attention is paid to electro-mechanical actuators and how they can contribute to improving vehicle dynamics and safety. The employment of individual wheel actuators at the vehicle's four corner results in a large degree of over-actuation. Over-actuation has a potential of exploiting the vehicle's force constraints at a high level and of controlling the vehicle more freely. One important reason for using over-actuated vehicles is their capability to assist the driver to experience the vehicle as desired. This thesis demonstrates that critical situations close to the limits can be handled more efficiently by over-actuation.

To maximise the vehicle performance, all the available actuators are systematically exploited within their force constraints. Therefore, force constraints for the individually controlled wheel are formulated, along with important restrictions that follow as soon as a reduction in the degrees of freedom of the wheel occurs. Particular focus is directed at non-convex force constraints arising from combined tyre slip characteristics.

To evaluate the differently actuated vehicles, constrained control allocation is employed to control the vehicle. The allocation problem is formulated as an optimisation problem, which is solved by non-linear programming.

To emulate realistic safety critical scenarios, highly over-actuated vehicles are controlled and evaluated by the use of a driver model and a validated complex strongly non-linear vehicle model.

it is shown that, owing to the actuator redundancy, over-actuated vehicles possess an inherent capacity to handle actuator faults, with less need for extra hardware or case-specific fault-handling strategies.

[1]wheel actuator=vehicle component which enables control of wheel kinematics and/or forces in the tyre's contact patch.

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I would like to express my gratitude to all the people involved, especially my academic supervisor, Professor Annika Stensson Trigell, at KTH for her support and incessant encouragement during the entire project. I am indebted to my industrial supervisors throughout the stages of this project, Dr Bengt Jacobson at VCC for offering an invaluable guidance on challenges needed to be solved within the automotive industry, and Mr Sigvard Zetterström for his excellent guidance concerning the discipline of applied chassis technologies and for introducing me to the exciting area of *Autonomous Corner Modules* (ACMs). Many thanks are extended to you for your constant involvement, unfailing support and stimulating advice.

I also express my appreciation to Johan Wedlin at Volvo 3P, formerly VCC, for the important role that he played in starting the project and for giving me sensible advice.

Furthermore, I would like to thank my co-authors for inspiring partnerships and for putting my own research into new perspectives, especially Dr Johan Andreasson at Modelon AB for his encouragement and friendship when I was confronted with diverse difficulties. I have received invaluable help and advice from him, particulary with the development of complex vehicle models. I am also grateful to Dr Oskar Wallmark at KTH for fruitful cooperation in merging aspects of control of electrical machines into vehicle dynamics and providing me with useful knowledge which PhD students must possess.

I would also like to thank Dr Fredrik Roos at Scania for our joint work on active control of vertical wheel loads and for his assistance in understanding mechatronics. I owe gratitude to Dr Stefan Solyom at VCC and Dr Jonas

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I express my appreciation to PhD student Johannes Edrén for our joint work on over-actuation, and I am impressed by his progress in developing a scaled ACM vehicle. I would also like to thank members of the steering committee who have not been acknowledged before, for their valuable guidance: Mr Per-Ola Fuxin and Adjunct Professor Urban Kristiansson at VCC.

Finally, I am very proud and fortunate to be married to my wonderful wife, Cathrin, who has given me support, care and understanding throughout. I am blessed by having three wonderful children, Frida, Simon and Sofia, my lights of inspiration. I thank you for your constant support.

Thank you all for your assistance, which has undoubtedly been indispensable for bringing this dissertation into existence.

Mats Jonasson
Göteborg, June 2009

List of publications

The following publications are included in this thesis:

A. M. Jonasson, S. Zetterström and A. S. Trigell, 'Autonomous corner modules as an enabler for new vehicle chassis solutions', FISITA Transactions 2006, paper F2006V054T, 2006.

B. M. Jonasson and O. Wallmark, 'Stability of an electric vehicle with permanent-magnet in-wheel motors during electrical faults', The World Electric Vehicle Association Journal, Vol. 1, pp. 100–107, 2007.

C. M. Jonasson and O. Wallmark, 'Control of electric vehicles with autonomous corner modules: implementation aspects and fault handling', International Journal of Vehicle Systems Modelling and Testing, Vol. 3, No. 3, pp. 213–228, 2008.

D. M. Jonasson and J. Andreasson, 'Exploiting autonomous corner modules to resolve force constraints in the tyre contact patch', International Journal of Vehicle System Dynamics, Vol. 46, No. 7, pp. 553–573, 2008.

E. M. Jonasson and F. Roos, 'Design and evaluation of an active electromechanical wheel suspension system', Journal of Mechatronics, Vol. 18, Issue 4, pp. 218–230, 2008.

F. M. Jonasson, J. Andreasson, B. Jacobson and A. S. Trigell, 'Modelling and parameterisation of a vehicle for validity under limit handling', Proceedings of the 9th International Symposium on Advanced Vehicle Control (AVEC´08), Vol. 1, pp. 202–207, Kobe, Japan, 2008.

G. J. Backmark, E. Karlsson, J. Fredriksson and M. Jonasson, 'Using future path information for improving stability of an overactuated vehicle', accepted for publication in International Journal of Vehicle Systems Modelling and Testing, 2009.

H. M. Jonasson, J. Andreasson, B. Jacobson and A. S. Trigell, 'Global force potential of over-actuated electric vehicles', accepted for publication in International Journal of Vehicle System Dynamics, 2009.

I. M. Jonasson, J. Andreasson, B. Jacobson and A. S. Trigell, 'Investigation of the non-convex force constraints imposed by individual wheel torque allocation', submitted for publication, 2009.

J. M. Jonasson, J. Andreasson, A. S. Trigell and B. Jacobson, 'Utilisation of actuators to improve vehicle stability at the limit: from hydraulic brakes towards electric propulsion', Proceedings of the 21st International Symposium on Dynamics of Vehicles on Roads and Tracks (IAVSD'09), Stockholm, Sweden, 2009.

Publications referred to in this thesis, but not appended are:

K. M. Jonasson, 'Aspects of autonomous corner modules as an enabler for new vehicle chassis solutions', Licentiate Thesis in Vehicle Engineering, TRITA-AVE2006:101, KTH Vehicle Dynamics, Stockholm, Sweden, 2007.

L. O. Wallmark and M. Jonasson, 'Vehicles with autonomous corner modules - control and fault handling aspects', Proceedings of the Program Review Meeting - MIT Industry Consortium on Advanced Automotive Electrical/Electronic Components and Systems, Seattle, U.S.A., 2007.

M. J. Andreasson and M. Jonasson, 'Vehicle model for limit handling - implementation and validation', Proceedings of the 6th Modelica Conference, Bielefeld, Germany, 2008.

N. J. Andreasson, M. Jonasson and H. Tummescheit, 'Modelica-Simulation aktiver Sicherheitsscenarios mit validierten Fahrzeugmodellen in Dymola', Proceedings of the ASIM-Workshop 2009, Dresden, Germany, 2009.

O. J. Edrén, M. Jonasson, A. Nilsson, A. Rehnberg, F. Svahn, J. Andreasson and A. S. Trigell, 'Modelica and Dymola for vehicle dynamics applications at KTH', to be presented at the 7th Modelica Conference 2009, Como, Italy, 2009.

P. J. Edrén, M. Jonasson, A. S. Trigell and J. Jerrelind, ‘The development of a down-scaled over-actuated vehicle equipped with autonomous corner module functionality’, submitted for publication, 2009.

Contributions of individual authors

The author of this thesis, is the first author of **Paper A**, **Paper B**, **Paper C**, **Paper D**, **Paper E**, **Paper F**, **Paper H**, **Paper I** and **Paper J**.

In **Paper A**, the author of this thesis undertook all the writing, while Mr Sigvard Zetterström and Professor Annika Stensson Trigell contributed the problem formulation and useful background material. In addition, they contributed valuable comments and took part in reviewing the paper.

The contributions of the individual authors in **Paper B** and **Paper C**, as concerns the modelling and the writing of the paper, are equal. In these papers, Dr Oskar Wallmark has described the electrical parts used and their behaviour during electrical faults. In addition, he has performed measurements on electrical machines and simulations of power electronics. The author of this thesis has contributed to the method of tyre force allocation adopted and the tyre models used.

In **Paper D**, the contribution from both authors are equal. The author of this thesis has prepared all the models and simulations and the paper has been jointly written. Dr Johan Andreasson has given essential inputs concerning the approach adopted. One of them is related to the interpretation of vertical tyre force allocation and the subsequent evaluation of the adhesion potential applied.

The work in **Paper E** was performed together with Dr Fredrik Roos whose support knowledge and methods in dimensioning electro-mechanical systems. Dynamic models and control laws were jointly developed and both authors took part in the evaluation and the writing of the paper.

In **Paper F** and **Paper J**, the author of this thesis defined the goals and tasks, while Dr Johan Andreasson contributed by modelling the vehicle in Dymola. The part of the work which involved planning of the tests, instrumentation of the vehicle and measurements was performed by the author of this thesis. The papers were written jointly.

In **Paper G,** Johan Backmark, Erik Karlsson and Dr Jonas Fredriksson all took part in the control design. The author of this thesis contributed the problem formulation and provided the vehicle, tyre and actuator models. The paper was written jointly.

Paper H and **Paper I** were written solely by the author of this thesis. Essential inputs to the formulation of the tyre constraints and the brute-force method were given by Dr Johan Andreasson. Professor Annika Stensson Trigell and Dr Bengt Jacobson contributed valuable comments and took part in reviewing the paper.

In **Paper J**, the author of this thesis developed the reference model, the vehicle controller and the force allocator, while Dr Johan Andreasson gave support concerning the driver and vehicle model in Dymola. The co-authors have provided useful ideas, as well as proofreading the manuscript. The paper was written solely by the author of this thesis.

Contents

Chapter 1

Introduction

This chapter provides the background for this research and presents the research question. Finally, an outline of the thesis is provided.

1.1 Research background

This research started at Volvo Car Corporation (VCC) in 2004 after a long period of promising ACM design proposals from the inventor Mr Sigvard Zetterström at VCC. The reason for the following research has been to prepare for the transition to a new type of vehicle chassis, partly driven by new propulsion technologies and energy buffers. Keeping the approaching shortage of cheap fossil fuels and the environmental challenges in mind, there is a need to understand chassis implications in the use of alternative propulsions other than those offered by the combustion engine. New energy converters and buffers set new conditions for chassis development in its entirety.

In addition, there has been an interest in understanding the implications of a vehicle configuration where each corner can act independently without any mechanical connections between the wheels and between the wheels and the power source.

Along with the development and evolution towards hybrid electric vehicles, electrical machines are being introduced as actuators for propulsion. This new class of actuators possesses a potential for designing the vehicle with less compromises to fulfill a variety of functional demands. From the perspective of the car manufacturers, it is important to promote the use this opportunity to improve preventative safety and lower energy consumption further.

With the above developments in mind, Professor Annika Stensson Trigell and Dr Johan Andreasson, a former PhD student, were in the process of investigating a new research approach to controlling vehicles involving generic methods of tyre force allocation. The thoughts behind these ideas came from knowledge of flight control. Therefore, KTH started a joint research project, supported by the Swedish National Energy Agency and VCC, designating ACMs as the target for research.

1.2 Research question

Demands on safety and a sustainable mobility have undergone a dramatic change during the past decade. In general, classical vehicle design does not allow for adding new functions, or improving existing ones, without adding a significant cost. Striving for better performance, the use of actuators, sensors and control architecture are questioned.

This thesis explains the opportunities for vehicle dynamics that over-actuation brings to vehicles. In investigating these opportunities, a vehicle restricted only by the constraints in the tyre contact patch serves as a representative of the 'ultimate' vehicle in its all manifestations. The 'ultimate' vehicle does not exist in production today, but it serves as an important reference in terms of setting the limits of vehicle dynamics. Another important reference is today's vehicle, which together with the 'ultimate' vehicle forms a span of interesting vehicle configurations in which vehicle motion can be generated and controlled.

This thesis explores in depth how different actuators support the generation of forces. Particular focus is directed on electric wheel torque actuators under combined slip conditions. The ACM chassis solution is used as an example of an implementation of the 'ultimate' vehicle. A portion of the work in this thesis formulates prerequisites for functional requirements of such technology.

Given this background, the research question is formulated as follows:

> *How can individual wheel actuators improve vehicle*[1] *dynamics and safety and how should the actuators then be used?*

In the context of this formulation, the term safety serves as expression of the vehicle's ability to perform the desired motion in a stable way. The desired motion can be based on human driver supported from environment sensors. Safety classes under consideration are *driver assistance* and *actuator redundance*, both in accordance with the definitions outlined in Chapter 2.

[1] Throughout this work, the vehicles under consideration are by default passenger cars, unless otherwise stated.

The research approach is based on an optimal use of the actuators and the exploitation of their full capacity of force generation.

1.3 Outline of thesis

This thesis is written as a compilation where the appended papers constitute the majority of the pages. Chapters 1–10 aim at giving an introduction to the research area and providing viable links between the appended papers.

Chapter 2 explains under- and over-actuation and the role that they play for a number of systems. The evolution of over-actuation in vehicles and the opportunities gained from over-actuation are also discussed. Chapter 3 introduces wheel corner concepts and their characteristics. Particular attention is paid on the ACM technology. The constraints arising from tyres and actuators are formulated in Chapter 4. The ‘ultimate’ vehicle is presented and also common force restrictions. In addition, this chapter also contributes a discussion concerning the force potential of over-actuation. Next, in Chapter 5, the control method used here to evaluate over-actuated vehicles is formulated. Chapter 6 presents the modelling of an environment to evaluate different types of wheel actuation. Chapter 7 gives a brief summary of all the appended papers and explains how they are linked together. Moreover, the main scientific contributions of the thesis, including the appended papers, are presented in Chapter 8. Finally, in Chapter 9 and 10, the concluding remarks and recommendations for future work respectively are presented.

Chapter 2

Introduction to over-actuation in vehicles

This chapter explains under- and over-actuation and the role that they play for a number of systems. The development of over-actuation in vehicles and the opportunities gained from over-actuation are also discussed.

2.1 Under-actuation versus over-actuation

Historically, vehicles have been developed with a low number of actuators to control vehicle motion. The combustion engine and friction brakes have served as actuators for vehicle longitudinal motion control, while driver-induced mechanical steering at the front axle has been the classical actuator for controlling cornering independently from traction. Initially, these actuators were used to control the vehicle's degrees of freedom separately; i.e. the combustion engine to support the positive propelling force and the brakes to support the negative propelling force. Most likely, the selection of the actuators was made to provide a sufficient level of functions with as low a complexity as possible. However, although this actuator topology has remained unchanged, the vehicle demands have undergone a remarkable change. The most obvious changes are the increasingly sophisticated demands for even more environmentally friendly and safe solutions, which are driving the development process of vehicles towards new and refined chassis solutions. Since the conventional vehicle can rarely be tuned, there has been an increasing interest in a higher level of over-actuation which meets the new demands and still provides simplicity.

If the number of actuators is less than the states intended for control, the system is designated as under-actuated [1]. Typical objects which belong to the class of under-actuated systems are cranes, underwater vehicles, missiles, spacecraft and marine vessels. One typical example of such a system is a marine vessel equipped with one rudder and one propeller only. Neglecting rolling, the marine vessel has mainly three degrees of freedom to control: rotation around the vertical axis and lateral and longitudinal translation. Since there are two actuators only for controlling the motion in three degrees of freedom, rotation and lateral translation become coupled, which in turn limits the manoeuvering capacity [2–5]. As soon as the number of actuators is equal to the states intended for control, the system is described as equal-actuated [1].

In contrast to under-actuation, a system becomes over-actuated as soon as the number of actuators exceeds the states intended for control (the degree of over-actuation will be further discussed in Chapter 5). Such systems are found frequently among the variety of systems in nature, where the majority of well-developed bio-mechanical systems possess over-actuation. Just one remarkable example of an over-actuated bio-mechanical system, developed by natural selection during 200 million years, is the crab. The crab has ten protruding legs, which can be considered as distributed actuators generating forces to its rigid body close to the contact points of the sea bed. Undoubtedly, the provision of over-actuation has been successful in maximising the crab's chance to survive and reproduce itself. An extraordinary overview of animal motion and the role the muscles of animals play as actuators is found in [6, 7].

Animals cannot compete with road vehicles when it comes to maintaining high speed for longer periods of time. For shorter periods, however, the agility and manoeuverability of many over-actuated animals exceed the capacities of a modern road vehicle. One important difference between the motion of animals and vehicles is the environment they are developed to act in. Passenger cars are designed with wheels and they are often restricted to being driven on relatively hard, smooth and level surfaces, while the over-actuated crab is not. When the vehicle is to cross over to uneven surfaces with acceptable comfort, an efficient wheel suspension must be added. This can be seen as a way of changing the tyre diameter to suit the unlevelled surface. An animal with limbs performs similar actions, but more efficiently by actively changing the length of the limbs [6]. For these reasons, over-actuated animals possess an adaptive capacity of moving themselves in a variety of environments. Any comparison between animals and vehicles can be conceived as a far-fetched speculation. However, they are both subjected to evolution, but with different objectives. While animals are aiming to secure the reproduction of their own species, vehicles are taking part in their own competition, which concerns the provision of the best value for a lower price.

Apart from in nature, over-actuated systems are found in more advanced marine vessels, aircraft and road vehicles. Advanced aircraft are highly over-actuated, and military attack aircraft can hold as many as 20 actuators, including tails, ailerons, flaps, rudders and spoilers, to control the aircraft's six degrees of freedom [8]. Figure 2.1 shows three objects equipped with different numbers of actuators, all poised to control three degrees of freedom.

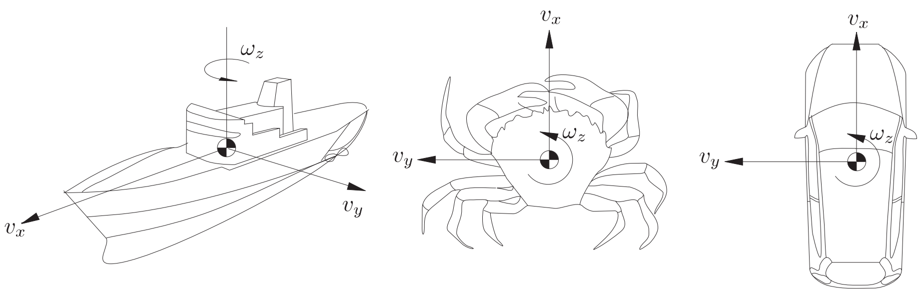

Figure 2.1: Three systems that mainly need to control their motion in the horizontal plane, with the corresponding velocities referred to as the longitudinal velocity (v_x), the lateral velocity (v_x) and the yaw rate (ω_z). The marine vessel with one rudder and one propeller is under-actuated, while the crab with ten legs and the modern passenger car with individual brakes are both over-actuated.

