

UNICARagil Dynamics Module

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Summary

In the project "UNICARagil", funded by the Federal Ministry of Education and Research, the leading German universities in the field of automated driving have joined with selected specialists from industry to rethink automated vehicles and their architecture. An essential component in this project is the so-called dynamics or corner module. In this module, the functions accelerating, braking and steering are combined and controlled via an assigned control unit. For that, a 48 V wheel hub motor and a perimeter brake are packaged into a custom designed wheel. The wheel in turn is connected to the vehicle by a double wishbone suspension with an integrated 48 V steering actuator. This paper gives an overview of the dimensioning and design of the individual components and the connection to a complete system.

1 Introduction

Today, by-wire systems are used in a wide range of technical areas. The first (analog) fly-by-wire system in civil aviation was realized in the Concorde in the 1970s, followed by the first system without a mechanical backup in the Airbus A320 in 1987. By now, drive-by-wire systems in form of electronic accelerator pedals are state of the art for passenger cars. However, both brake-by-wire and steer-by-wire systems are still not used in the automotive series production, mainly due to higher costs and complexity to fulfill safety requirements. [1]

But due to political and social pressure to reduce CO₂-emissions, particularly in the transport sector, combined with efforts to automate motor vehicles, brake and steer-by-wire systems are becoming increasingly important. The combination of electrification and automation is especially beneficial, because automation depends on the possibility to control components by-wire, which electric machines require or offer inherently. Several concept and research vehicles already make use of this synergy (e. g.: concepts of Bosch, BMW and Mercedes-Benz, Rinspeed Snap, Navya shuttle, ...). The same is the case in UNICARagil, but with the focus to realize not only a driving platform but also a vehicle architecture for autonomous driving that is modular and scalable. Modularity and scalability are demonstrated by building four prototype vehicles: On the one hand, a shuttle and a delivery vehicle with long wheelbase and

on the other hand a taxi and a private vehicle with short wheelbase [2]. In this regard, the dynamic module plays an important role, since two or more modules can be used to propel and steer vehicles depending on the application. The aspect of modularity and scalability is also the reason why each module has its own control unit.

This paper gives an overview of the dimensioning and design of the previously mentioned dynamics module. For this purpose, the overall vehicle requirements are discussed in the first part of section 2. Subsequently, the application-specific design of the drive and brake system is described before explaining the design process of the chassis. This is followed by the electrical dimensioning of the steering system based on relevant driving maneuvers. To conclude section 2, the wheel design, adapted to accommodate the electric wheel hub motor and a mechanical brake, is described. Section 3 then gives an overview of the control unit and inverter. Based on this, the structure of the software is outlined before the paper ends with conclusions and an outlook.

2 Vehicle Requirements and Component Design

The development process of the research project UNICARagil is intended to be disruptive and agile, which is why only a few overall vehicle requirements were defined at the beginning of the project. A key point is that all electrical components operate at a voltage level of 48 V or below. This enables people without high voltage training to work on the vehicles, which is very well suited for the use as research vehicle at different institutes. Another decision that was initially made, is the use of electric wheel hub motors in order to be able to realize large steering angles up to 90°. The steering system is also electric and actuated by-wire, which allows each wheel to be powered and steered individually. This enables completely new forms of movement such as bidirectional driving, driving sideways or driving on rail-like trajectories. These trajectories are calculated in a higher-level control unit, the so-called cerebrum, and transmitted to the brain stem. The brain stem is responsible for vehicle dynamics control and provides the target values for the dynamics modules. In order to compensate faults and errors in the control units of the dynamics modules they are all connected to form the spinal cord. The described architecture is illustrated in Fig. 1. [2]