2.1.1 Over-actuation in vehicles

According to the definitions of under- equal and over-actuation previously provided, these vehicles were equal-actuated if the planar motion is considered only (three actuators, three states). Since the engine and brakes both are intended to control wheel torques but with different signs, it is clear that they for most circumstances complement each other. However, if the brakes and engine, for the sake of simplicity, both are considered to operate at the same axle(s), those actuators can theoretically be replaced with just one actuator which admits torque to be generated independently of the wheel rotational direction. Thereby, the vehicle becomes under-actuated (two actuators, three states). This reasoning indicates that the measure of under- equal and over-actuation does not warrant the effectiveness of the implemented actuators. This will be discussed later on in Chapter 4, where a *brute force* technique is used to illustrate the effectiveness of actuators on a global vehicle level.

For the early-stage vehicle, each actuator had a corresponding driver instrument: the steering-wheel to control the front steering angles, as well as the accelerator and the brake pedal to control the engine torque and brake torque respectively. It is noteworthy that along with the increasing demands on vehicle dynamics, and in particular dynamic safety and fuel economy, vehicle technology has developed and a trend towards providing the vehicle with more actuators for propulsion has emerged. The most common provision is individually actuated friction brakes, which have become a standard in the majority of passenger cars in the premium segment. The transition from commonly actuated to individually actuated friction brakes has been driven by the development of anti-lock braking systems and electronic stability control systems. Adding the four individual brakes to the mechanical steering and the engine, the modern premium car can be considered as over-actuated (six actuators, three states).

Here it is worth mentioning that the degree of freedom influenced by one actuator varies depending on the driving situation. For example, hand and foot brakes are driver instruments which can be used to control deceleration only when driving straight ahead. As soon as the vehicle is cornering, the handbrake influences the translation and rotation of the vehicle differently. The reason for this is that the foot brakes engages both axles, while the handbrake engages the rear axle only. Another example is vehicles undergoing an actuator failure, where one or more states risk being left out of control. In severe cases, the reduction of control jeopardises the tracking of the vehicle and passenger safety is threatened. From this discussion it is clear that failures of actuators intended for vehicle motion control could quickly cause a transition from over-actuation to under-actuation.

Classically, vehicles have been provided with additional actuators to strengthen specific attributes. One way to illustrate this, is to divide it into branches depending on the actuated wheel degrees of freedom. Figure 2.2 shows the equal-actuated vehicle (on the left side in the illustration), as it was designed before 1980; i.e. provided with a combustion engine, mechanical steering and a simultaneously actuated brake system. From this level, the actuation can be further extended by exciting either wheel torque or steering or vertical loads. This is, however, a large simplification, since there also exist combinations of the individual branches. Nevertheless, combinations among them have been rare, mainly because of excessively complex and costly solutions. Moreover, the camber degree of freedom is intentionally ignored in Figure 2.2. Although, findings about using camber control to enhance vehicle dynamics have been reported in the literature, e.g. in [9–11] and in **Paper A**. The evolution towards control of the vertical loads is predominantly motivated by an increased ride comfort, while control of steering has been widely reported in the scientific lit-

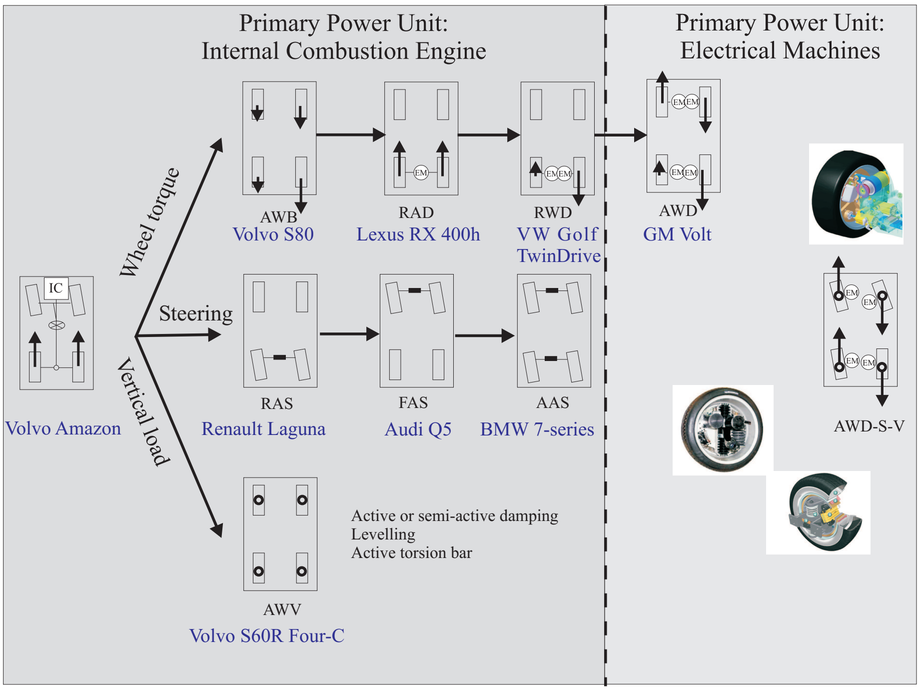

Figure 2.2: The implementation of over-actuation in cars divided into branches depending on the wheel actuated. Each branch contains a number of vehicle configurations, where the configuration nomenclature is defined in **Paper H**. The dashed vertical line represents the transition to electrical machines as the primary power units. The blue text gives examples of products on the market.

erature as solution to support for better vehicle handling. The branch involving the control of wheel torques includes brake systems as well as drive systems.

For each branch in Figure 2.2, there is a variety of different actuator set-ups forming different vehicle configurations. In **Paper H**, a general nomenclature is developed to classify over-actuation in road vehicles. The introduced nomenclature is motivated by the need to clarify more distinctly whether the forces are generated at a specific axle or wheel. Generally, configurations can be classified by the information vector:

$$[\text{actuated axle}][\text{differentiation of actuation}][\text{degree of fredom under control}] =$$
$$[\text{All}, \text{Front}, \text{Rear}][\text{Wheel}, \text{Axle}, \text{Side}][\text{Drive}, \text{Brake}, \text{Steer}, \text{Camber}, \text{Wheel load}]$$

where at least three letters are used. The first two letters denote which wheels are engaged, and the third letter denotes which degree of freedom is under control. The degree of freedom gained from mechanical front steering is disregarded. Note that *Drive* includes both positive and negative wheel torques actuated independently of the wheel's direction of rotation, while *Brake* means the wheel torque counter-directed to the rotation. *Wheel load* means the the vertical tyre force and *Camber* means the camber tyre angle.

Examples associated with Figure 2.2:

AWB (*All Wheel Brake*): All wheels individually braked
RAD (*Rear Axle Drive*): Both rear wheels commonly driven
RWD (*Rear Wheel Drive*): Both rear wheels individually driven
AWD (*All Wheel Drive*): All wheels individually driven
RAS (*Rear Axle Steer*): Both rear wheels commonly steered
FAS (*Front Axle Steer*): Both front wheels commonly steered [1]
AAS (*All Axle Steer*): Both axles individually steered
AWW (*All Wheel Wheel-load*): All wheels individually vertically actuated
AWD|S|W (*All wheel drive and steer and wheel-load*): All wheels individually driven, steered and vertically actuated

Whenever it is necessary, more letters can be added to expand the description, e.g. AWB|FAD means that the front wheels are commonly driven combined with individually actuated brakes.

The trend described in Figure 2.2 depends on an important transition, marked with a dashed vertical line, of the primary power unit from the combustion engine to electrical machines. The transition, which is ongoing, involves the development of hybrid electric vehicles. The wheel torque branch is strongly influenced by the transition, since electrical machines intended for propulsion can be used to control wheel torques differently. One example is the opportunity to divide the electrical power unit, without adding any significant cost, into several units. This division can be used to design an AWD vehicle, without the need for a complex transmission between the axles and wheels, which is needed when the combustion engine constitutes the only power unit. Thereby, the change of actuator type along with a larger degree of over-actuation provides a functional advantage (namely wheels being driven individually) with a smaller number of sub-systems.

Since combinations among the branches in Figure 2.2 traditionally have resulted in technically complex solutions, there has recently been an increasing

[1] Superposition of steering angles from mechanical front steering, also commonly referred to as active front steer.

interest in chassis solutions where each wheel is controlled individually and designed with fewer force and kinematic wheel-to-wheel restrictions (see the right-hand side of Figure 2.2). The existence of such solutions relies on electrical machines, and an electrical architecture of the power source and power distribution is essential if they are to materialise.

In a strategic review of wheel corners [12], it is claimed that the development is expected to go through four steps. The first step is the adoption of two in-wheel motors on the rear axle to provide a non-complex four wheel drive (RWD|FAD). The second step is the implementation of regenerative braking at the two rear in-wheel motors to reduce the fuel consumption. The third step is an extension with electromechanical steering on the rear wheels. The final and fourth step is the implementation of four wheel corners, with individual drive and steer, as well as electromechanical suspension.

2.2 Opportunities of over-actuation in vehicles

Traditionally, vehicles have been provided with additional vehicle motion actuators to enhance a specific function of vehicle dynamics. As an example, actively controllable couplings have been used to allocate torque between the front and the rear axle in two axle drive vehicles (AAD) and vehicles provided by in-wheel motors (AWD) have been suggested in e.g. [13] to reduce delay times for quick yaw and lateral response. Over-actuation influences the vehicle design and its attributes in many ways, some of which are directly noticeable for customers, such as product cost, styling, motion capability, comfort, handling and safety. There are also characteristics which are not directly noticeable to customers, such as a modified development and production process, logistics, packaging, the use of standardised components etc. This section serves to give a background to how over-actuation is related to technical complexity and to explain the use of over-actuation to enhance safety and energy consumption.

2.2.1 Technical complexity

As previously mentioned, increasing the number of actuators has traditionally resulted in more technically complex and expensive solutions. Even if extra actuators are added for increase a benefits concerning the vehicle motion, the additional complexity can be considered too high in relation to the benefits gained. One reason is that the additional actuators, in many cases, are installed without changing the basic structure of the vehicle design. Figure 2.3 shows different steering systems seen from above. In the system shown in Figure 2.3a, the steering system is defined by a classic design with a steering rack and pinion. Adding an extra steering actuator and a planetary gear to this system, shown in Figure 2.3b, results in possibility of adding an additional steering angle. Since the modification is made without removing existing parts, the degree of complexity, at least as concerns the number of sub-systems, is increased. The steering system can, however, be further modified with a lower number of sub-systems adding even more actuators, as for example is shown in Figure 2.3c. The selection of the best solution depends on various factors, where the essential ones being legislation, the sub-system cost and the engineering time. Systems (a) and (b) need a steering column whose design is highly determined by passive safety demands. In contrast, the steering column is not needed in system (c). Nevertheless, new challenges emerge simply beacause that mechanical links are replaced by steer-by-wire. Some of them are the legislation concerning steer-by-wire, control design and force feedback to the driver. Another challenge with system (c) is to achieve acceptable physical dimensions of the actuators, since they are closely distributed to the wheel and partly contribute to the wheel's unsprung mass. Therefore, the choice of actuator type plays an important role.

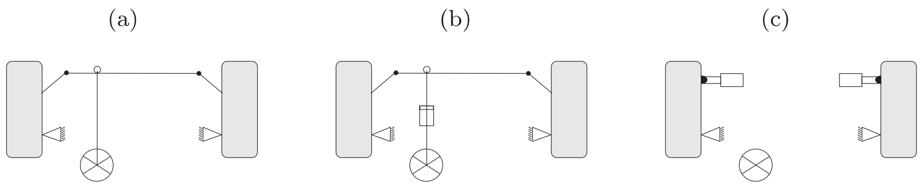

Figure 2.3: Top view of different steering systems. (a) A classical rack and pinion steering system. (b) Steering system extended with an extra steering actuator which adds an offset angle between the steering-wheel and wheel via a planetary gear (also referred to as 'Active Front Steering'). (c) Steer-by-wire steering system.

System (c) poses challenges which are typical of over-actuated systems in vehicles and need to be considered during the early stage of the development process.

2.2.2 Safety

The main roles which over-actuation plays for increased safety can be divided into three different classes of safety functions:

1. Actuator redundance

2. Vehicle assists driver

3. Vehicle overrides driver

Over-actuation results in that one or more actuators being available to control a specific degree of freedom of the vehicle. In the case of an actuator failure, this means that the *actuator redundance* can be exploited by using the remaining actuator(s) to eliminate or mitigate the consequences of the failure. See [14] regarding the development fault-tolerant vehicles. In **Paper B**, **Paper C** and **Paper L**, actuator redundance is utilised to maintain vehicle stability during electrical faults in actuators in electric vehicles.

More actuators can also be used to assist the driver in his/her task of safely controlling the vehicle. Linguistically, the assistance obtained is expressed in terms such as ‘quicker’, ‘harder’, ‘of higher accuracy’ etc. One typical example is Antilock Braking System (ABS), which shorten the braking distance and allows simultaneous steering when the driver steps too hard on the brake pedal. Owing to the more precise longitudinal tyre slip control which ABS offers, the wheel is prevented from being locked. Another example is Electronic Power Assisted Steering (EPAS), which adds an aligning steering-wheel torque using an electric motor, and in turn, indirectly influences the steering-wheel angle applied by the driver.

The vehicle overriding the driver implies that actuation is commanded from the vehicle, independently of what the driver desires. As long as the level of threat is low, the vehicle is expected to follow the driver’s commands. However, when the passengers’ safety becomes severely threatened, the driver’s control authority can be questioned. In such circumstances, a vehicle controller can intentionally be designed to give commands which add to or contradict the driver’s commands.

It is, in many cases, difficult to distinguish strictly between assisting and overriding the driver. For example, ABS can be seen as assisting the driver to avoid locking the wheels, but if the driver really wants to lock the wheels, then ABS could be seen as overriding. This leads to the conclusion that the classification above probably is best applied to a certain safety design, including the particular safety problem/scenario to which the design applies.

Safety states

Vehicle functions involving additional actuation work differently depending on the level of threat. One structured way of studying factors that influence an accident before, during and after an accident, is the *Haddon matrix* [15]. Haddon was one of the pioneers of injury prevention and described a two-dimensional model for approaching injury and its causes during pre-event, event, and post-event. In order to clarify more distinctly the desired actions under the threat of a collision in the light of over-actuation, a state diagram is developed here and shown in Figure 2.4. This figure outlines the states and transitions that must be fulfilled before jump from one state to another. In particular, focus is put on pre-event rather than event and post-event. For each state, as explained below, there exists corresponding desired actions which can be supported by functions. Figure 2.4 shows a number of examples of functions.

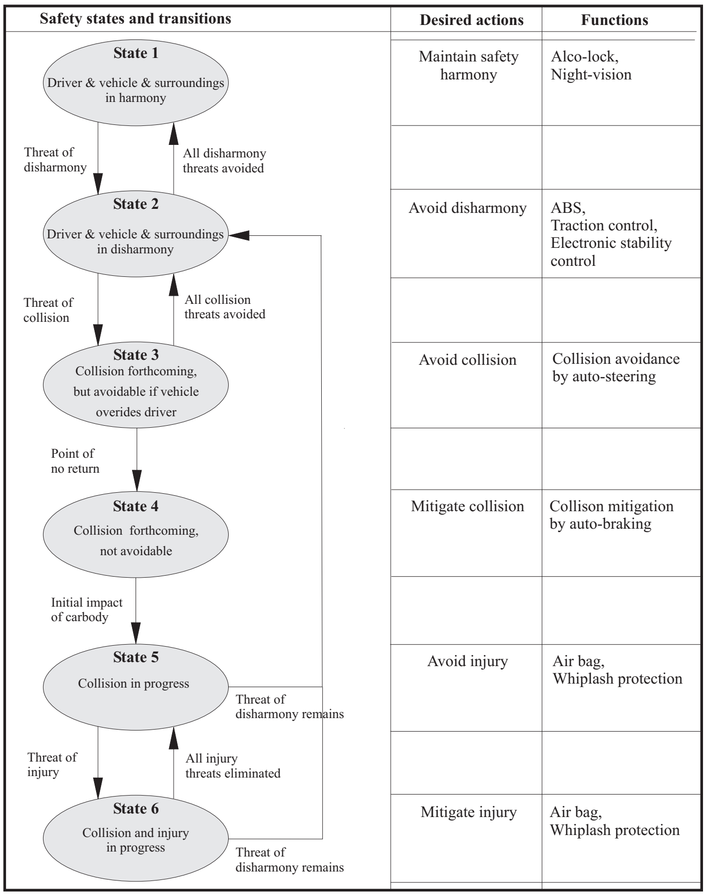

Figure 2.4: Safety states and transitions between them along with desired actions and examples of functions, which aims to transit upwards in the state diagram.

State 1

As long as the vehicle is not exposed to any threats, the driver, the vehicle and its surroundings are in harmony. Within State 1, any vehicle-controlled actions aim to preserve the harmony, and for an example, vehicle functions such as night-vision (display showing what is in front of the vehicle even when it is dark) can support the driver.

State 2
As soon as the harmony is interrupted, a transition is made from State 1 into State 2. Within State 2, the driver, the vehicle and its surroundings are in disharmony and there is a significant risk of a collision. However, the collision is still avoidable if the driver gives proper commands to the vehicle. In order to assist the driver to avoid threats more easily and to fall back to State 1, functions such as ABS and Electronic Stability Control (ESC) can be used. The implementation of those functions today involves actuation of individual wheel torques, but other types of interventions can also be used.

State 3
The driver cannot alone avoid the collision and no driver assistance functions are able to prevent the collision. However, the collision can still be avoided if the vehicle overrides the driver. One example of such a function is collision avoidance by auto-steering. This function is able to control vehicle cornering independently from driver commands.

State 4
The collision is impossible to avoid, but it can be mitigated. As soon as State 3 is entered, it is desirable to exploit actuators to mitigate the collision. An example is automatically actuated brakes, which are engaged independently of driver brake commands.

State 5
As soon as there is a carbody impact, State 5 is entered. In this situation, there are only two transitions from this state; to State 2 and to State 6.

State 6
Collision and injury in progress. In this situation, there is only one transition to leave; State 2.

The classical employment of over-actuation for higher safety is found with the presence of State 2, while State 3 has great future potential. Essentially, the research in this thesis covers those two states.

Finally, it is worth mentioning, over-actuation does also allow for influencing passive safety. By packaging the driveline at a distance from the front of the vehicle, the design of efficient deformation zones at the front can be facilitated.

2.2.3 Energy efficiency

The use of electric machines for motion control makes it possible to provide vehicle functions that traditionally have been achieved by complex and costly mechanical drivelines, linkages and servo assistance systems. An electrical powertrain can relatively simply and with limited extra cost be divided into several units and be distributed closely to the wheels. Thereby, a number of vehicle components, such as the transmission, may not be needed. For this reason, vehicle weight can be reduced.

Owing to a distributed powertrain, transmission losses can be kept low. In conventional vehicles, friction losses arise in drive joints, gears, clutches, bearings etc. This is particulary prominent in an All Axle Drive (AAD) vehicle, where the torque transfer to the rear axle generates high friction losses. In addition, the transmission adds weight, which in turn increases the need for energy due to higher rolling resistance. Masses and inertias also result in a higher energy consumption when not driving in a steady-state (assuming an non-ideal regenerative recovery of energy during e.g. braking). In a simulation study conducted on four wheel drive vehicles, it is concluded that the two main influential factors for increased consumption in an AAD vehicle are additional weight and decreased transmission efficiency, while the effect of the additional rotating masses is relatively small [16]. Furthermore, it is concluded that a transmission weight increase of 5% for a 1.800 kg vehicle can result in an average energy consumption increase of approximately 7.5%, with 3.1%-units of this increase being caused by the additional weight and 4.4%-units by transmission losses.

When a pure electric powertrain is applied, components such as exhaust pipes are not needed, and the vehicle admits the possibility of being designed to offer a lower air drag. In addition, when the vehicle is designed with levelling control capabilities, the clearance between the road and underbody can be kept at a minimum.

As soon as wheel alignments can be controlled to perfection, tyres can be developed to favour a low rolling resistance. One example is found in the ‘Carving car’ which was developed by Mercedes (see **Paper A**) where camber angle control enabled the development of a new tyre technology. For normal driving conditions, the tyre contact patch used a section which possessed a low rolling resistance. When high lateral acceleration was needed, another section specifically developed for high grip became quickly engaged by a change of the tyre camber angle.

The actuators used for vehicle motion control naturally have a significant role in vehicle energy consumption. Electrical machines have, in comparison with combustion engines, a relatively high energy efficiency. It varies depending on the speed and torque, but Permanent-Magnet Synchronous Machines (PMSM) intended for vehicle propulsion offer an efficiency of 60-95% [17] (the lower values being valid for lower revolution speeds). At higher motor speeds, the efficiency of PMSMs is in the range of 80-92% [18]. Apart from their high energy efficiency, electrical machines also allow kinetic energy to be recovered into electric energy during regenerative braking [19]. Studies of regenerative braking for in-wheel motor based chassis have been examined in [20, 21] in particular. When regenerative braking is engaged in over-actuated vehicles, brake forces may be distributed differently between the wheels. As an example, when the brake actuators' energy efficiency varies as a function of brake torque, the distribution among the brake actuator can be taken into account.

Chapter 3

Wheel corners

This chapter introduces wheel corner concepts and their characteristics. Particular attention is paid to the ACM technology, where functions of vehicle dynamics are software-controlled. Contributions of design proposals exploiting this opportunity are also discussed.

3.1 Design proposals

This thesis pays particular attention to a highly over-actuated vehicle equipped with Autonomous Corner Modules (ACM), a concept that was invented by Mr Sigvard Zetterström at VCC in 1998 [22]. The name *autonomous* indicates that the wheel forces and kinematics are individually controlled, supporting a common task. The design proposal, as illustrated in Figure 3.1, was invented primarily to offer:

(i) A chassis design that could be reused in the development process for new vehicle platforms.

(ii) Functions to be regulated by software.

(iii) Individual control of each wheel; propulsion/braking, steering/camber and vertical wheel load.

From these design goals, a chassis solution was found by equipping the vehicle with in-wheel hub motors combined with actuators for steering/camber and vertical wheel load [23]. For a literature review of findings associated with ACM technology, see **Paper A**.

Another similar concept is the *eCorner* [24, 25], as illustrated in Figure 3.2a, which is an integrated solution with an in-wheel motor and actuators on each wheel for steering and vertical wheel load combined with a wedge brake. However, wheel camber can not be controlled in this concept.

In addition, the concept car *HY-LIGHT*® must also be mentioned, which involves a solution where in-wheel motors are combined with individual control of the wheel suspension [26]. The vertical alignment of the wheel, provided by an electrical actuator, as seen in Figure 3.2b, is supported by a transverse control arm, which can be used to roll the car body as desired (not illustrated in Figure 3.2b). Wheel camber cannot be controlled in this concept and steering is made using an ordinary rack-and-pinion arrangement.

3.2 Concept characterisation

In order to characterise and compare the design proposals' abilities to move and control the wheel along different axes and directions, these concepts are presented in Table 3.1. Here, a conventional premium vehicle is added to Table 3.1 as a comparison. This vehicle is assumed to be four wheel driven and provided with a McPherson front strut and multilink rear axle.

For some concepts, motion is a result of actuators performing the main mechanical work in the close vicinity of the wheel. Here, this attribute is denoted *distributed actuation*. In some cases the degree of freedom of the wheel is denoted as *vehicle controlled* (denoted as V in Table 3.1), which indicates that the actuators' control inputs are composed of a combination of signals from other co-existing vehicle functions. One example to consider is vehicle-controlled steering, where signals from the vehicle and its environment can be combined with the steering wheel angle to create the steering actuator control input. Thus, the driver is assisted in his/her control task to follow the anticipated vehicle response. On the contrary, *driver controlled* steering (denoted as D in Table 3.1) is characterised by the use of steering wheel signal only to determine the control input. The use of vehicle-controlled functions embraces a variety of control structures, which have been classified in depth for vehicles in [27]. Table 3.1 also shows motion restrictions between wheels, which are present for some concepts.

Wheel camber is predetermined and bound by the suspension kinematics (denoted as B in Table 3.1) for the McPherson and the multilink suspensions.

Table 3.1: Degrees of freedom under control for front and rear wheels, V=Vehicle-controlled, D=Driver-controlled, I=Invariable with respect to car body coordinates, B=Bound by wheel kinematics

	ACM	*eCorner*	*HY-LIGHT®*	*Conventional*
	front/rear	front/rear	front/rear	front/rear
Drive and Brake	V/V	V/V	V/V	V[1]/V[1]
Steer	V/V	V/V	D/B	D/B
Camber	V/V	I/I	I/I	B/B
Wheel load	V/V	V/V	V/V	V[2]/V[2]

[1] Drive restrictions between wheels.

[2] Semi-active damping instead of active damping and levelling.

Thus, the resulting wheel camber depends on the wheel travel only. Suspensions, where camber is invariable for the entire wheel travel range with respect to car body coordinates, is denoted as *invariable* in Table 3.1. However, under the influence of lateral tyre forces, all these concepts influence the camber in different ways due to the suspension elastokinematics. This effect is normally not desired when handling the vehicle. However, if camber is allowed to be vehicle-controlled, this effect can be suppressed and camber can be set to fulfil the wanted performance.

The concepts' ability to control vertical loads in a force–velocity graph representation is often referred to as *active* damping or *semi-active* damping respectively [28]. The former represent a class of dampers that requires energy to be supplied directly to the actuator of the targeted actuation. Thus, the damper is able to generate forces in *all* four quadrants. In contrast, semi-active dampers do not require a power supply, except for the energy needed to alter the desired damper rates. Normally, this damper dissipates energy by converting it to heat. The semi-active damper is, due to the non-demanding energy consumption, restricted to operate in two quadrants in the force–velocity graph. *Passive dampers* are restricted to operate along predefined curves in two quadrants.

The three novel design proposals described here demonstrate three properties that are typical for this class of chassis configuration: vehicle controlled functions, few motion restrictions and distributed actuation. These design proposals also indicate a trend towards electric actuation. Nevertheless, functions can be developed using other power sources than the electric counterparts, e.g. hydraulic power. However, pure electric concepts are flexible for use in different vehicle configurations, such as; hybrid electric vehicles, electric vehicles and fuel-cell vehicles.

The transition from conventional vehicles to vehicles with distributed actuation, as illustrated in Figure 3.3, implies functions to be physically implemented at the vehicle corners. This, in turn, enables the development of a modularised chassis configuration. In [29], it is claimed that *modules* have the following characteristics that make them fundamentally different from corresponding groups of components in a subassembly:

(1) Modules are co-operative subsystems that form a product.

(2) Modules have their main functional interactions within rather than between modules.

(3) Modules have one or more well defined functions that can be tested in isolation from the system and are a composite of the components of the module.

(4) Modules are independent and self contained.

Vehicle-controlled functions facilitate a freer adoption of the transfer functions between driver and the actuators in question. In addition, mechanical connections can then be replaced by by-wire technology. Figure 3.3 illustrates the ACM vehicle, where all chassis functions are by-wire controlled.

The ACM concept is used for evaluation in a number of papers within this thesis. In **Paper E**, the feasibility of designing an electromechanical damper with an acceptable machine size is investigated. Moreover, the ACM is used for evaluation of fault handling in **Paper B**, **Paper C** and **Paper M**. In **Paper D**, the ACM force constraints are further developed, to involve vertical forces also, and used for controlling a vehicle. More information about ACM is also found in **Paper L**.

Since the ACM technology, as it is described above, uses novel chassis components and involves functions being completely vehicle-controlled, the development of the technology is challenging. Most likely, vehicles with less novel components and with a limited capacity for being vehicle-controlled will enter the market in the first phase. One example is vehicles with in-wheel motors. In the light of this evolution, **Paper H**, **Paper I** and **Paper J** also investigate AWD, RWD and AWB chassis solutions.

In 2006, VCC and KTH started a joint project together with MAGNA STEYR to develop a conceptual ACM design. The cooperation resulted in a design for a rolling chassis (see Figures 3.1b and 3.3b), with all ACM functions incorporated but partly built with existing chassis components [30]. The car was assumed to be four-seated with a curb weight of 1200 kg and a maximum vehicle speed of 150 km/h.

(a)

(b)

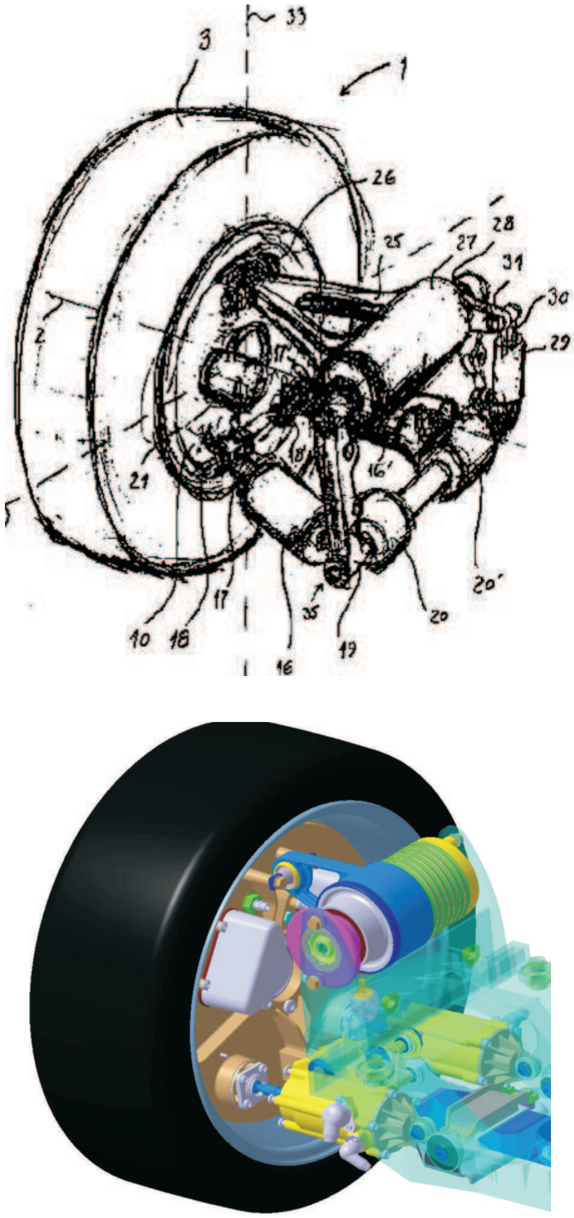

Figure 3.1: (a) The ACM disclosure from the patent application [22] and (b) The ACM concept further developed within this project in cooperation with MAGNA STEYR (illustration reproduced with permission of MAGNA STEYR Fahrzeugtechnik AG & Co KG, Engineering, Advanced Development Chassis).

(a) (b)

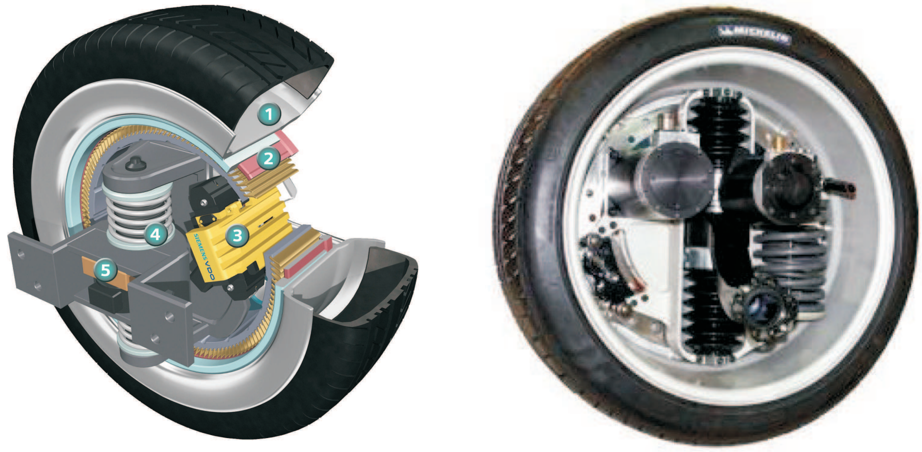

Figure 3.2: (a) The Siemens VDO *eCorner*: (1) wheel rim, (2) in-wheel motor, (3) electronic wedge brake, (4) active suspension, (5) electronic steering [25] and (b) Michelin's wheel unit *HY-LIGHT*® (illustrations reproduced with permission of Siemens VDO and Michelin respectively).

(a) (b)

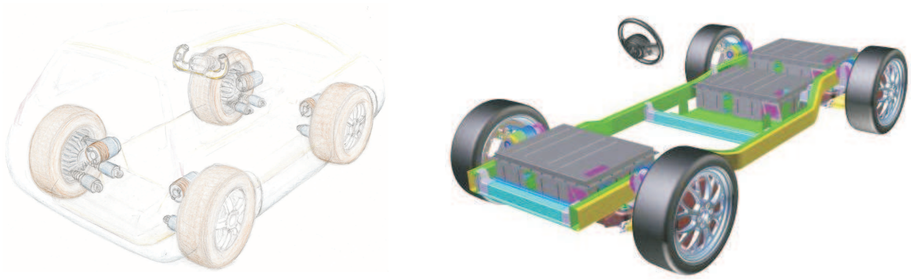

Figure 3.3: ACM vehicle with vehicle-controlled functions and distributed actuation (a) designed by Mr Sigvard Zetterström and (b) further developed by MAGNA STEYR.

Chapter 4

Configurations and constraints

This chapter formulates the constraints arising from tyres and actuators. The 'ultimate' vehicle is presented and also common force restrictions. In addition, this chapter also contributes a discussion concerning the force potential of over-actuation.

4.1 Constraints

Vehicles possess constraints of various types, which bound the attainable set of variables, such as force, position, angles and velocity. Constraints are also present at different levels in the vehicle topology. Some of them act between wheels, while others act within a wheel [31]. Front axle steering, for example, bounds the two wheel-to-road steering angles of the front wheels together, and consequently they cannot be individually controlled.

The tyre itself possesses significant force constraints. A commonly used assumption, see for example [32], is that the forces in longitudinal and lateral direction in the contact patch lie within a friction ellipse in an $f_x - f_y$ plane, where f_x and f_y denote the forces in the tyre's longitudinal and lateral direction respectively. Neglecting the wheel-spin around the tyre's vertical axis and denoting the force limits in the pure longitudinal and lateral directions $f_{x,\max}$ and $f_{y,\max}$ respectively, the friction ellipse is governed by

$$\frac{{f_x}^2}{{f_{x,\max}}^2} + \frac{{f_y}^2}{{f_{y,\max}}^2} = 1 \tag{4.1}$$

The well established MF-tyre model, introduced in [33] and further developed in [34, 35], describes these force limits empirically, stating that

$$f_{x,\max} = \mu f_z * (p_{\mathrm{Dx1}} - p_{\mathrm{Dx2}} \mathrm{d}f_z)(1 - p_{\mathrm{Dx3}} \gamma^2) \tag{4.2}$$

$$f_{y,\max} = \mu f_z * (p_{\mathrm{Dy1}} - p_{\mathrm{Dy2}} \mathrm{d}f_z)(1 - p_{\mathrm{Dy3}} \gamma^2) \tag{4.3}$$

where μ is the friction coefficient and f_z the wheel load. The relative change in the wheel load is governed by

$$\mathrm{d}f_z = \frac{f_z - f_z^{\mathrm{nom}}}{f_z^{\mathrm{nom}}} \tag{4.4}$$

where $p_{\mathrm{Dx1}}, p_{\mathrm{Dx2}}, p_{\mathrm{Dx3}}, p_{\mathrm{Dy1}}, p_{\mathrm{Dy2}}, p_{\mathrm{Dy3}}$ and f_z^{nom} are tyre parameters [35, 36].

The wheel load and the friction coefficient are the two most dominant variables influencing the friction ellipse. Moreover, the camber angle, γ, plays a role for the tyre constraints. For passenger cars, tyres are in general designed to be orientated with a 90 degree inclination.