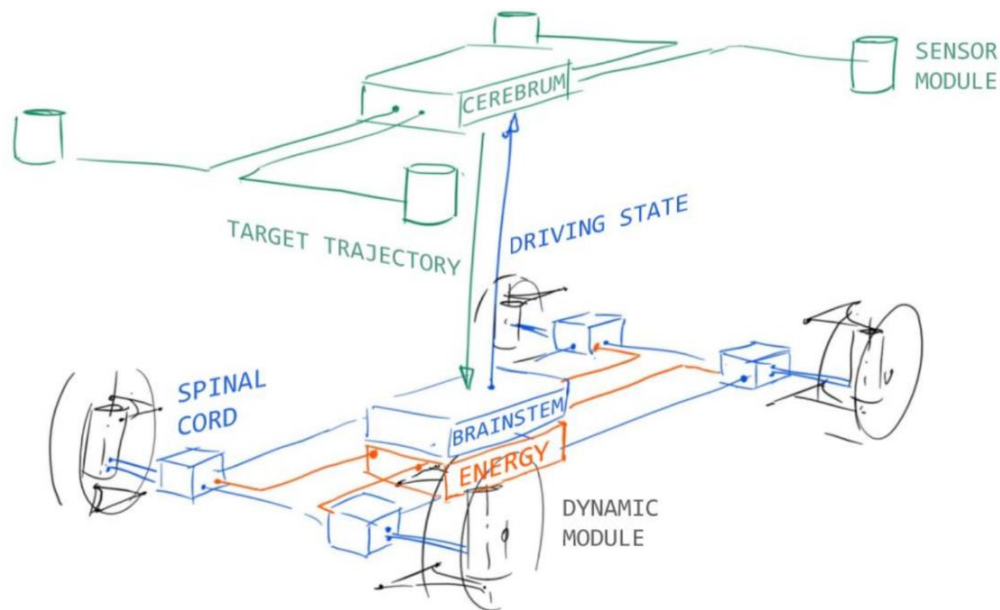


Fig. 1 Control-unit architecture analogous to the human nervous system [2]

2.1 Wheel Hub Motor and Brake

In addition to the overall vehicle requirements mentioned in the previous paragraph, there are further requirements and parameters, which must be known initially in order to design the powertrain. As the design of the powertrain and the vehicle body started at the same time, parts of the requirements and parameters are reference values or estimations. These are, for example, vehicle mass, rolling resistance coefficient, aerodynamic drag coefficient, frontal area, maximum velocity and maximum acceleration. The target maximum velocity for instance is 70 km/h, which is also set for the powertrain dimensioning. In bidirectional operation, on the other hand, the speed is limited to 30 km/h to ensure sufficient cooling. In order not to over-dimension the powertrain of the vehicles and to establish an agile development process, the approach is chosen to focus the design on the operation during testing and demonstration according to the use-case. The shuttle in particular has a high total mass and the maximum torque that can be delivered by the wheel hub motor is limited by the installation space in the wheel. Because of that, those maneuvers that require high torque at the wheels are considered especially relevant. These are primarily the maximum acceleration and recuperative deceleration as well as the maximum climbing ability. With regard to the gradients, potential locations for testing are primarily Munich, Braunschweig and Aachen. The maximum values for the enclosed test field in Aldenhoven near Aachen are below 1 % and possible routes in the city or on testing grounds in Munich are also considered flat. Therefore, possible test routes in Braunschweig and Aachen are examined to take account of the impact. Fig. 2 shows two routes in Aachen. The "Melaten lap" for instance, is used in the project "L3Pilot" for automated test drives and is therefore a suitable example. The route 'campus'-'city center'-'campus', on the other hand, is intended to show how an automated shuttle could effectively supplement public transport.

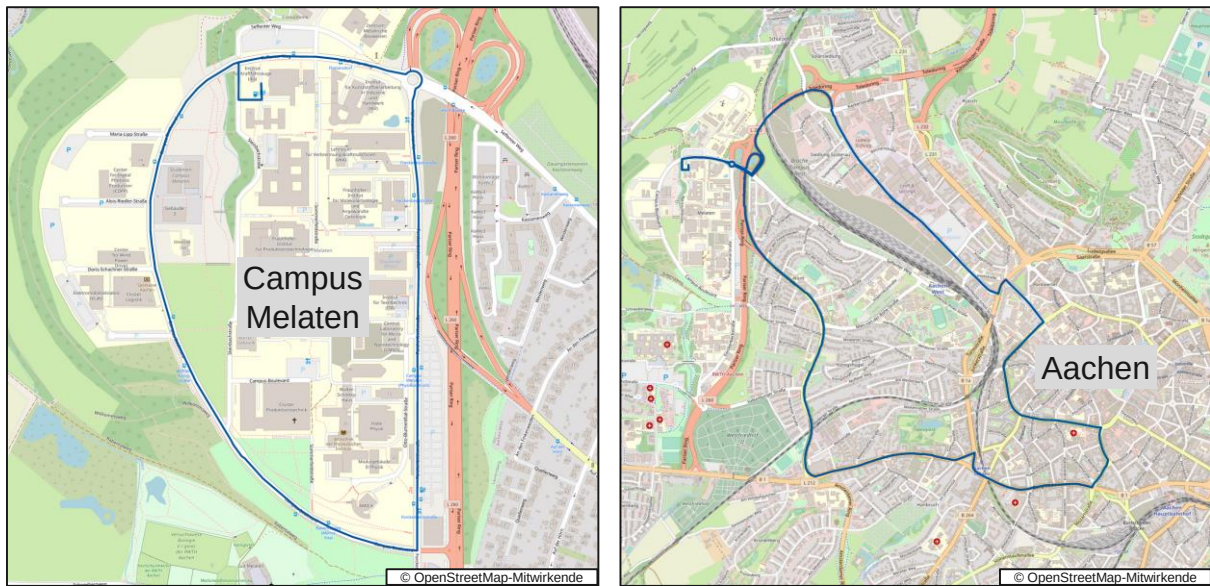


Fig. 2 Route on campus Melaten (left) and example route 'campus'-'city center'-'campus' (right)

The top part of Fig. 3 gives, as an example, the elevation of the route on campus Melaten. From that, the gradient is calculated, which is shown in the bottom of Fig. 3. However, if the raw values are used for the calculation, there are, as clearly visible in the graph, unrealistically large jumps in the gradient. The height profile is therefore smoothed prior to the gradient calculation. In order to evaluate deviations due to smoothing, the height profile is recalculated from the smoothed gradient. If the deviations from the original height profile are too large, the smoothing can be adjusted iteratively.