If the wheel load is considered as a variable in Equation 4.1, the tyre force boundary can be depicted by a surface stretched in the x-y-z space forming an elliptic cone, and the cross-section along the z axis forms the friction ellipse (see **Paper D**). For an increasing wheel load, directed along the z-axis, larger forces in the horizontal plane are allowed to be generated. Ideally, the vehicle is able to utilise the tyres to their utmost limits. This means that, if the tyre forces are represented as vectors growing from the cone's tip, these forces are allowed to be generated inside four cones for a four wheeled vehicle, as illustrated in Figure 4.1. Consequently, it is not feasible to generate tyre forces outside these cones for a four wheeled vehicle.

4.1.1 Force restrictions

Ideally, the vehicle is able to generate tyre forces freely inside the four cones previously described. Under such an assumption, the tyres constitute the only force constraints. In practice, limitations in actuators set constraints on the tyre resources for force generation. As soon as these are present, the cones become restricted [37], which means either that the cones are shrunk or that there exist restrictions in tyre force generation in between them.

Vertical force restrictions

As explained already, the pure tyre force constraints form a cone (Figure 4.1), where an increasing wheel load in turn increases the tyre's capability to generate longitudinal and lateral forces. When there is a need for large tyre forces

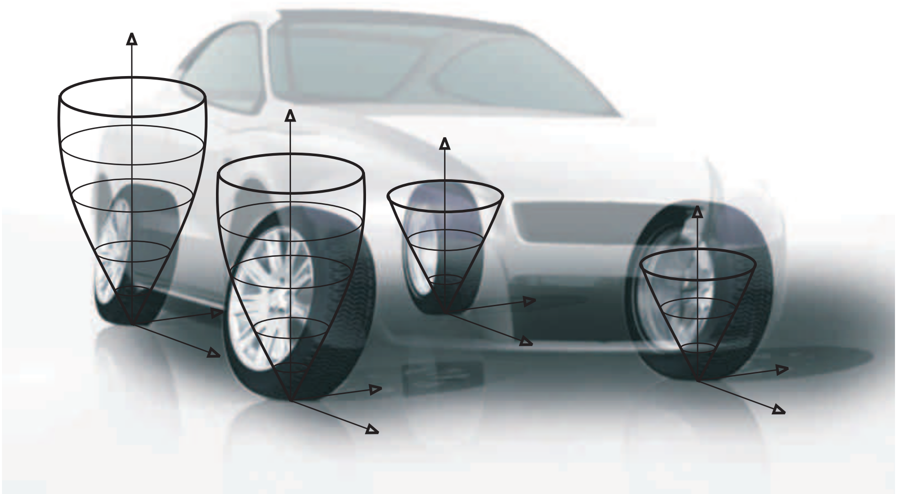

Figure 4.1: Pure tyre force constraints illustrated as cones.

in the horizontal plane, the general problem is to maintain high wheel loads for longer periods on all the wheels (without increasing the vehicle weight). Assuming the vehicle to be a rigid body, and denoting the four wheel loads $f_{z_1}, f_{z_2}, f_{z_3}$ and f_{z_4}, the force equilibrium around the vehicle's vertical axis and the torque equilibrium around the rolling and pitching axes in static conditions can be expressed as

$$\begin{bmatrix} 1 & 1 & 1 & 1 \\ w_l & -w_r & w_l & -w_r \\ -l_f & -l_f & l_r & l_r \end{bmatrix} \begin{bmatrix} f_{z_1} \\ f_{z_2} \\ f_{z_3} \\ f_{z_4} \end{bmatrix} + \begin{bmatrix} 0 & 0 & 0 \\ 0 & h & 0 \\ 0 & 0 & -h \end{bmatrix} \begin{bmatrix} 0 \\ F_y \\ F_x \end{bmatrix} = \begin{bmatrix} mg \\ 0 \\ 0 \end{bmatrix} \tag{4.5}$$

where g is the gravitational constant, m the vehicle mass and w_l, w_r, l_f and l_r represent vehicle geometry as shown in Figure 4.2a. The sum of the longitudinal tyre forces, F_x, and of the lateral tyre forces, F_y, result in pitch and yaw moment respectively due to the height above the ground, h, of the centre of gravity. Since there are four unknowns and three equations, the matrix Equation 4.5 is under-determined and the solution holds an infinite set of wheel loads [38]. However, as soon as the rigid body is equipped with wheel suspensions, the stiffnesses of them result in unique solution for the wheel loads for a given F_x and F_y. For conventional vehicles, the stiffnesses are fixed, since springs and dampers in general cannot be adjusted quickly. More advanced sus-

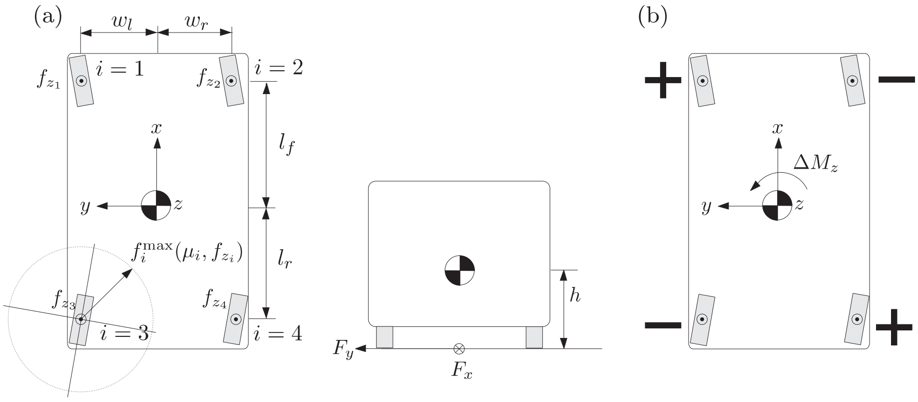

Figure 4.2: Distribution of wheel loads in static conditions. (a) Definition of vehicle geometry. (b) Allocation of wheel loads which results in an additional yaw torque ΔM_z.

pension designs, designated to improve comfort, admit the possibility of quick adjustment. From Equation 4.5, it is clear that wheel loads can be differently allocated along the vehicle diagonals [39], as illustrated in Figure 4.2b, without changing the equilibrium. This means that the extra freedom gained from the four wheels can be utilised to actuate wheel loads differently. Increasing the wheel loads at one of the vehicle diagonals brings a capacity for generating higher levels of horizontal tyre forces there and lower levels at the remaining diagonal. Diagonal actuation can also be used dynamically without directly affecting the pitch, yaw and bounce dynamics. Since lateral tyre forces increase non-linearly with respect to wheel loads, lateral axle forces can be manipulated by the diagonal allocation of wheel loads. This type of actuation is used in e.g. [40] to change rapidly the difference between the left and right wheel loads at the front or rear axle. Due to the tyre non-linearities, the axle side forces changes and an additional yaw torque is obtained. Here it is worth mentioning that an increasing wheel load leads to, at least in steady state, energy having to be provided. On the other hand, energy can be transferred from one wheel to another, which is the case with for example the torsion bar. The torsion bar provides stiffness between left and right wheel, mainly to reduce vehicle roll. Increasing roll stiffness also results in a larger portion of side load-transfer on that specific axle. As a result, the remaining axle experiences less side-transfer, which in turn changes the balance of the lateral front and rear axle forces. Hence, the torsion bar is widely used as a tuning element to adjust for a sufficient level of understeering.

Keeping the four cones in mind, these can be used dynamically at a high level. One or several wheels can for a short time be actuated with large wheel loads violating the force equilibrium in Equation 4.5. Thereby, the vehicle undertakes a vertical acceleration, roll angular acceleration or pitch angular acceleration.

Force restrictions in the road plane

In order to utilise the whole area in the friction ellipse for a given wheel load, friction coefficient and camber angle, the force generation in the tyre's longitudinal and lateral directions must be made with few restrictions. The slip's counterpart to force is governed by introducing tyre longitudinal slip, κ, and lateral slip , $\tan(\alpha)$, so that

$$\kappa = \frac{\omega r_w - v_{xw}}{v_{xw}} \tag{4.6}$$

$$\tan(\alpha) = \frac{v_{yw}}{v_{xw}} \tag{4.7}$$

where ω is the angular wheel speed, r_w corresponds to the wheel's effective rolling radius. v_{xw} and v_{yw} are the longitudinal and lateral wheel speeds in the wheel coordinates (see Figure 4.3).

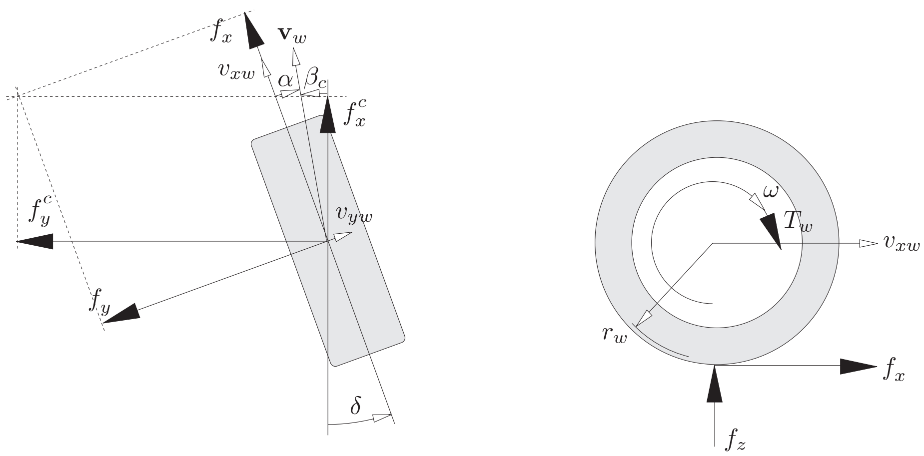

Figure 4.3: Definition of tyre and corner forces and velocities. See also the Nomenclature.

Longitudinal and lateral tyre forces are generated from longitudinal and lateral slip components respectively. The ideal tyre, which is able to attain all the force vectors (f_x, f_y) within the friction ellipse, must in turn be capable of generating longitudinal tyre slip, $\kappa \in (-\infty, \infty)$ %, and lateral tyre slip, $\tan(\alpha) \in (-\infty, \infty)$ rad.

The force generation, with respect to both κ and α, is non-linear and forces do not increase monotonically for higher levels of slip, see Figure 4.4. In addition, the force generation in the longitudinal and lateral direction is interconnected by the *combined slip*. In Figure 4.4 this becomes clear when the presence of combined slip reduces the force levels. For example, as soon as lateral slip is present in the $f_x - \kappa$ graph, κ needs to be adjusted to higher levels to counteract the reduction in f_x, compared to the case with no lateral slip. Eventually, a counteraction of tuning κ cannot fully compensate for the reduction in f_x.

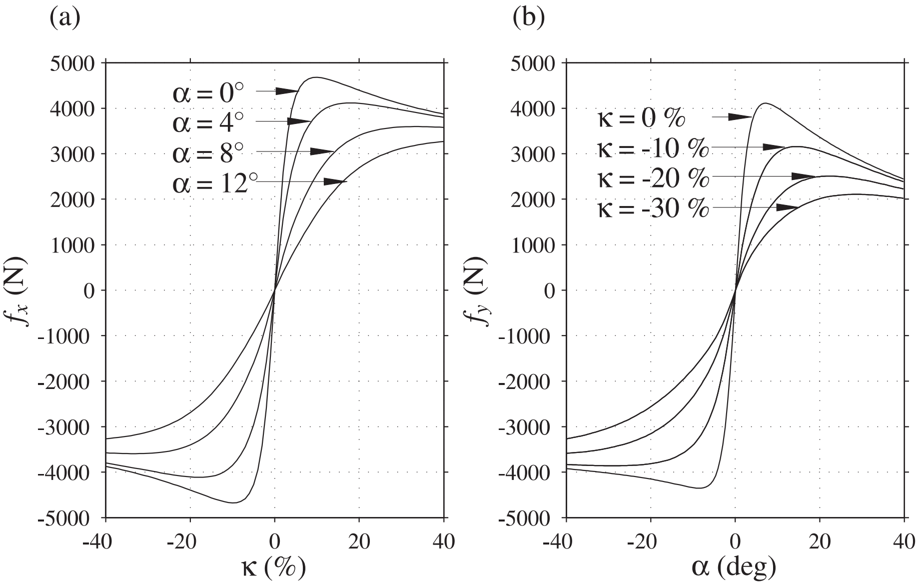

Figure 4.4: Tyre forces under combined slip for f_z=4000 N retrieved from a MF-tyre model. (a) Longitudinal force characteristics under the influence of four different side slip angles. (b) Lateral force characteristics under the influence of four different longitudinal slip.

The tyres' combined slip can also be presented in an $f_x - f_y$ graph with iso curves for a constant κ and α, see Figure 4.5. This graph is designed by sweeping one slip quantity while keeping the other constant. From the figure, it is evident that the combined slip characteristics have a significant impact on the tyre's force generation.

When wheel steering is used to generate lateral tyre forces, limitations in the wheel's steering angles result in limitations of possible lateral tyre forces.

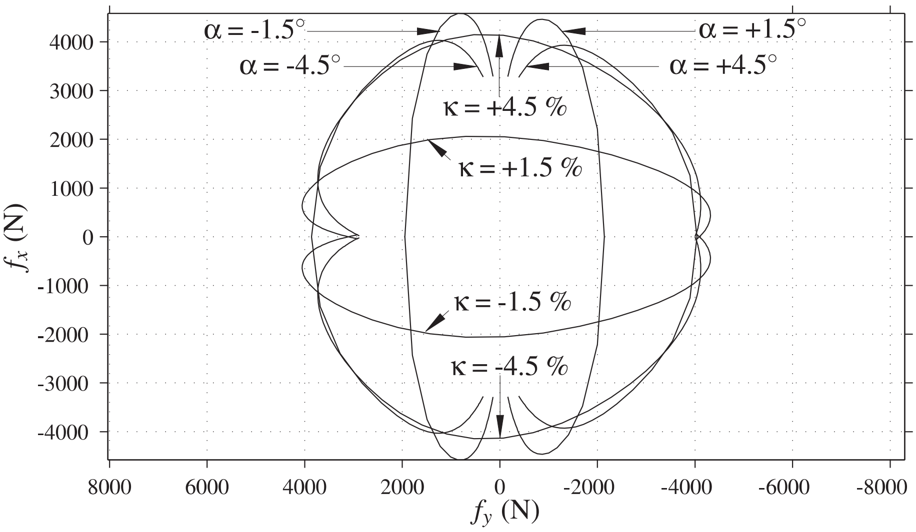

Figure 4.5: Tyre forces with iso curves for κ and α for f_z=4000 N retrieved from an MF-tyre model. κ is swept from -100→ +100 % (for each constant-α-curve) and α is swept from -0.5 → +0.5 rad (for each constant-κ-curve). When the sweeps are extended to even larger slips, the curves slowly approach origin.

Due to the combined slip characteristics, the tyre forces are captured inside two iso curves for constant lateral tyre slip, see Figure 4.6. Denoting the steering angle δ and wheel side slip β_c, their relations to the tyre side slip can be assumed to be expressed as

$$\alpha = \beta_c - \delta \tag{4.8}$$

$$\tan(\beta_c) = \frac{v_{yc}}{v_{xc}} \tag{4.9}$$

where v_{xc} and v_{yc} are the longitudinal and lateral velocities, respectively, for the corner of the vehicle in question in a vehicle fixed coordination system. Depending on β_c, the limitation in f_y can be unsymmetrical around $f_y = 0$, even for a symmetrical limitation of δ (which is the case for Figure 4.6b). Figure 4.6 shows the gradual reduction of the envelope of the maximum achievable tyre slips and forces. A state slip vector, marked as (κ^*, α^*), points out the current operational point. This vector has a unique corresponding force state vector denoted (f_x^*, f_y^*). In contrast, owing to the non-linear tyre characteristics, there is no guarantee of finding a unique image from a force vector (f_x^*, f_y^*) to a slip vector (κ^*, α^*).

As mentioned above, a limitation in δ results in both α and f_y of being

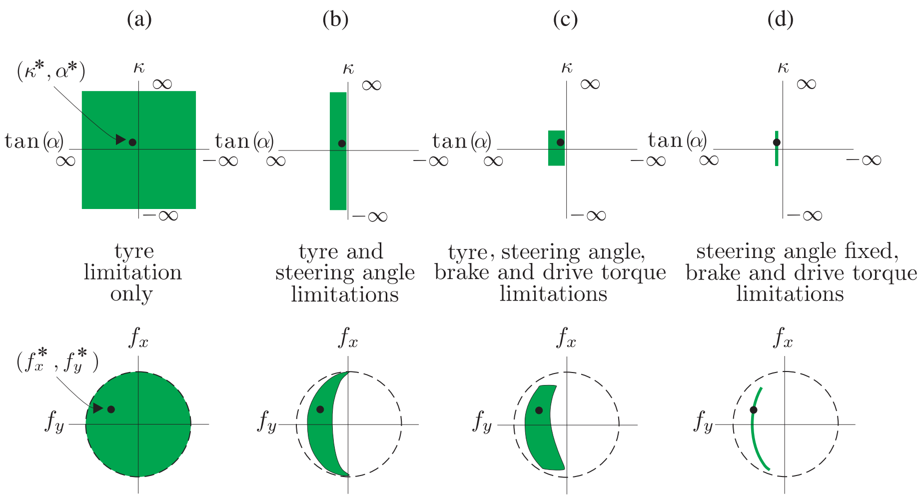

Figure 4.6: Conceptually composed tyre force constraints due to actuator limitations showing the gradual reduction of the envelope of the maximum achievable tyre slips and forces. Depending on the actuator restrictions, the region of attainable tyre forces (marked with green colour and referred to as the set S) varies.

kept in a closed interval. Since the longitudinal slip is still allowed to be freely actuated, $f_{x,\max}$ can still be reached. Under the influence of combined slip, f_y becomes weaker, which is seen in Figure 4.6b, where the area of attainable forces is curved.

As soon as a limitation becomes active for the generation of κ, the envelope of the maximum achievable force will change. If the actuator that controls the wheel's rotation were to acts as an ideal angular speed source, capable of generating longitudinal slip within a lower and upper limit, the tyre forces would be captured inside two iso curves for a constant κ. However, most actuators for rotating a wheel are primary torque-controlled with a lower and upper torque limit. For such wheel torque actuators, f_x is captured within a closed interval. Hence, a limitation in the steering angle combined with the wheel torques gives rise to a curtailed area, as illustrated in Figure 4.6c.

Finally, in Figure 4.6d, the steering is fixed and a closed interval of traction force is the only way to excite the tyre. When steering is changed slowly compared with traction/braking, this ellipse represents of all possible tyre forces. However, any underlying mathematical function describing the combined slip, such as the MF-tyre equations, is complex and can holds a high level of uncertainty.