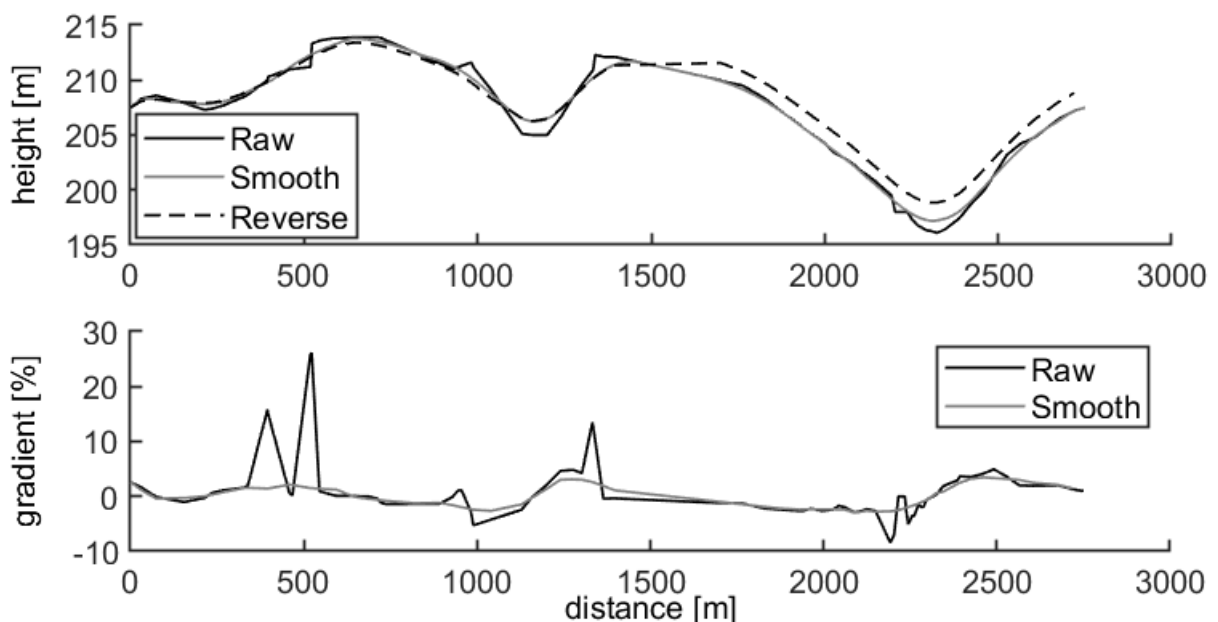


Fig. 3 Altitude profile and calculated gradient of the route on campus Melaten

Besides the climbing ability, the desired acceleration of the vehicle is also a design criterion for the powertrain and by that for the electric machine. Regardless of the vehicle variant, all prototypes in UNICARagil drive autonomously. In this case, the perception of comfort and safety differs significantly from manual driving and both longitudinal and lateral acceleration have an influence on this [3]. As a baseline, requirements for the longitudinal and lateral acceleration are discussed in [4] and [5] based on real-driving data of conventional vehicles. Results of the data evaluation are that only 0.5 % of the overall accelerations exceed 1.7 m/s^2 in longitudinal and 2 m/s^2 in lateral direction [4]. Higher values are identified in [5], with the addition that a combination of longitudinal and lateral acceleration lowers these values. This correlation is shown in Fig. 4. There are also publications and studies, in which comfort ranges or maximum values for longitudinal and lateral dynamics are investigated for public transport or automated driving. For public transportation, accelerations shall not be higher than 1.3 m/s^2 if people are standing in the vehicle [6]. Since this is the case in the autonomous shuttle, this value shall be taken into account. In dangerous situations, however, it must be ensured that higher decelerations are possible. According to the German Road Traffic Licensing Regulation (StVZO) a motor vehicle must achieve an average deceleration of at least 5.0 m/s^2 [7]. A similar value is demanded for the approval of passenger cars in the European regulation No. 13-H, which is at least 5.76 m/s^2 [8]. As emergency braking occurs rarely, 1.3 m/s^2 is set as the target value for electrical acceleration and deceleration. To carry out emergency braking and to serve as a fallback level, an additional mechanical brake is to be integrated.

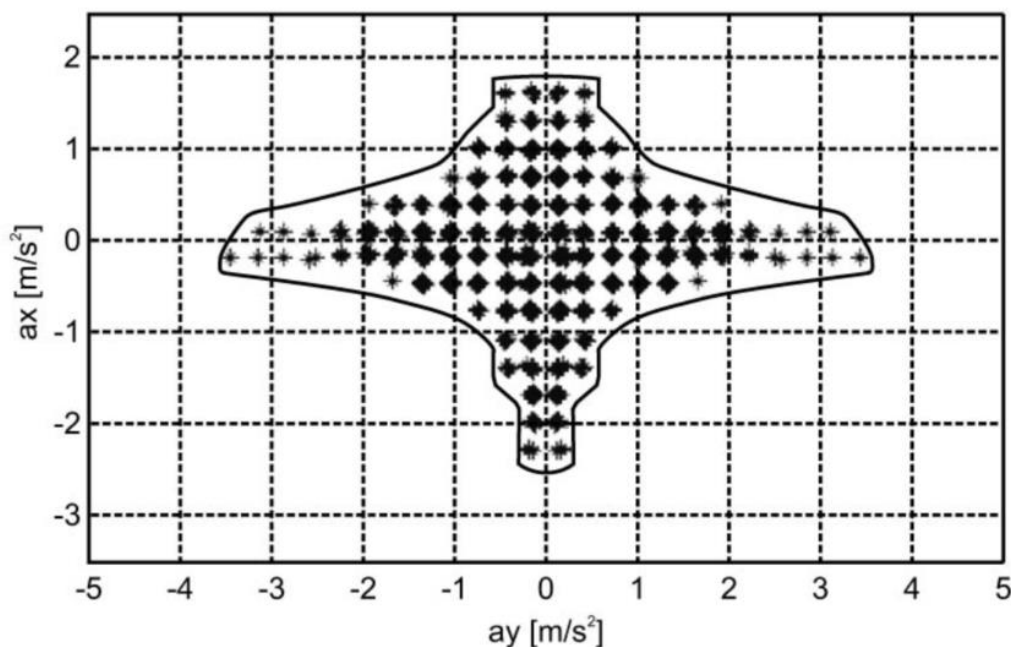


Fig. 4 85 % percentile of longitudinal and lateral acceleration of average drivers in public road transport [5]

With the knowledge of the required climbing ability and acceleration, the wheel hub motor and brake system can be investigated. The basis for the design is a 48 V wheel hub motor available at the project partner Schaeffler. The schematic assembly is shown in Fig. 5.