For a part of the work in this thesis, the combined slip is approximated by an ellipse in the $f_x - f_y$ plane. Figure 4.7a shows combined slip measurements at two different side slips. In Figure 4.7b, simulated combined slip curves are presented using the MF-tyre model and also a simple elliptic approximation. As seen, the elliptic approximation is similar to the MF-tyre under low longitudinal slip. However, as soon as the peak in the $f_x - \kappa$ graph is passed, there is a significant deviation between them. Hence, the elliptic approximation has sufficient validity for the monotone part of the side slip characteristic only. In this region, the ellipse captures the essential effects of combined slip.

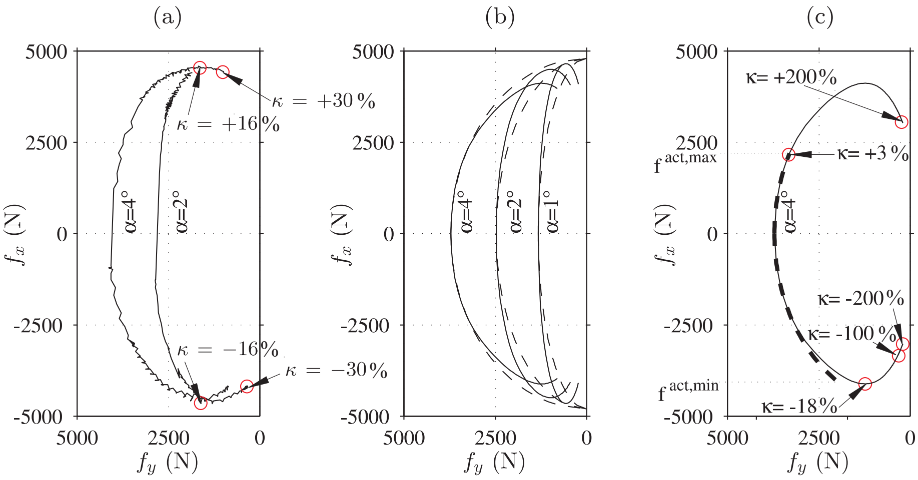

Figure 4.7: Combined tyre slip characteristics represented in an $f_x - f_y$ plane where the longitudinal tyre slip (κ) is continuously swept, while the tyre side slip angle (α) is kept constant. (a) Measurement results for two fixed side slip angles. (b) Simulation results from an MF tyre model (solid lines) and an elliptic approximation (dashed lines). (c) An MF tyre model (solid lines) and the elliptic approximation (bold dashed line), which also undergoes longitudinal force restrictions ($f^{\text{act,min}}$ and $f^{\text{act,max}}$) from a wheel torque actuator.

Interested readers are referred to **Paper H** and **Paper J** to find more information about the in-wheel constraints. Findings concerning restrictions present between wheels, such as differentials and axle steering are presented in [37].

4.2 Actuator dynamics

The maximum achievable tyre forces constitute a fundamental limit of force generation. Other aspects are the dynamics of the force generation. Dynamics are present in the actuators, transmission, wheels and tyres. Even for ideal actuators, the tyre dynamics caused by the tyre's relaxation length set fundamental limits to how quickly forces are generated. Tyre dynamics are present in both the longitudinal and the lateral direction and are substantially dependent on the tyre design and e.g. relation between the belt height and the rim width. Recommended publications about tyre relaxtion are [42, 43]. The time constant is approximately found by the quotient between relaxation length and the wheel's velocity, which indicates that tyre dynamics are dominant at lower vehicle speeds. High levels of slip and low wheel loads lower the time constant significantly. Driving on main roads, both the longitudinal and the lateral time constants from tyre dynamics are typically in the region of 0.05–0.10 s [42].

Combustion engines have relatively slow dynamics, arising from e.g. the dynamics of building a manifold air pressure and of bringing components with inertia to motion [44, 45]. The time constants for producing torque are in the region of 1 s.

Hydraulic friction brakes suffer from a time delay, of which a significant portion stems from the time required to move the brake pad to the brake disc. If the hydraulic brake system is not pre-charged, this time is approximately 0.3 s. This figure is however uncertain, depending on the condition of the brake system before it is intended for use. In addition, there are dynamics involved in building up a pressure between the pad and the disc depending on e.g. elasticity in hydraulic systems such as brake pipes. That time constant is in the region of 0.3 s.

Compared with the systems mentioned above, electrical actuators are significantly quicker. The time that it takes to produce torque or force is in principle a matter of how quickly an electromagnetic field is generated (which in turn depends on machine the inductance and the applied voltage). The electric actuator has a time constant in the region of 1 ms [46]. The electric actuator itself has a dynamic behaviour that in most cases can be ignored. However, transmissions used to transfer torque to the wheel often add dynamics due to the non-negligible elasticity and inertia of drive shafts. The typical time constant of a short drive shaft, and arisen from the shaft elasticity only, to a modern passenger car is in the region of 10 ms.

In **Paper J**, electrical machines are used as brakes by connecting the rotor to the wheel hub. Such brakes benefit from being quicker than a hydraulic friction brake system. Interestingly, electrical machines produce torque quicker than time the constants of the tyre, which is opposite case for the hydraulic friction brake system.

Even when quick actuators are used, the total time for sensing, decision making and actuation must be considered. For example, the communication between an Electronic Control Unit (ECU) and the actuator under control can also add a significant time delay.

4.2.1 Actuator rate constraints

Due to actuator rate constraints, the attainable tyre force region cannot be fully utilised in a short-term perspective. Considering an actuator with the force limits u_{min}, u_{max} and the rate limits r_{min}, r_{max}, the rate can be approximated as

$$r_{min} \leq \frac{u(t) - u(t-T)}{T} \leq r_{max} \tag{4.10}$$

where T is the system's sampling time. Accounting the actuator rate limits, the actuator forces are bounded such that [41]

$$\max(u_{min}, u(t-T) + Tr_{min}) \leq u(t) \leq \min(u_{max}, u(t-T) + Tr_{max}) \tag{4.11}$$

4.3 Global force potential of over-actuation

In Section 4.1.1, restrictions of force generation at the tyre's level have been presented. This section serves to present *global vehicle forces* and to explore their limits. The reason for dealing with global vehicle forces is firstly to give a better understanding of how different actuators support the vehicle. The definition of over-actuation that was given in Chapter 2 does not express how well all the actuators, as a whole, contribute to vehicle motion control. The well-established Milliken Moment Method (MMM-method) and Beta-method have been developed to investigate attainable global forces and moments for front steering input as well as vehicle side slip [47, 48]. However, as soon as the number of control inputs increases, the MMM-method and Beta-method become unsuitable. For this reason, there is a need for a method that can be used to illustrate global forces independently of the number of actuators.

Secondly, vehicle motion control, as it is used in **Paper B**, **Paper C**, **Paper D**, **Paper H**, **Paper I** and **Paper J** is enhanced if the attainable global forces are known. Thirdly, the division of forces in different force layers within the vehicle topology enhances the understanding by dividing a large complex control problem into several less complex control problems.

4.3.1 Global vehicle forces

The global vehicle forces and moments (from now on referred to as global vehicle forces) act on and around the vehicle's centre of gravity when the vehicle is considered as a rigid body. In practice, there are no real actuators which engage the global forces directly, which explains why global forces also are referred to as *generalised* or *virtual* forces. To cover a complete description of the rigid body, the global vehicle force vector, $\mathbf{f}^{\text{glob}} = [F_x\ F_y\ F_z\ M_x\ M_y\ M_z]^{\text{T}}$, includes three forces and three torques (see Figure 4.8, and **Paper D**). If pitch, roll and heave motions are neglected, the reduced global vehicle force vector, $\mathbf{f}^{\text{glob}} = [F_x\ F_y\ M_z]^{\text{T}}$, is found by a linear matrix multiplication of the tyre forces, $\mathbf{f} = [f_{x_1}\ f_{y_1}\ f_{x_2} \ldots f_{y_4}]^{\text{T}}$, according to

$$\mathbf{f}^{\text{glob}} = \mathbf{ATf}, \quad \mathbf{f} \in \text{S}, \tag{4.12}$$

The 8×8 transformation matrix $\mathbf{T}$ recalculates tyre forces in wheel coordinates to be valid in vehicle coordinates so that

$$\begin{bmatrix} f^c_{x_i} \\ f^c_{y_i} \end{bmatrix} = \begin{bmatrix} \cos(\delta_i) & -\sin(\delta_i) \\ \sin(\delta_i) & \cos(\delta_i) \end{bmatrix} \begin{bmatrix} f_{x_i} \\ f_{y_i} \end{bmatrix} = \mathbf{T}_i \begin{bmatrix} f_{x_i} \\ f_{y_i} \end{bmatrix}, \quad i = 1,2,3,4 \tag{4.13}$$

$$\mathbf{T} = \text{diag}(\mathbf{T}_1, \mathbf{T}_2, \mathbf{T}_3, \mathbf{T}_4) \tag{4.14}$$

where δ_i is the steering angle, $f^c_{x_i}$ and $f^c_{y_i}$ are the longitudinal and lateral *corner forces*, respectively, and f_{x_i} and f_{y_i} are the corresponding tyre forces for the i:th wheel as seen in Figure 4.8. The geometry matrix[1] follows

$$\mathbf{A} = \begin{bmatrix} 1 & 0 & 1 & 0 & 1 & 0 & 1 & 0 \\ 0 & 1 & 0 & 1 & 0 & 1 & 0 & 1 \\ -w_l & l_f & w_r & l_f & -w_l & -l_r & w_r & -l_r \end{bmatrix} \tag{4.15}$$

where w_l, l_f, l_f and l_r represent the vehicle geometry as shown in Figures 4.8 and 4.2. The set S is the time varying tyre force constraints for the vehicle configuration in question (see for example the green regions in Figure 4.6). The longitudinal, lateral and yaw accelerations of the vehicle in the horizontal plane are linked to the set of global forces according to

$$[a_x\ a_y\ \dot{\omega}_z]^{\text{T}} = [\text{diag}(m\ m\ J_z)]^{-1}\mathbf{f}^{\text{glob}} \tag{4.16}$$

where m is the vehicle mass and J_z represents the vehicle inertia around the vertical axis at the centre of gravity.

4.3.2 Attainable global vehicle forces

The attainable forces is the set, Ω, of all the possible forces acting on the vehicle. Ω is found by a linear matrix multiplication of all the possible tyre forces

[1] Also known as the control effectiveness matrix.

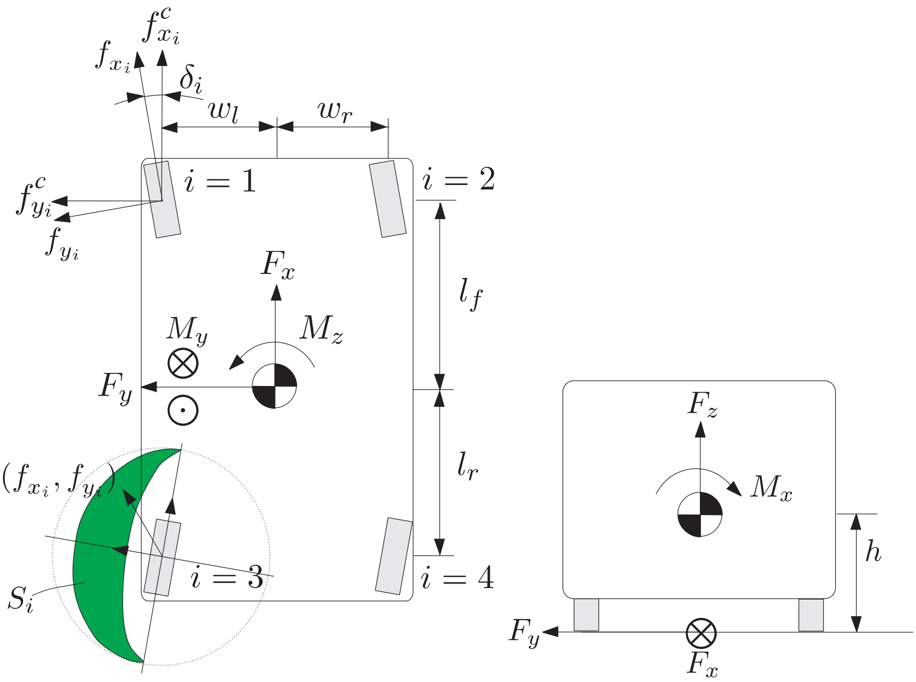

Figure 4.8: Definition of tyre forces, corner forces and global forces. The tyre force constraint valid for each tyre is denoted S_i.

according to

$$\Omega = \{ f^{\text{glob}} \in \mathbb{R}^m | \mathbf{ATf} = f^{\text{glob}}, \mathbf{f} \in \mathrm{S} \} \subset \mathbb{R}^m \tag{4.17}$$

When the vehicle's longitudinal, lateral and yaw global forces are considered, the mapping from a given set of attainable tyre forces in $\mathbb{R}^8$ results in a set of attainable global vehicle forces in $\mathbb{R}^3$. Note that the appearance of the Ω space depends to a great extent on the time-varying parameters that determine S. These parameters are dependent on vehicle states (such as the tyre side slip and the wheel load distribution), the road conditions (such as μ for the individual tyres, which can change quickly) and the constraints arising from actuators and tyres. In **Paper H**, a well-known method referred to as brute-force method is applied to visualise Ω for a number of important vehicle configurations. By mapping the estimated set of attainable tyre forces, as shown in Figure 4.9, according to Equation 4.17, Ω is presented in Figure 4.10 for the AWB and the AWD|S vehicle configurations respectively, at a specific global position during the single lane change. For the AWB vehicle configuration, it is assumed that the steering angles are held fixed and that the brake torques can be applied to reach the boundary of the friction ellipses. This means that the tyre forces are just valid along a part of the ellipse, with the bending of the ellipse depending on the tyre side force (see **Paper H** for details).

The plots of attainable global forces reveal actuators that do not complement each other. For example, if an actuator is added to the vehicle and it operates similarly to an existing one, the appearance of the plot will not change. In addition, owing to the analysis made on the global force level, the results do not rely on an implementation of a vehicle controller.

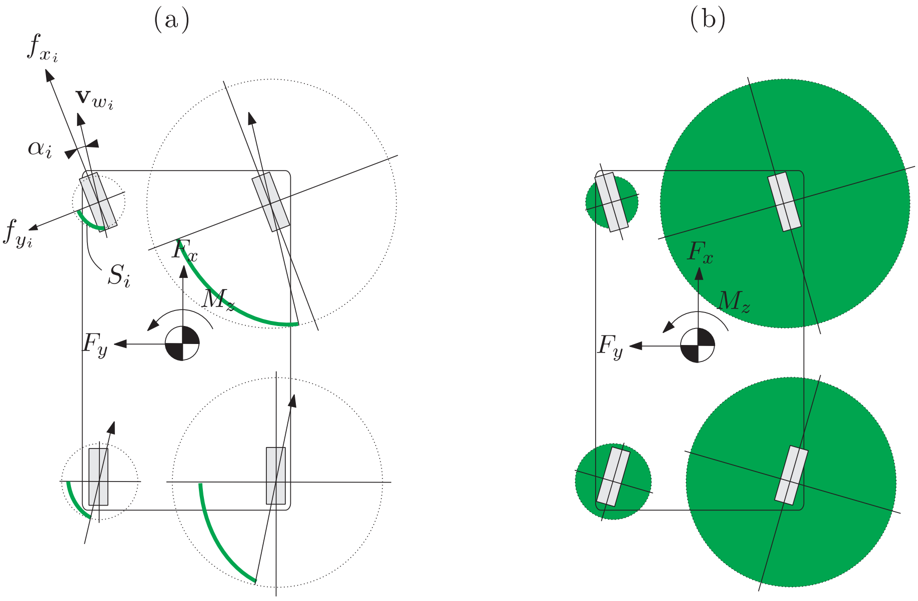

Figure 4.9: The principle behind the estimation of attainable tyre forces, marked with green colours, (a) for the AWB configuration and (b) the AWD|S configuration during left cornering. In (a), the wheel velocity vectors, $\mathbf{v}_{w_i}$, limit the brake forces to $\min(f_{x_i}) = f_{x_i}^{\max}\cos(\alpha_i)$, for the AWB configuration under the assumption that the angular velocities of the wheels, ω_{w_i}, do not become negative. Interested readers are referred to **Paper H**.

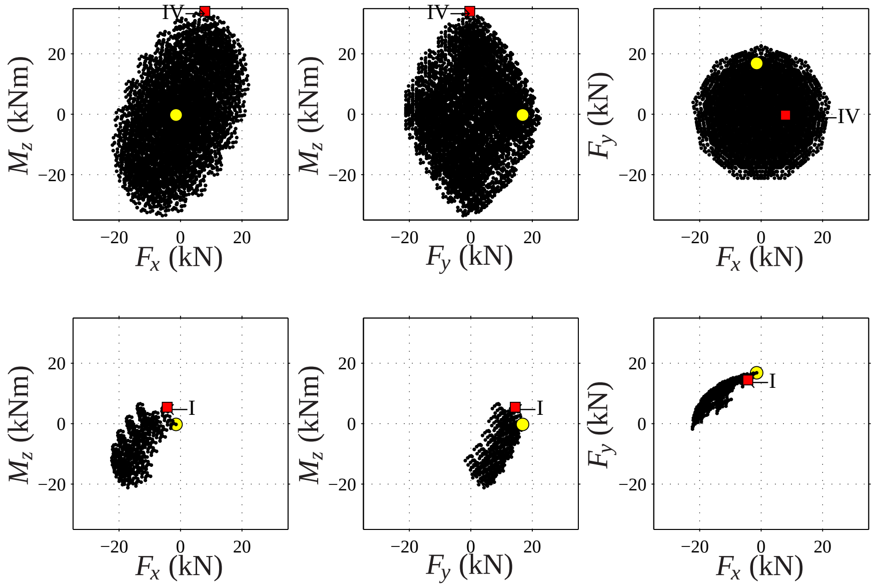

Figure 4.10: The space of attainable global vehicle forces, Ω, for the AWD|S configuration (first row) and the AWB configuration (second row) at a specific global position with actual tyre forces according to Figure 4.9. The yellow circle shows the value for the non-actuated vehicle. In the first row, the red square (IV) shows ‘full torque vectoring by drive and steer’. In the second row, the red square (I) shows the global forces obtained by ‘full inner rear brake’. Interested readers are referred to **Paper H**.