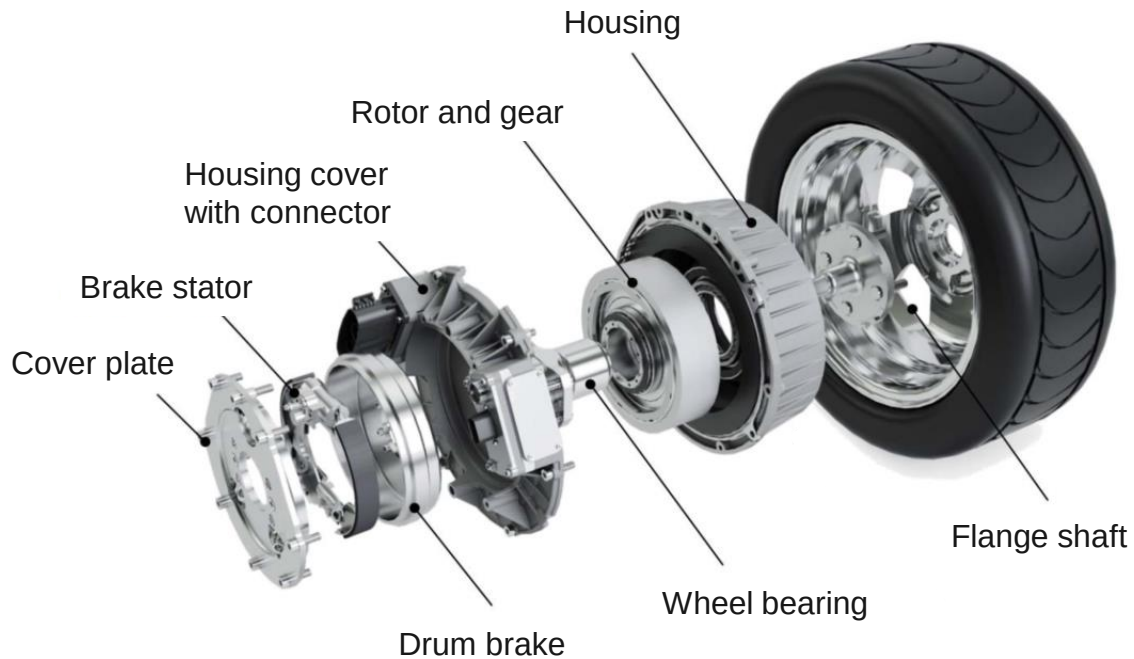


Fig. 5 Wheel hub motor assembly

Besides the standard components for an electric machine, the design also includes a drum brake and a gear stage to increase the output torque. However, since the drum brake cannot provide sufficient braking power to achieve the deceleration required by the StVZO and ECE R13-H for the heavier vehicles, it is not part of the final design. The advantage here is that more package space is available for the transmission stage and a more robust wheel bearing, which is necessary for the shuttle and the delivery vehicle. With the gear stage, realized as planetary gear, and considering power restrictions from the battery the wheel hub motor can deliver a peak torque of 500 Nm at the wheel and a peak power of 14 kW. The continuously available maximum values for torque and power are 280 Nm and 10 kW. This results in 56 kW and 2000 Nm for the whole vehicle in peak performance and 40 kW and 1120 Nm for continuous performance respectively. In order to evaluate these values, a simple assessment, based on driving resistance equations, for acceleration and climbing ability is carried out. Peak power and torque are used to calculate the maximum acceleration whereas the calculation of the climbing ability is based on the continuous power and torque. As can be read off in the upper part of Fig. 6, the requirement of an acceleration of 1.3 m/s^2 is met up to a speed of approx. 42 km/h and at 50 km/h an acceleration of 1 m/s^2 is still possible. Since the peak values can be delivered for 20 s and an acceleration to 50 km/h takes less time, the use of the peak values is reasonable. Regarding the climbing ability, a maximum gradient of 7 % is found in the route 'melaten'-'city center'-'melaten', which is continuously drivable up to a speed of 50 km/h. The continuous power is used to make sure that the gradients can be climbed even in derating e.g. due to high motor temperatures or lower supply voltages. This also justifies the rather strong smoothing of the height profile. However, by using the peak torque even higher gradients of a maximum of nearly 19 % are climbable.