Chapter 5

Principles to evaluate over-actuation

This chapter formulates the control method used and further developed to evaluate the behavior of over-actuated vehicles.

5.1 Global chassis control

The control task to coordinate the vehicle's actuators in a structured way on a global level is referred to as *global chassis control* or *integrated chassis control*. An overview of developments and methodologies adopted within this area is found in e.g [27, 49]. The control methods used in this thesis are intended to control vehicle motion at a global level to utilise and coordinate all the available actuators within their force constraints to systematically maximise vehicle performance. The internal vehicle model used in the controller uses knowledge of the vehicle's most significant dynamics. In its simplest form is considered as a rigid body.

5.1.1 Control allocation

The motion of the rigid car body can be considered to be induced via global vehicle forces. Generally, these forces are virtual in the sense that each of them normally cannot be manipulated independently by a single actuator. As a consequence, global vehicle forces need to be allocated by an algorithm among the available actuators. The process described is referred to as *control allocation* or *force allocation* and has widely been used to control over-actuated aircraft [8, 50–52]. In addition, control allocation has been applied in other disciplines, such as the control of bio-mechanical systems [53] and marine vessels [54–56]. The control of over-actuated vehicles has traditionally been dealt with by using rule-based controllers such as ESC. Particulary during the past decade, the need for more systematic ways of dealing with the different possible configurations has been identified, and especially allocation by means of optimisation has been explored, in [37, 57–66]. One reason for this interest is that force allocation techniques admit the possibility of dividing the complex control task into two separate tasks. The first task is to establish the desired closed-loop dynamics and the second deals with the management of the actuator redundance caused by the over-actuation. Although the control task is divided into two tasks, the approach is found to be effective compared to solving the problem as one composite problem [41].

In Figure 5.1, the principal layout for the use of control allocation in this thesis. From a motion planning process, carried out by the driver and possibly supported by a vehicle controller, a reference trajectory, $\mathbf{x}_{\mathrm{ref}}$, of instantaneous states is generated. Next, a multi-input-multi-output dynamic motion controller minimises the tracking error by delivering global forces $\mathbf{f}_{\mathrm{ref}}^{\mathrm{glob}}$. These forces are in turn allocated, strongly restricted by tyre and actuator constraints, to all the four tyres of the non-linear vehicle. In the case of several actuators being present for controlling a specific degree of freedom, there may also be an actuator coordination process that follows.

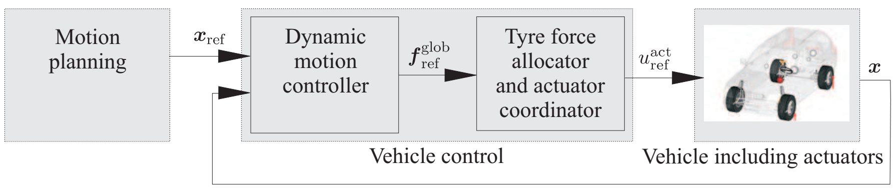

Figure 5.1: Simplified block scheme of control allocation adopted for an over-actuated vehicle. Typical variables used for controlling the desired vehicle states $\mathbf{x}_{\mathrm{ref}} = [v_x, v_y, \omega_z]_{\mathrm{ref}}^{\mathrm{T}}$ on a global level are $\mathbf{f}_{\mathrm{ref}}^{\mathrm{glob}} = [F_x, F_y, M_z]_{\mathrm{ref}}^{\mathrm{T}}$. After force allocation and coordination, these are re-formulated to a reference actuator control vector $\mathbf{u}_{\mathrm{ref}}$. When the actuators intended for motion control consist of steering actuators, electrical wheel torque actuators and friction brakes, $\mathbf{u}_{\mathrm{ref}}^{\mathrm{act}}$ is typically equal to $[\delta_1, \cdots, \delta_4, T_{e_1}, \cdots, T_{e_4}, T_{f_1}, \cdots, T_{f_4}]_{\mathrm{ref}}^{\mathrm{T}}$. After actuation, the vehicle's dynamic behaviour results in the actual vehicle state vector $\mathbf{x}$.

Control allocation together with over-actuation benefits from being *reconfigurable*. This means that the controlled system is capable of minimising losses during fault events without changing any higher level control laws. As soon as one actuator is degraded, the remaining actuators compensate for losses that otherwise could threaten the vehicle behaviour. The reconfigurable control ability also involves other actuator inefficiencies, such as actuator saturation and active rate constraints. In **Paper B** and **Paper C** the inherent capacity, owing to the reconfigurable control, of maintaining vehicle stability is shown.

When the desired control task is not feasible, control allocation provides a task prioritising. Commonly, desired motions along different degrees of freedom cannot be achieved simultaneously. This is highly relevant for road vehicles, where limit handling often involves a combination of translational and rotational motion. In **Paper I** and **Paper J** vehicles are exposed to unfeasible desired tasks, which are solved by prioritisation. The task prioritisation is one part of what is referred to as *fundamental allocator goals* (see **Paper I**).

In addition to prioritising among desired control tasks, control allocation also brings the ability of utilising actuators according to some preferences. In **Paper I**, these preferences are formulated in the form of *optional allocator goals*. One important example of such a goal is to keep the control energy at a minimum. The compromise to control energy buffers and driveline aiming to minimise the fuel consumption is presented in [58].

5.1.2 Model predictive control

One disadvantage of control allocation is that only the instantaneously desired states are used to control the vehicle. In many situations, the vehicle control is improved if prediction is also involved. Along with the development of sensors such as radars and cameras, prediction of the vehicle's future trajectory provides useful information for the controller. One structured state-of-the-art approach to handling hard constraints and the prediction of systems is referred to as *Model Predictive Control* (MPC). This approach was originally developed in the 1970s in the chemical industries for their need to predict the future actions of a process and then apply the optimal control signals for a desired behaviour [67]. MPC, which is becoming an increasingly popular approach as computational capacity increases, has been applied to aircraft as a natural extension to the standard control methods [68, 69]. For automotive usage, MPC is applied in e.g [70] for active steering and in [71] for brake torque distribution. In **Paper G**, MPC is employed to improve the vehicle stability by prediction of the future path. In contrast to control allocation, MPC admits to include actuator and tyre dynamics relatively easily in the model-based control design (which is applied in **Paper G**). However, one dilemma with a linear MPC solver is the necessity of involving the non-linear tyre constraints. Nevertheless, non-linear MPC methods exist, but their computational burden rapidly increases as soon as the control and prediction horizon increases. Therefore, for now, non-linear MPC is probably unpractical for implementation in vehicles.

To investigate the vehicle capability, independently of the implementation of a vehicle controller and along the path in global coordinates, dynamic optimisation can be used. See for example [72] where a dynamic optimisation methodology is illustrated with a comparison of rear axle steering and wheel torque distribution. One challenge with such a method is the determination of the initial and final vehicle conditions, which usually has a significant impact on the result.

5.2 Rigid body dynamics

Denoting the translational velocities $\mathbf{v} = [v_x, v_y, v_z]^{\mathrm{T}}$ and the rotational velocities $\omega = [\omega_x, \omega_y, \omega_z]^{\mathrm{T}}$ in a vehicle-fixed coordination system, the global vehicle forces (and torques) are governed by Newton's second law, so that

$$[F_x, F_y, F_z]^{\mathrm{T}} = m(\dot{\mathbf{v}} + \omega \times \mathbf{v}) \tag{5.1}$$

$$[M_x, M_y, M_z]^{\mathrm{T}} = \mathbf{J}\dot{\omega} + \omega \times \mathbf{J}\omega \tag{5.2}$$

where $\mathbf{J}$ is the inertia matrix for the rigid body. Assuming the non-diagonal elements in the inertia matrix to be negligible, $\mathbf{J} = \mathrm{diag}(J_x, J_y, J_z)$, which results in

$$\begin{bmatrix} F_x \\ F_y \\ F_z \\ M_x \\ M_y \\ M_z \end{bmatrix} = \begin{bmatrix} m\,(\dot{v}_x + \omega_y v_z - \omega_z v_y) \\ m(\dot{v}_y + \omega_z v_x - \omega_x v_z) \\ m(\dot{v}_z + \omega_x v_y - \omega_y v_x) \\ J_x\dot{\omega}_x + J_z\omega_y\omega_z - J_y\omega_z\omega_y \\ J_y\dot{\omega}_y + J_x\omega_z\omega_x - J_z\omega_z\omega_x \\ J_z\dot{\omega}_z + J_y\omega_x\omega_y - J_x\omega_y\omega_x \end{bmatrix} \tag{5.3}$$

In **Paper D**, all the six degrees of freedom are under control and the full set of global forces is used to control the vehicle. However, if roll, pitch and heave motions are excluded as controlled variables, ω_x, ω_y and v_z are commonly small numbers and the equations of the rigid body can be approximated as

$$\begin{bmatrix} F_x \\ F_y \\ M_z \end{bmatrix} \approx \begin{bmatrix} m\,(\dot{v}_x - \omega_z v_y) \\ m(\dot{v}_y + \omega_z v_x) \\ J_z\dot{\omega}_z \end{bmatrix} \tag{5.4}$$

Hence, the dynamics of the rigid body under planar motion can be re-expressed such that

$$\mathbf{M}\dot{\mathbf{x}} = \mathbf{h}(\mathbf{x}) + \mathbf{f}^{\mathrm{glob}} \tag{5.5}$$

where $\mathbf{x} = [v_x, v_y, \omega_z]^\mathrm{T}$ and $\mathbf{f}^{\mathrm{glob}} = [F_x, F_y, M_z]^\mathrm{T}$. The mass matrix, $\mathbf{M}$, coupling terms, $\mathbf{h}(\mathbf{x})$, are found as

$$\mathbf{M} = \begin{bmatrix} m & 0 & 0 \\ 0 & m & 0 \\ 0 & 0 & J_z \end{bmatrix}, \quad \mathbf{h}(\mathbf{x}) = m \begin{bmatrix} x_2 x_3 \\ -x_1 x_3 \\ 0 \end{bmatrix} \tag{5.6}$$

5.3 Corner and tyre forces

The global forces are related to the individual corner forces, as seen in Figure 5.2, via the augmented geometry matrix, $\mathbf{A}$, so that

$$[F_x \; F_y \; F_z \; M_x \; M_y \; M_z]^\mathrm{T} = \mathbf{A} \left[f^c_{x_1} \; f^c_{y_1} \; f^c_{z_1} \cdots f^c_{x_4} \; f^c_{y_4} \; f^c_{z_4} \right]^\mathrm{T} \tag{5.7}$$

where

$$\mathbf{A} = \begin{bmatrix} 1 & 0 & 0 & 1 & 0 & 0 & 1 & 0 & 0 & 1 & 0 & 0 \\ 0 & 1 & 0 & 0 & 1 & 0 & 0 & 1 & 0 & 0 & 1 & 0 \\ 0 & 0 & 1 & 0 & 0 & 1 & 0 & 0 & 1 & 0 & 0 & 1 \\ 0 & h & w_l & 0 & h & -w_r & 0 & h & w_l & 0 & h & -w_r \\ -h & 0 & -l_f & -h & 0 & -l_f & -h & 0 & l_r & -h & 0 & l_r \\ -w_l & l_f & 0 & w_r & l_f & 0 & -w_l & -l_r & 0 & w_r & -l_r & 0 \end{bmatrix} \tag{5.8}$$

At this stage, the vehicle is not provided with wheels. However, it is assumed that the vehicle can be excited with longitudinal and lateral forces at four specific corners. Observe that Equation 5.7 is under-determined and that the inverse of $\mathbf{A}$ does not exist. Hence, this unconstrained allocation problem holds an infinite set of solutions of corner forces.

Assume now that four wheels with tyres are attached to the corners, and that each wheel is steered with a steering angle δ_i. Under the assumption that the tyre's lateral force depends on the steering angle, the relation between the corner forces and the tyre forces can, as previously described, be expressed as follows

$$f_{x_i} = \cos(\delta_i) f^c_{x_i} + \sin(\delta_i) f^c_{y_i}, \quad f_{y_i}(\delta_i, f_{x_i}) = -\sin(\delta_i) f^c_{x_i} + \cos(\delta_i) f^c_{y_i} \tag{5.9}$$

From Equation 5.9, both f_{x_i} and δ_i can be calculated from $f^c_{x_i}$ and $f^c_{y_i}$. However, Equation 5.9 is non-linear with respect to $f_{y_i}(\delta_i, f_{x_i})$ and also due to the sinus and cosinus functions involved. One reasonable simplification is to consider

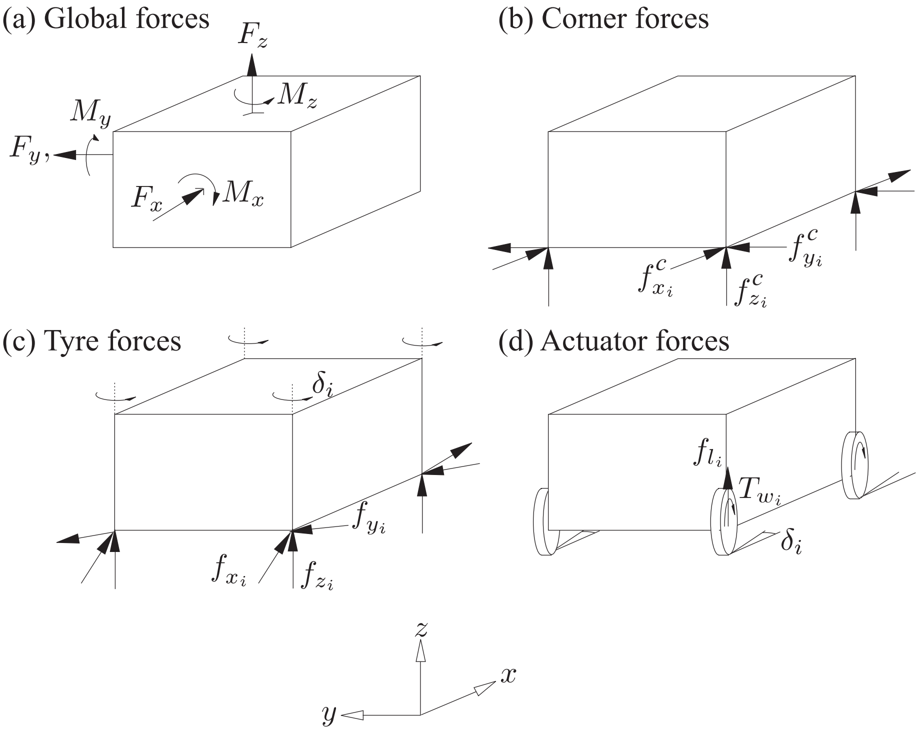

Figure 5.2: Allocation of (a) desired global vehicle forces to (b) corner forces, (c) tyre forces and (d) actuator quantities. In this example, the actuator quantities consist of wheel torques T_{w_i}, steering angles δ_i and vertical loads f_{l_i}.

the change of steering, $\delta_i(k) - \delta_i(k-1)$, to be negligible within a sampling time. Then the known steering angle $\delta_i(k-1)$ can be employed as a parameter in Equation 5.9, and in turn, it can be re-written so that

$$\begin{bmatrix} f_{x_i}(k) \\ f_{y_i}(k) \end{bmatrix} = \begin{bmatrix} \cos(\delta_i(k-1)) & \sin(\delta_i(k-1)) \\ -\sin(\delta_i(k-1)) & \cos(\delta_i(k-1)) \end{bmatrix} \begin{bmatrix} f^c_{x_i}(k) \\ f^c_{y_i}(k) \end{bmatrix} \tag{5.10}$$

Despite this simplification, $f_{y_i}(\delta_i)$ still remains as a non-monotone function. In general, this implies that for a given f_{y_i}, there exist two solutions of δ_i, one of which belongs to the degressive region of the tyre's force-side slip characteristics.

5.4 Force allocation to actuator quantities

Revisiting the rigid body dynamics, formulated in Equation 5.5, the global force control vector is governed by

$$\mathbf{f}^{\text{glob}} = \mathbf{A}\mathbf{f}^{c} = \mathbf{A}\mathbf{T}\mathbf{f} = \mathbf{A}\mathbf{T}\mathbf{g}(\mathbf{x},\mathbf{u}^{\text{act}}) \tag{5.11}$$

where the non-linear function $\mathbf{g}(\mathbf{x},\mathbf{u}^{\text{act}})$ represents tyre forces which depends on the state vector $\mathbf{x}$ and the actuator quantities $\mathbf{u}^{\text{act}}$.

When the actuators are assumed to work around an operation point $\mathbf{u}_0$, the function $\mathbf{g}(\mathbf{x},\mathbf{u}^{\text{act}})$ can be linearised such that

$$\mathbf{f}^{\text{glob}} = \mathbf{A}\mathbf{T}\mathbf{g}(\mathbf{x},\mathbf{u}^{\text{act}}) \approx \mathbf{A}\mathbf{T}\left(\mathbf{g}(\mathbf{x},\mathbf{u}_0^{\text{act}}) + \frac{\partial \mathbf{g}}{\partial \mathbf{u}}(\mathbf{x},\mathbf{u}_0^{\text{act}}) \cdot (\mathbf{u}^{\text{act}} - \mathbf{u}_0^{\text{act}})\right) \tag{5.12}$$

Let

$$\tilde{\mathbf{f}}^{\text{glob}} = \mathbf{f}^{\text{glob}} - \mathbf{A}\mathbf{T}\mathbf{g}(\mathbf{x},\mathbf{u}_0^{\text{act}}) + \mathbf{B}\mathbf{u}_0^{\text{act}}, \quad \mathbf{B} = \mathbf{A}\mathbf{T}\frac{\partial \mathbf{g}}{\partial \mathbf{u}}(\mathbf{x},\mathbf{u}_0^{\text{act}}) \tag{5.13}$$

Thereby, the allocation problem can now be re-written such that

$$\tilde{\mathbf{f}}^{\text{glob}} = \mathbf{B}\mathbf{u}^{\text{act}} \tag{5.14}$$

Assume that $\mathbf{u}^{\text{act}} \in \mathbb{R}^n$ and that $\tilde{\mathbf{f}}^{\text{glob}} \in \mathbb{R}^m$ constitutes the global forces intended for control. Let $\mathbf{B}$ be an $m \times n$ matrix and assume that Equation 5.14 describes the allocation process $\mathbb{R}^m \rightarrow \mathbb{R}^n$. Obviously, as soon as $n > m$ there are more actuator controls than global forces intended to be controlled. Thus the degree of over-actuation is here defined as $n - m$. Re-visiting the discussion in Chapter 2 as concerns under- equal- and over-actuation, the vehicle is under-actuated when $n - m < 0$, equal-actuated when $n - m = 0$ and over-actuated when $n - m > 0$.

Note that over-actuation does not guarantee that all global forces on the vehicle is controllable. Consider for example the planar motion of a vehicle with five identical torque actuators at one wheel. Since $n - m = 5 - 3 = 2 > 0$, the vehicle is over-actuated, but the three global forces are hardly coupled.

Equation 5.14 is solvable for $\mathbf{u}^{\text{act}}$ if rank($[\mathbf{B} \mid \tilde{\mathbf{f}}^{\text{glob}}]$)=rank($\mathbf{B}$). When any solution exists, the solution space has dimension n-rank($\mathbf{B}$).

5.5 Constrained allocation of tyre forces

Instead of allocating global forces directly to actuator quantities, such as torque and steering angles, allocation can be performed via an intermediate stage on the tyre force level. As soon as the wheel holds manifold actuators and allocation is performed directly to actuators as it is described in Equations 5.12 and 5.14, it is risk that potential solutions not are found [37].

Keeping in mind the non-linear tyre force constraints S discussed in Chapter 4, the tyre forces **f** (and hence corner forces) cannot be freely allocated as dictated in Equation 5.7 if force constraints not are to be violated. The non-linear constrained allocation problem formulated as

$$\begin{aligned} \mathbf{f}^{\text{glob}} - \mathbf{ATf} = \mathbf{0} \\ \mathbf{f} \in \mathrm{S} \end{aligned} \tag{5.15}$$

has infinitely many solutions, a finite set of solutions, or no solution at all. Researchers have used different methods to solve Equation 5.15. For example in [8] *direct control allocation* is employed and in [73] *daisy-chaining* is applied. Pseudo-inverse control allocation methods cannot handle constraints, but progress in developing this class of methods to handle constraints by iterations has been made [52, 74]. Non-linear programming has been employed for allocation in road vehicles, since it enables non-linear force constraints to be involved in the allocation formulation [37]. For reviews of control allocation methods, readers are referred to [51, 57].

Since the solution to Equation 5.15 may not exist, which typically is the case during limit handling, one alternative is to re-formulate it, so that

$$\begin{aligned} \min g(\mathbf{f}) = \min_{\mathbf{f}} \frac{1}{2} \left(\left\| \mathbf{WATf} - \mathbf{Wf}^{\text{glob}} \right\|_2^2 + \varepsilon \left\| \mathbf{f} - \mathbf{f}_0 \right\|_2^2 \right) \\ \text{subjected to} \quad \mathbf{f} \in \mathrm{S} \end{aligned} \tag{5.16}$$

where **W** is a weight matrix to prioritising among the global forces and **T** is a transformation matrix for adaptation from tyre to corner coordinates. The last term in the function g is added as an example of an optional allocator goal, to choose **f** close to the desired $\mathbf{f}_0$. The weight ε is normally a small number, used to prioritise the optional allocator goal. If necessary, more optional allocator goals can be added to Equation 5.16, such as the minimum actuator and tyre wear, the prevention of large roll angles etc. See **Paper I** for more information about fundamental and optional allocator goals.

As seen, the force allocation in Equation 5.16 is performed at the tyre level. As mentioned above, force allocation can also be made directly to the actuators, see for example [58]. The choice of the tyre level is, however, motivated throughout this thesis due to the high degree of over-actuation studied.

5.6 Actuator forces

If global vehicle forces are allocated to tyre forces, as depicted in Equation 5.16, these must be further coordinated to actuator quantities. One challenge that follows with this approach is to find inverse relations between tyre forces and actuator quantities with an appropriate level of accuracy. As an example, for a force reference pair (f_x, f_y), exact one corresponding (T, δ) is desired. As previously mentioned, both $f_y(\delta)$ and $f_x(\kappa)$ are non-monotone functions, which implies that their inverses do not exist. In this thesis, force allocation is employed for the stable region of the tyre. As soon as there is any intention to control tyre slip at or close to the unstable regions, it is suggested to allocate slip quantity instead of force.

For configurations where steering angles are controlled, δ_i must be calculated from a given f_{y_i}. In **Paper B**, a formulation of an inverse tyre model is suggested.

The relation between the wheel torque, T_{w_i}, and the longitudinal tyre force, f_{x_i}, is governed by the following equation (if rolling resistance is neglected)

$$J_{w_i}\dot{\omega}_i = T_{w_i} - r_w f_{x_i} \tag{5.17}$$

As soon as the wheel's angular acceleration is low, the wheel dynamics are assumed to be neglected and the desired wheel torque is then found as

$$T_{w_i} = f_{x_i} r_w \tag{5.18}$$

If the longitudinal tyre slip is high (macro slip has started), wheel dynamics often constitute a dominant part. Hence, it is suitable, along with control allocation, to control ω_i as a closed loop. Then the control allocation would generate a reference ω_i instead of a reference T_{w_i}. If T_{w_i} is used only as a reference under high slip, there is a risk that the wheel will lock. However, if combined slip is included in the force allocation constraints, the effect of a locked wheel on the vehicle level is limited. The reason is that the allocator take account of and accepts a weak side force under large longitudinal forces.

When regenerative brakes are used together with frictions brakes, T_{w_i} must be coordinated (brake blending) between the two categories of wheel torque actuators. See for example in **Paper C** for an implementation of brake blending.

5.7 Evaluation criteria

Criteria used to evaluate the behavior of differently actuated vehicles in this thesis are shown below.

5.7.1 Dynamic safety during emergency handling

Deviation from intended states

In **Paper B** and **Paper C** controlled and uncontrolled vehicles were compared with respect to how well they recovered from deviations from reference trajectories. In **Paper J**, vehicles were monitored if they drove over cones or if they underwent roll-over (a restricted comfortable roll motion can be seen as part of a desired trajectory). A driver experience the vehicle unstable as soon as the state deviation is considered unacceptable. In severe cases, unstable means that a small perturbation put the vehicle into a condition where the deviation from the intended states grows. In such situation, the driver has usually lost his/her control authority.

Over-shoots

The presence of over-shoots during side motions could threaten the vehicle (since it can crash with other obstacles). Over-shooting means passing outside and then return to the intended path with respect to the road's system of coordinates. In **Paper G**, the over-shoots were part of the evaluation criteria when comparing different prediction horizons.

Maximum entry speed

To evaluate the ability of a vehicle to undertake an evasive manoeuvre without losing control, one could establish the maximum side displacement for a given vehicle speed. Another method is to establish the maximum entry or exit speed to perform an evasive manoeuvre for a given fixed displacement. In **Paper J**, the maximum entry speed is used for evaluation. One explanation for this selection is that a higher entry speed implies that more vehicles can avoid crashes (since a larger speed interval, statistically in real-life traffic, captures more accidents). The accident scenario especially in mind is that where the host vehicle risks a collision between its front and the rear-end of a vehicle ahead, or risks losing lateral control when evasively overtaking the vehicle ahead.

5.7.2 Comfort during driving on uneven roads

Comfort

Comfort is strongly related to the accelerations of the vehicle body. Therefore the evaluation criteria for comfort entails the formulation of keeping accelerations at a minimum. In **Paper E**, the vertical acceleration of the sprung vehicle mass is used as a performance index.

Road holding

It is preferable to achieve as little variation in the wheel load as possible. Firstly, large variations in vertical tyre contact patch forces jeopardise the tyres ability to generate tyre forces in the horizontal plane. Secondly, variations in the wheel loads induce forces, via the unsprung wheel mass, to the vehicle sprung mass. See **Paper E** for definitions of cost functions for evaluating wheel suspensions.

5.7.3 General factors for any vehicle attribute and manoeuvre

Control energy

The impact of the control energy caused by control allocation, defined as the L_2-norm of the longitudinal and lateral tyre forces, was studied in **Paper I**. Furthermore, in **Paper E** the power dissipated/regenerated in the electromechanical suspension studied was analysed.

Friction utilisation

By defining the friction utilisation, η_i, as in [38] and in **Paper D**:

$$\eta_i = \sqrt{\frac{(f_{x_i})^2}{\left(f_{x_i}^{\max}\right)^2} + \frac{(f_{y_i})^2}{\left(f_{y_i}^{\max}\right)^2}} \tag{5.19}$$

the margins for even tougher tasks are indicated. When $\eta_1 =, \cdots, = \eta_4 = 1$, tyre forces are saturated and no margins exist. Alternative formulations of Equation 5.19 have been discussed. The tyre force margins can also be weighted unequally between the tyres and between the longitudinal and lateral terms for each tyre. For example, [75] suggests valuing lateral margin on the rear axle higher, which turns this criterion into a safety evaluation criterion. In **Paper D**, the friction utilisation is investigated for an ACM vehicle and a conventional front steered vehicle.

Chapter 6

Modelling and validation

This chapter presents a modelling environment to evaluate different types of wheel actuation. Vehicle and driver models are presented. The validation of the vehicle model is included. Additionally, a methodology that admits the possibility of increasing the numbers of actuators in a validated vehicle model is also described.

6.1 Physical vehicle vs. vehicle model

The rigid body that has been employed as a simple description of the vehicle's dynamic behaviour in Chapter 5, is used as an internal model in the vehicle's controller. In general, there is a great deviation between the behavior of the physical vehicle and the model counterpart. The real physical vehicle is strongly non-linear, certainly under fast transients and strong lateral accelerations. In addition, there is a large portion of model uncertainties involved, including parametric uncertainties concerning the tyres and the vehicle and subsystems. In this context, validation is defined as ascertaining how well a virtual vehicle model is able to imitate the real physical counterpart. Through such imitation the vehicle model and the real physical vehicle should both respond equally if they are exposed to the same inputs. Note that both systems under consideration are multi-input-multi-output systems where typical inputs and outputs are the steering wheel angle (in), the wheel torques (in) and the vehicle translation and rotation velocities (out).

Obviously, the real physical vehicle can be successfully used as a test object. However, vehicle modelling becomes a viable option mainly due to the following:

- Repeatability, for example independence of unpredictable weather conditions (such as friction and wind).
- Avoidance of dangerous tests that would threaten the driver's safety.
- Easy re-configuration of subsystems and their parameters.

6.2 Vehicle model validation

In **Paper B**, **Paper C**, **Paper D** and **Paper G** extensions of the well-established bicycle/single track model [76] are employed to cover the non-linear tyre force characteristics as well as to take account of additional states, e.g. vehicle's roll and pitch dynamics. However, as soon as validity is required for a broad range of manoeuvres, a higher level of model details is required. Failure to find a proper level of detail with the corresponding parameterisation may cause the simulation to deviate from the expected scenario, especially under limit handling situations. Therefore, in order to obtain a validated vehicle model suitable for control in such situations, the validated vehicle model utilised within the framework of this thesis was developed externally by Modelon AB, Sweden (see Figure 6.1b).

In order to control, test and compare differently actuated vehicles (**Paper J**), the work on model the vehicles starts with modelling an existing conventional vehicle. The vehicle can then be tested and its behaviour can be compared to the simulated model behaviour. Since it was initially established that the level of model uncertainties must be kept low, even under limit handling, great efforts were made to develop the vehicle model with a proper level of detail. In addition, a number of measurements on the vehicle and its subsystems were suggested and made to lessen the parametric uncertainties by identifying model parameters (**Paper M**).

The developed vehicle model holds 6 degrees of freedom for the car body, 4+4 degrees of freedom concerning the wheel rotations and steering, 4x6 degrees of freedom for the wheel hub compliance and 4 degrees of freedom for the wheel travel, see **Paper M** for details. A strong principle running throughout the work has been component-based design where physically based parameterisation is performed on subsystem levels, i.e. tuning of the final vehicle models has not been permitted.

Before the vehicle model was ready for being equipped with different actuators, the vehicle model passed through a validation process (**Paper F**). In this process, the vehicle model was exposed to different steady-state and transient driving situations. Simultaneously, a physical vehicle (see Figure 6.1a) was

(a) (b)

Figure 6.1: (a) Handling measurements carried out with an instrumented vehicle. (b) vehicle model developed in Dymola.

instrumented and exposed to the same driving conditions. By comparing the vehicle state variables gained from model simulations and real-word measurements respectively, a sufficient level of validity was finally established.

This thesis has directed particular focus on the use of validated vehicle models as plants for simulations. There are two main reasons for this: firstly, the driving conditions studied are close to the limits of vehicle stability, and secondly, there is a need for absolute measures of performance.

6.3 Methodology for actuator extensions

Starting from the validated vehicle model, a topology of differently equipped vehicles was developed in a Dymola/Vehicle Dynamics Library[1], see **Paper J**. Owing to the model hierarchy adopted, actuators were easily exchanged to form a number of differently actuated vehicles. Figure 6.2 illustrates the methodology developed to investigate differently actuated vehicles including modelling and validation.

To be able to emulate the different vehicle configurations, the developed vehicle model was provided with a flexible interface to the vehicle motion controller. The following vehicle model inputs were present; Individual wheel torques from electrical machines and friction brakes respectively, a steering-wheel angle and individually super-positioned steer angles.

[1]Dymola is registered trademark of Dynasim AB, Lund, Sweden.

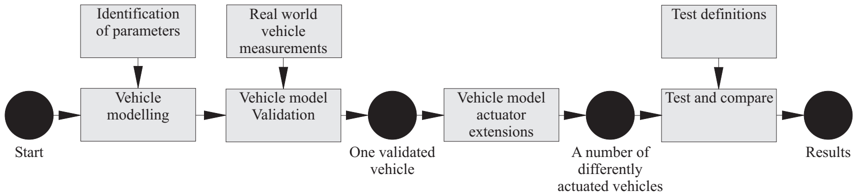

Figure 6.2: The methodology developed to investigate differently actuated vehicles.

6.4 Motion planning

In **Paper B**, **Paper C**, **Paper D** and **Paper G** vehicles are controlled under the assumption that they should follow a pre-defined state-trajectory. Even when the actuators are optimally used, there is still a question mark over the selection of the state-trajectory used. A pre-defined trajectory can be interpreted either as the the driver's desire or the result of automated motion planning performed by the vehicle itself. In **Paper B**, **Paper C** and **Paper D**, vehicles are forced to follow a state-trajectory of instant motion variables ($[v_x, v_y, \omega_z]^T$) and in **Paper G**, the state-trajectory also involves the full path ($[X, Y, \theta]^T$) defined in the road's inertial coordinates. The latter trajectory definition benefits from control decisions that in the short term are not guaranteed to be effective, but certainly are better in the long run.

To emulate the complex interplay between the driver, the vehicle and its surroundings, without defining the state-trajectory of instant motion variables in advance, an evaluation environment is suggested (**Paper J**). In that paper, a driver model executes the motion planning and delivers steering wheel angles, which in turn, are interpreted by a reference model to give desired states. The selected driver model is a standard model from Dymola/Vehicle Dynamics Library and is based on the point-following method, see for example [77]. It is assumed that that the driver plans the vehicle's motion by turning the steering wheel so that the front wheels point towards the driver's intended path at a given distance ahead.

6.5 Scaled prototype of an ACM vehicle

As an alternative to mathematical modelling as described above, the building of a real vehicle which is simpler to handle and cheaper to develop is worth considering. A project that recently started at KTH Vehicle Dynamics, 'Generic vehicle motion modelling and control for enhanced driving dynamics and energy management', aims to control a down-scaled prototype of the ACM described in Chapter 3. The built prototype, scaled to 1:6, is equipped with actuators to provide individually actuated wheel torques, steering angles and vertical loads. For practical reasons, wheel suspension and actuator layout is not exactly as depicted in Chapter 3. However, at a functional level, they are equal. All the actuators are electrical motors, powered by a battery and controlled by an on-board computer. Figure 6.3 shows the down-scaled prototype, which has also been virtually reproduced in Dymola, see **Paper O**. See also **Paper O** for more insight in the design of the vehicle and the selection of electrical actuators and sensors used to provide all ACM functions. Up to the stage of the writing of this thesis, the production of the down-scaled prototype is not fully completed. Consequently, all ACM functions are not tested. The most important contribution the down-scaled prototype brings, is the opportunity to validate force allocation in real time with the use of non-ideal sensors.

(a) (b)

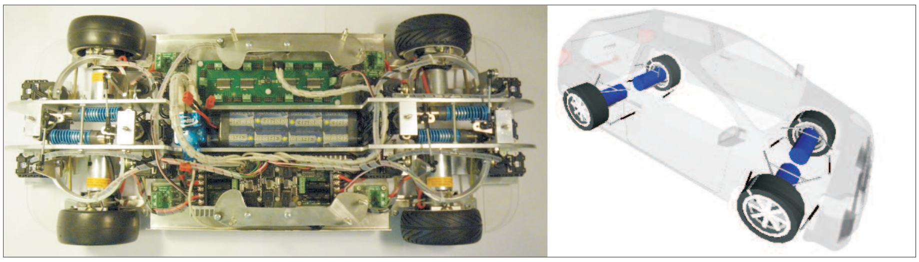

Figure 6.3: (a) A down-scaled physical prototype of an ACM vehicle (b) and its corresponding Dymola model (**Paper O**).

Chapter 7

Summary of appended papers

This chapter gives a brief summary of all the appended papers and explains how they are linked together.

Paper A: "Autonomous corner modules as an enabler for new vehicle chassis solutions"

This paper describes and reviews associated findings within the research field of this thesis. Research findings indicate that the Autonomous Corner Module (ACM) is capable of covering an extensive range of "stand-alone" functions which have been used for conventional vehicles to achieve benefits in vehicle dynamics. Associated findings suggest methods for chassis control, where tyre forces can be allocated from a vehicle trajectory description. In addition, these findings indicate that the ACM introduces new opportunities and shows itself to be a promising enabler for vehicle dynamic functions.

Paper B: "Stability of an electric vehicle with permanent magnet in-wheel motors during electrical faults"

This paper presents an analysis of the stability of an electric vehicle equipped with in-wheel motors and individual steering actuators. The vehicle stability has been evaluated by simulations when faults have been introduced, arising from a 30 kW electric permanent-magnet synchronous-machine. It is shown that the electrical fault risks causing a major decline in vehicle stability if the correct counteraction is not quickly undertaken. However, by introducing a motion controller combined with a rule-based procedure for tyre force allocation, it is shown that stability can be maintained. This inherent capacity to handle faults is attractive, especially since less additional case-specific fault-handling strategies or hardware are needed.