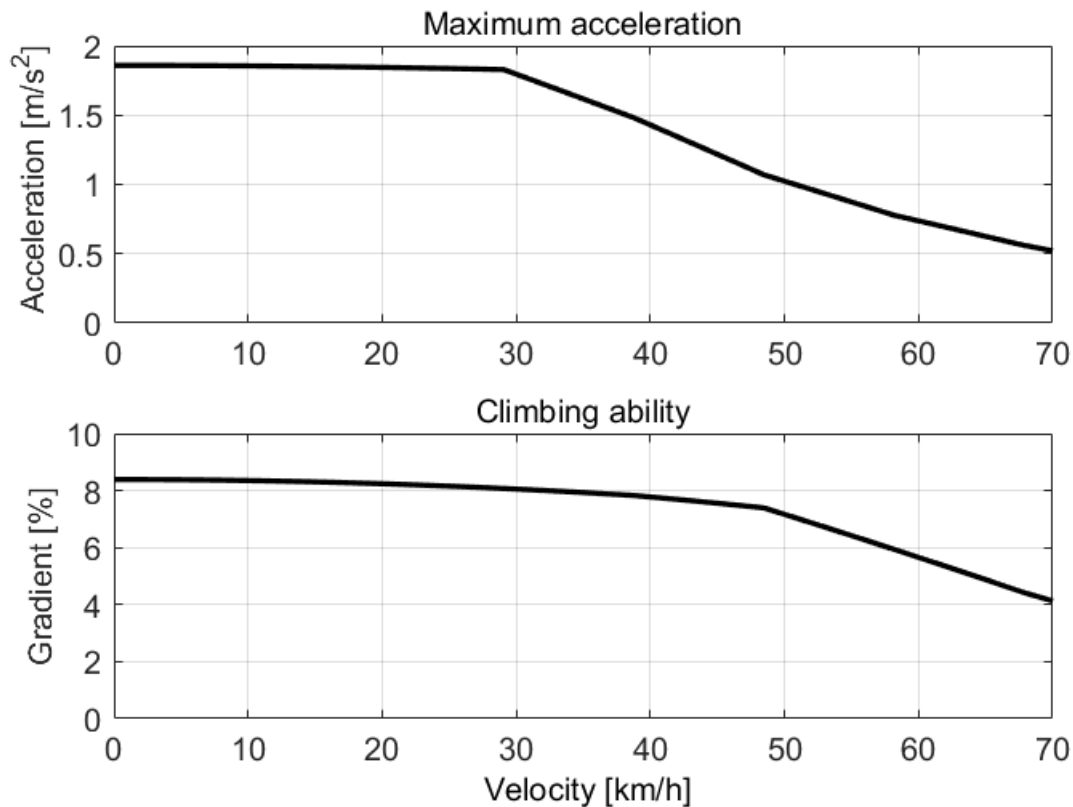


Fig. 6 Maximum acceleration and climbing ability depending on vehicle speed

2.2 Chassis

In the following section, the chassis concept of the highly maneuverable fully automated UNICARagil vehicles is presented. The idea of modularity is also reflected in the wheel suspension. For the scalable vehicle platforms with wheelbases of 2800 mm and 3400 mm, the chassis consists of four equivalent dynamics modules.

Chassis concepts for autonomous driving with steer-by-wire systems differ significantly from conventional chassis concepts. By eliminating the drive shaft and the steering linkage, high steering angles are achievable. In addition, dynamic effects caused by changes in the wheel position, which in conventional chassis are influenced by the design, can be specifically generated or compensated by controlled steering systems. The development of the UNICARagil dynamics modules is based on the design of the front axle concept of the research vehicle SpeedE, developed at the Institute for Automotive Engineering at RWTH Aachen University (ika). The front axle of SpeedE described in [9] also uses a steer-by-wire system and high steering angles of 60° on the outer and 90° on the inner wheel. Due to the double wishbone suspension, changes of the wheel position relative to the body that occur during compression and rebound are variable in a wide range. From a driving dynamics point of view the aim is to achieve optimum contact between the wheels and the road surface. The position of the roll pole or roll center and the amount of anti-dive compensation can also be specifically influenced in order to meet driving stability or comfort criteria.

The focus of the further development is the integration of the aforementioned electric wheel hub motor, while simultaneously increasing the range of the steering angles. Another objective of the design is to enable the UNICARagil vehicles to perform the following driving maneuvers that are also depicted in Fig. 7.

- Driving sideways with a steering angle of 90° at all wheels.
- Turning maneuver with the rear inner wheel set as instantaneous center [9].
- Turning on the spot around the vertical axis of the vehicle.