Paper C: "Control of electric vehicles with autonomous corner modules: implementation aspects and fault handling"

In this paper, the ACM is studied when adopting the procedures of tyre force allocation by optimisation, when a linear description of tyre and actuator constraints being formulated and used in the allocation process. In order to evaluate the proposed vehicle control principle from **Paper B**, the ACM vehicle is exposed to realistic fault conditions. If these conditions occur during extreme driving scenarios, the motion controller can hardly maintain stability. However, if the constraints in the optimisation procedure used for tyre force allocation are adapted to the specific fault, it is shown that stability can be secured. The paper also demonstrates how a limited computational capacity used in the optimisation solver can result in unwanted interactions between the individual actuators, and thus, indirectly affect vehicle stability.

Paper D: "Exploiting autonomous corner modules to resolve force constraints in the tyre contact patch"

In this paper, a vehicle exploiting ACMs has been forced to follow a trajectory identical to a conventional front-steered vehicle trajectory during an evasive manoeuvre. In addition to the allocation technique developed in **Paper C**, the vehicle's roll, pitch and heave motions are also under control. This is achived by the introduction of vertical force actuators which are placed at all four corners and are also involved in the force allocation process. The paper aims to identify the tyre forces needed and to analyse how they deviate from the tyre forces in a conventional front-steered vehicle. A suggested approach for the control of steering actuators is presented. Finally, the force allocation strategy involves the ability to control the vehicle slip independently from the vehicle yaw rate, which is used to increase the adhesion potential.

Paper E: "Design and evaluation of an active electromechanical wheel suspension system"

This paper presents a design proposal for an electromechanical wheel suspension which is intended to be a part of an ACM. The electromechanical wheel suspension is allowed to work in a fully active mode. The control structure laid out enables each wheel suspension to be controlled locally without burdening the global vehicle controller developed in **Paper D**. The complex design task involving both the control of the electric damper and its design is tackled by genetic optimisation. During the design, design and control parameters are optimised to keep the power dissipation of the electric damper as low as possible, while maintaining acceptable comfort and road-holding capabilities. The results of the evaluations carried out demonstrate that the proposed suspension can easily adapt its control parameters to obtain a better compromise of performance than that offered by passive suspensions. The size of the electric damper with the suggested control structure, including how it regenerates energy, is also discussed.

Paper F: "Modelling and parameterisation of a vehicle for validity under limit handling"

The vehicle models employed for simulations in **Paper B**, **Paper C** and **Paper D** are relatively simple. This paper presents the development of vehicle and subsystem simulation models, valid also for transient maneuvres during high lateral acceleration. Component-based modelling with validated subsystem behaviour is used to ensure the validity of the model. This is attained through experiments and measurements using torque measuring wheels and a gyro-platform. Special attention is given to an investigation of how a change of parameters affects the simulation results for various manoeuvres.

Paper G: "Using future path information for improving stability of an overactuated vehicle"

In this paper, model predictive control is applied for controlling an over-actuated vehicle. The use of model predictive control is shown to be a suitable method for distributing the tyre forces if the vehicle's future trajectory is known (in **Paper B**, **Paper C** and **Paper D**, the vehicle was controlled instantaneously without prediction). Simulation studies that are conducted show that access to information in advance, even if such information is restricted to be less than 1 s, significantly contributes to maintain vehicle stability. Furthermore, a longer prediction horizon results in earlier actions and stabilises the vehicle even better.

Paper H: "Global force potential of over-actuated electric vehicles"

This paper formulates force constraints of over-actuated road vehicles. In particular, the focus is directed on different vehicle configurations provided with electrical drivelines. It is demonstrated that a number of the configurations possess non-convex tyre and actuator constraints, which differ significantly from the convex force constraints offered by the ACM used in **Paper C** and **Paper D**. By mapping the actuator forces to a space on a global level, the potential of global vehicle forces is investigated for the different vehicle configurations. A method is developed to illustrate the global force potential. It is concluded that vehicles with individual drive, compared with those with individual brakes only, have a great potential for yaw motion, even under strong lateral acceleration.

Paper I: "Investigation of the non-convex force constraints imposed by individual wheel torque allocation"

This paper explains how the force allocation process is influenced by the non-convex tyre force constraints on electric vehicles with individual drive developed in **Paper H**. The roles of the allocator are also discussed. One important role is to deliver solutions even if the requirements arising from motion planning are not physically feasible. To secure vehicle stability, fundamental allocator requirements are formulated. However, more demands are needed to avoid undesired vehicle behaviours. These demands, which belong to the optional allocator requirements, are described. Given the force constraints and the allocator requirements, solutions of the allocation problem for four electrical machines are solved by non-linear programming and demonstrated with different settings of weight factors.

Paper J: "Utilisation of actuators to improve vehicle stability at the limit: from hydraulic brakes towards electric propulsion"

This paper investigates the capability of over-actuated vehicles to maintain stability during limit handling. A number of important differently actuated vehicles, equipped with hydraulic brakes and more advanced chassis solutions, are presented. A virtual evaluation environment is specifically developed to cover the complex interaction between the driver and the vehicle under control. The evaluation environment includes a driver model, a reference model and the vehicle model, that was developed in **Paper F**. To exploit fully the different actuator set-ups and the hard non-convex constraints they possess, the principle of control allocation, developed in **Paper I**, is successfully employed by non-linear optimisation. The final evaluation is carried out by exposing the driver and the over-actuated vehicles to a safety-critical manoeuvre. Thereby, the differently actuated vehicles are ranked by a quantitative indicator of pass/fail for the incident in question.

Chapter 8

Scientific contributions

This chapter presents the main scientific contributions of this thesis and its appended papers, which include answering the overall research question formulated in Chapter 1.

The main scientific contributions related to the appended publications in this thesis can be summarised as follows:

1. The ACMs capability and a qualitative description of its possible uses related to associated research findings (**Paper A**).

2. The closed-loop control principle of ACM using a rule-based mechanism of force allocation is shown to have an inherent robustness to faults. This inherent capacity ensures vehicle stability during fault events and less additional case-specific fault-handling strategies or hardware are needed (**Paper B**). Feedback of actuator constraints into the force allocation process is of importance even when faults occur. Without adaptation of actuator constraints it is shown that vehicle stability is threatened, especially during extreme driving situations (**Paper C**).

3. The ACMs capability of imitating the motion characteristics of a conventional front-steered vehicle trajectory, but with a better utilisation of the adhesion potential. By reducing the vehicle side slip, the ACM demonstrates an improvement in the adhesion potential (also referred to as friction utilisation) (**Paper D**). An allocation method is presented which simultaneously allocates horizontal as well as vertical tyre forces by optimisation.

4. A design method for an electromechanical wheel suspension based on optimisation of both controller and actuator parameters is presented (**Paper E**). It is shown that the compromise between comfort, handling and energy dissipation can be controlled by the adaptation of the dimensioning method concerning the control and actuator parameters during the development process.

5. A significant improvement of the ACM vehicle stability can be obtained with access to path information in advance, even if such information is restricted to be less than 1 s (**Paper G**).

6. The adaptation of a brute-force method to investigate the vehicle force potential of vehicles equipped with different actuators for vehicle motion control (**Paper H**). The method illustrates important differences in the ways in which the configurations could be actuated.

7. The incorporation of combined slip, by an elliptic approximation, in the force constraints used for control allocation of drive and brake forces (**Paper I**).

8. A method to investigate the potential of differently actuated vehicles with optimisation of their force resources (**Paper J**). The evaluation of the vehicles has been performed by investigating their potential to maintain stability during extreme driving conditions and with a human being in the loop.

9. A quantification of the potential for emergency avoidance manoeuvres of differently actuated vehicles is presented (**Paper J**). Since conventional the vehicle of today is included, the improvements versus today's vehicle can be justified. Significant improvement (some 4% higher entry speed in double lane change) can be reached already with individual electric drive on rear axle. Wheel individual drive at both axles does not significantly increase entry speed. Wheel individual drive and steering on both axles do improve the entry speed by 10%. Having friction braking only is a very competitive alternative, since it reduces the speed in the later part of the manoeuvre, but it suffers from poor response time.

Chapter 9

Concluding remarks

This chapter summarises the conclusions drawn from the work presented in this thesis including the appended papers.

This thesis presents the potential of over-actuated road vehicles with individual wheel actuators. Such vehicles possess a potential for further improvement of their vehicle dynamics. Important over-actuated vehicle configurations suited to an electric powertrain and a control architecture are presented. Particular attention is paid to chassis technology design with *Autonomous Corner Modules* (ACMs).

A larger degree of over-actuation is associated with the ability to both exploit tyre force constraints at a high level and to control tyre forces more freely. This freedom can be used to increase the magnitudes of the vehicle's force generation and also to control them more independently of each other. In **Paper J**, it was shown that an 'ultimate' vehicle capable of driving and steering all four wheels independently, could further increase entry speed in a double lane change maneuver by 10% compared with a vehicle utilising individual actuated friction brakes only. This thesis has explained the set of attainable forces for a number of important classes of configurations. Particular focus was directed on configurations equipped with wheel torque actuators. It was shown that with individual brakes only, the vehicle's yaw acceleration is strongly dependent on the lateral acceleration. By introducing two wheel torque actuators at the rear, both able to produce positive and negative torque independent of the spin directions, this dependence can be reduced. However, since small electrical machines have a significant torque limitation, the performance gained from them is also limited.

One important reason for using over-actuated vehicles is their capability to assist the driver to experience the vehicle as desired. The fundamental desire to avoid unstable situations can be handled more efficiently by over-actuation. It was demonstrated that over-actuated vehicles are able to use all available actuators to maintain stability. Along with engine torque and transmission couplings, today's vehicles use individual friction brakes to counteract instability. Since hydraulic friction brakes and combustion engines are hard to control to perfection, they are just used with acceptance of large deviations from the desired motion. Hence, the nature of the brake intervention is on-and-off rather than seamless. Here, the electrical machine becomes a viable option; the response time is low and the output torque is precisely controlled. This thesis has shown that these advantages allow further improvement of the vehicle stability. The motion of the electrically actuated vehicle can be corrected even for minor deviations from the desired motion. However, if the actuators are used continuously with large efforts, the energy consumption may be considered unacceptable.

By enabling rear axle steering, the vehicle side slip can be controlled in a way that is more uncoupled to vehicle yaw rate. It was shown that by allowing side slip to be controlled uncoupled to yaw rate, the adhesion utilisation was further improved. However, there is an important question of how the two degrees of freedom should be mixed during cornering. Preventative safety, sportiness and convenience require different approaches in how the rear axle steer angle is controlled. Therefore, the three categories cannot all be optimised at the same time. In many cases, preventative safety and convenience involve contradictory actions. While the former often leads to side motion without yawing (to achieve high lateral acceleration), the latter requires the vehicle's side slip to be kept low. However, if the traffic situation can be sensed/detected and hence a desired path can be predicted, a trade-off between the three wishes is facilitated. One example is threat detection ahead of the vehicle, which can provide useful information to prioritise preventive safety even more.

Over-actuated vehicles also hold a high level of actuator redundancy. Particular attention was paid to handling actuator faults. The studies carried out demonstrated that the stability of the vehicle can be seriously degraded when an actuator fault occurs. The counterattack to compensate for faults in actuators could be handled with extra inverters and/or controls. However, due to the actuator redundance, it was shown that the remaining working actuators were quickly capable of securing vehicle stability. Moreover, it was established that the over-actuated vehicle holds an inherent capacity to handle actuator faults, without the need for extra hardware or extensions of case-specific fault-handling strategies. During the detection of the fault, the need for a rapid re-calculation of the force constraints was found to be a key issue. The actuator redundance in over-actuated vehicles also offers the opportunity to utilise the road-tyre adhesion differently. By distributing the force margins more evenly, there is a potentially better margin for even tougher tasks or disturbances influencing the vehicle. Conventional front-steered vehicles suffer from an uneven distribution of force margins: one or a few tyres can be saturated while there still exist substantial force margins at the remaining tyres. However, it was shown that the ACM configuration can imitate conventional front-steered vehicle manoeuvres with larger force margins.

Implementations of over-actuated vehicles often lead to actuators being more closely distributed to the contact patch. If they are electromechanical, the natural question to be raised is whether their physical dimensions can be considered acceptable. If electrical wheel torque actuators are used, there is normally a significant torque limitation involved. If their limitations should be kept small, the machines' weight and size may be unacceptably large. A study was therefore initiated to get more insight into the feasibility of introducing an electromechanical suspension. A design and a method were proposed to tackle the preferred compromise between comfort, handling and energy dissipation. If the vehicle is to maintain acceptable performance during severe driving conditions, it turned out that the damper has to be unrealistically large. However, if the electric damper is combined with a hydraulic damper, the size of the electric damper is significantly reduced.

The conventional vehicle can achieve better dynamic performance by adding a few actuators. Throughout the development of vehicles, a number of different actuators and functions for active chassis control have been used to upgrade the level of vehicle dynamics, adding one or a few desired properties. These upgrades, which are rather costly and technically complex, are being refined to perform a specific task perfectly, but do not offer the flexibility to influence functions in a wider perspective. In turn, if more functions are needed, then additional chassis systems need to be added. However, owing to the ongoing electrification of chassis and driveline technology, a new situation has emerged. Electrical actuators, as opposed to combustion engines and mechanical rack steering, can be divided into several units, without largely increasing the total cost and weight, facilitating the development of new functions. Given this situation, this thesis has contributed to better knowledge and methods on how to use over-actuated vehicles employing actuators to generate forces more independently at each corner.

Chapter 10

Recommendations for future work

This chapter provides suggestions for future work that is building on the knowledge gained in the work presented in this thesis

Three suggestions for research continuations are recommended:

1. The research laid out in this thesis is based on access to ideal sensor signals. However, many variables from the vehicle and its subsystems are difficult and costly to measure or estimate with a sufficient level of accuracy. Traditionally, friction coefficients and side slip from the vehicle and tyres are examples of signals which are unmeasured or estimated with a large portion of uncertainty. Even in the case of the best over-actuated vehicle, an optimal control algorithm cannot exploit the vehicle's full potential if such variables are too uncertain. Today, velocity and acceleration sensors are commonly being used for controlling the vehicle. Force and torque sensors are considered as too expensive and are typically rarely used. However, the introduction of electrical machines gives new opportunities, since rotor torques can be instantaneously measured via the machine current. As a research continuation, it is proposed to further investigate in what manner the non-ideal sensor data influences the over-actuated vehicle and how it should be adapted to the uncertainties.

2. Along with the automotive usage of preview sensors, such as radar, global-positioning-systems, cameras, etc, it is highly interesting to develop path optimisation for over-actuated vehicles further. Since force allocation

controls the vehicle instantaneously, long-term effects are not considered. Particulary, indirect effects of actuation are difficult to find. This implies that the vehicle in the short term may perform actions that are irrelevant or inefficient, but in the long run, are favorable. One pedagogic example is brake actuation before entering a curve to avoid a potential accident. Therefore it is recommended to make further studies on dynamic path optimisation. Here it is important that the vehicle's non-linear force constraints should be taken into account and also that a prediction of the available force constraints should included in the formulation.

3. As soon as the rear axle steering angle can be controlled along with the front axle, the vehicle side slip can be controlled independently from yaw rate. This extra freedom can be utilised to satisfy different preferences. Most likely, a small or no side slip is preferred for best comfort, while a large side slip can be needed if the vehicle is exposed to an evasive manoeuvre. Since comfort and safety do not for certain go hand in hand, this control problem concerns a trade-off between two preferences. Hence, it would be interesting to investigate further how a mixing of these two quantities should be performed, assuming knowledge of the current traffic scenario.

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Appendix A

Nomenclature and glossary

Latin symbols

a_x, a_y	Longitudinal and lateral vehicle acceleration
$\mathbf{A}$	Vehicle geometry matrix
df_z	Relative change in wheel load
f_x, f_y, f_z	Longitudinal, lateral and vertical tyre forces
f_z^{nom}	Nominal wheel load
F_x, F_y, F_z	Longitudinal, lateral and vertical global vehicle forces
$\mathbf{f}$	Tyre force vector
$\mathbf{f}^c$	Corner force vector
$\mathbf{f}^{\text{glob}}$	Vector of global forces and moments
g	Gravitational constant
h, l_f, l_r, w_l, w_r	Vehicle geometry parameters
$J_x, J_y, J_z, \mathbf{J}$	Vehicle inertia and vehicle inertia matrix
m	Vehicle mass
M_x, M_y, M_z	Global vehicle roll, pitch and yaw moments
$p_{Dx1}, p_{Dx2}, p_{Dx2}, p_{Dy1}, p_{Dy2}, p_{Dy3}$	MF tyre parameters
r	Rate limit
r_w	Effective rolling radius
S	Attainable set of tyre forces
u	Actuator quantities as vehicle control input
$\mathbf{v}$	Vehicle translation velocity vector

$\mathbf{v}_w$	Wheel velocity vector
t, T	Time and sampling time step
T_e, T_f	Electric actuator and friction brake wheel torques
T_w	Total wheel torque
$\mathbf{T}$	Transformation matrix
v_x, v_y, v_z	Vehicle longitudinal, lateral and vertical velocities
$\mathbf{W}$	Weight matrix
$\mathbf{x}$	Vehicle translational and rotational velocity states
X, Y	Vehicle longitudinal and lateral displacement in road coordinate system

Greek symbols

α	Tyre side slip angle
β_c	Wheel side slip angle
γ	Camber angle
δ	Wheel steering angle
η	Tyre force utilisation
θ	Vehicle yaw angle in road coordinate system
κ	Longitudinal tyre slip
μ	Friction coefficient
ω	Wheel rotation speed
ω	Vehicle rotation speed vector
Ω	Attainable set of global forces

Subscripts and superscript

act	Actuator
c	Variable valid for a vehicle corner in the vehicle's system of coordinates
$glob$	Global
i	Wheel index
ref	Reference
w	Variable valid for a tyre in the tyres's system of coordinates

Abbreviations

ABS	Antilock Braking System
ACM	Autonomous Corner Module
ECU	Electronic Control Unit
EPAS	Electronic Power Assisted Steering
ESC	Electronic Stability Control
KTH	Royal Institute of Technology in Stockholm, Sweden
MF	Magic Formula tyre model
MMM	Milliken Moment Method
MPC	Model Predictive Control
PMSM	Permanent-Magnet Synchronous Machine
VCC	Volvo Car Corporation

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Paper B

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Paper J

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[1] Errors have been corrected.