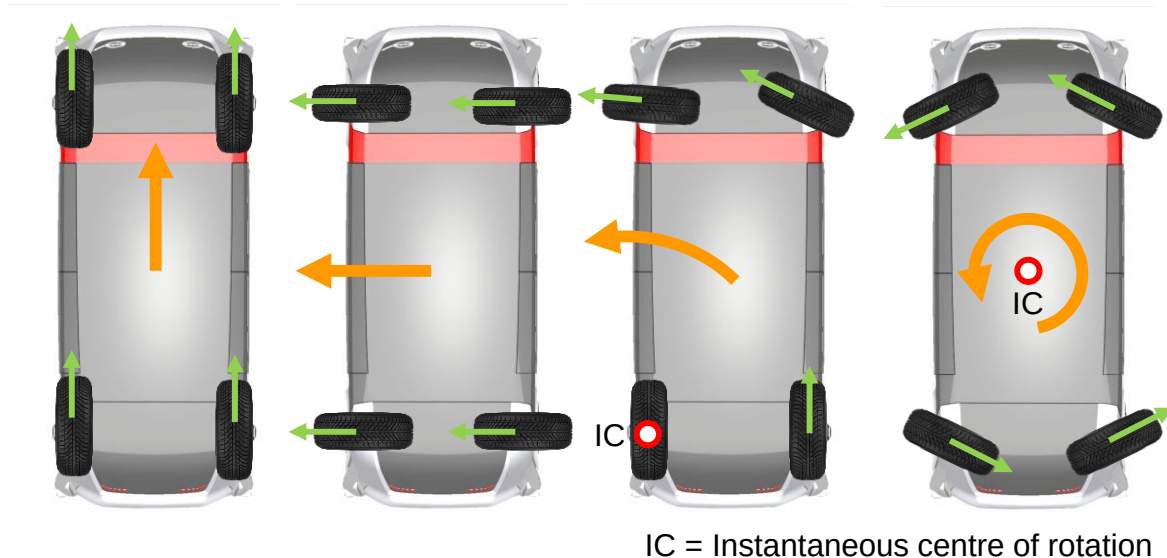


Fig. 7 From left to right: Straight driving, sideways driving, turning around a rear wheel and turning on the spot

For this purpose, the steering angle range must be extended to 66° on the outer and 93° on the inner wheel. In addition, the scrub radius should be as small as possible, as it has a significant influence on the steering system's actuating torque. Due to the comparatively high mass of the UNICARagil vehicles and especially of the shuttle, a tire width of at least 195 mm is required. In combination with the requested steering angles and minimal scrub radius, the primary challenge is to realize a compact packaging. The design potential of the wheel kinematics for a double wishbone suspension is hence significantly restricted in preference to higher steering angles. For precise control of the steering angle via the steering system, the smallest possible toe angle change caused by the wheel kinematics is required. Wheel slip angles due to the wheel kinematics cannot be detected by the steer-by-wire system and can only be estimated even with additional sensors. According to conventional wheel kinematics, however, changes in toe angle over wheel travel are used to counteract an oversteering tendency. The outer wheel of the front axle should therefore toe-out while the outer wheel of the rear axle should toe in during jounce movement. Between the two requirements, a compromise shall be identified and realized. In order to counteract a reduction of the tyres contact patch caused by a body rolling motion during cornering, the negative wheel camber shall increase during jounce movement.

Since the shuttle in particular is a heavy vehicle with a high center of gravity (COG), the suspension travel and subsequently the wheel kinematics have to be adapted in order to realize roll stabilization and anti-dive compensation. Roll stabilization depends on the spring forces acting on the vehicle body due to wheel load shifts during lateral acceleration. In consequence of the modularity, no anti-roll bars are installed, so the roll stabilization has to be achieved solely by the spring-damper system. The roll angle for UNICARagil vehicles shall be limited to 3° at a lateral acceleration of 4 m/s^2 . The spring stiffness and thus the spring forces, however, cannot be increased infinitely due to vibration and comfort characteristics of the vehicles. One possibility to increase the spring forces and thus the torque reaction around the COG due to roll angles is to realize the largest possible lever between COG and the springs. This results in additional limitations in the assembly space and in the positioning of the steering axle.

With regard to modularity, a few compromises have to be made. In order to realize the desired steering angles and thus driving maneuvers, the steering angle ranges at the front and rear axle need to be mirrored. Since this already creates different variants for the front and rear axle and the vehicle also has a preferred driving direction, the axle kinematics are also adjusted accordingly. For the kinematic design of the double wishbone suspension, it is also necessary to mirror the dynamics modules on the vehicle's longitudinal axis. Another adjustment is the use of different springs depending on the vehicle mass and thus the UNICARagil vehicle type to improve the driving comfort. Fig. 8 shows the assembly of the dynamics modules of the front axle with the components of the double wishbone suspension, the steer-by-wire system, the electric wheel hub motor and the mechanical perimeter brake.

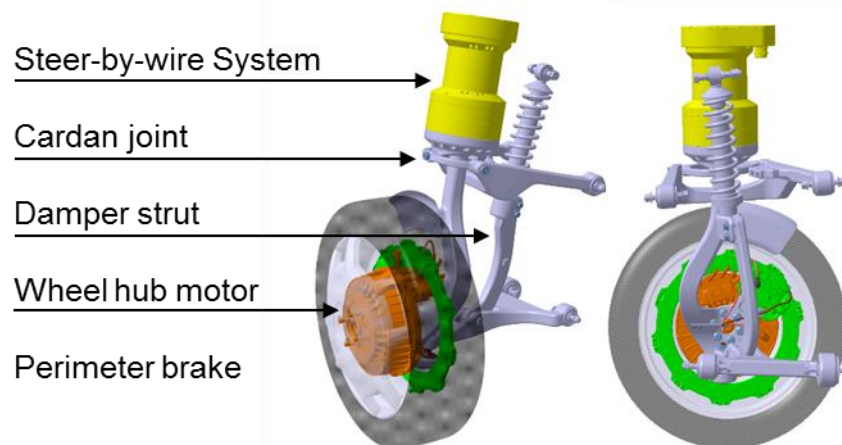


Fig. 8 Assembly of the dynamics module of the front axle

The use of the perimeter brake in combination with a 20" wheel provides additional assembly space within the wheel. The brake caliper, the connection to the wheel carrier and the 14" wheel hub motor including cables can be positioned, at least partially, in this installation space. This helps to reduce the scrub radius, as the steering axis can be positioned closer to the wheel contact point. Despite all efforts, this still results in a scrub radius of 73.86 mm. However, even this value is still within the range of common steering systems with scrub radius between -20 and +80 mm [9]. The double wishbone suspension consists, comparable to conventional designs, of an upper and

lower control arm, that are connected to the body via vibration-isolating and angularly movable rubber-metal bushings. Through the wheel carrier and two joints, the kinematic four-link chain is closed and the position of the steering axle is determined. The tie rod and the upper ball joint are replaced by a cardan joint, which consists of two defined axes of rotation and the interlinked inner and outer cardan ring. The outer cardan ring is connected to the upper control arm. The inner cardan ring is attached to the housing of the steering system.

The steer-by-wire system consists of an electric motor, sensors for measuring the steering angles and torques around the steering axis, a holding brake and a strain wave gearing. The strain wave gearing is fixed to the wheel carrier using an adapter. In this way, the steering torque can be supported on the vehicle body via the cardan joint and the upper wishbone, while also enabling relative rotation between the wheel carrier and the upper control arm during jounce and rebound movement. The design and effects of such a cardan joint are described in detail in [10]. In order to meet the requirement for the largest possible spring track width, the design of the cardan joint was modified from flat, interlocking cardan rings to a design in which the screw flange of the inner cardan ring lies above the outer cardan ring. This reduces the required horizontal assembly space even with a larger dimensioned electrical actuator.

The dynamics module is completed by the spring-damping-system, which is attached to the lower control arm via a damper strut. The sickle-shaped damper strut is designed to meet the two main requirements steering angle range and roll stabilization. In this way, the spring can be positioned with a maximum distance to the vehicle's longitudinal axis and clearance is created for the brake caliper during steering and jounce movement. In total, the measures to increase the roll stabilization result in a spring track width of 778.55 mm for a track width of 1700 mm. Fig. 9 shows the roll angle as a function of lateral acceleration determined in the multibody simulation (MBS) software "Adams Car". The values are simulated for the heaviest UNICARagil vehicle, the shuttle, for stationary cornering. A lateral acceleration of 4 m/s^2 results in a roll angle of 2.26° .

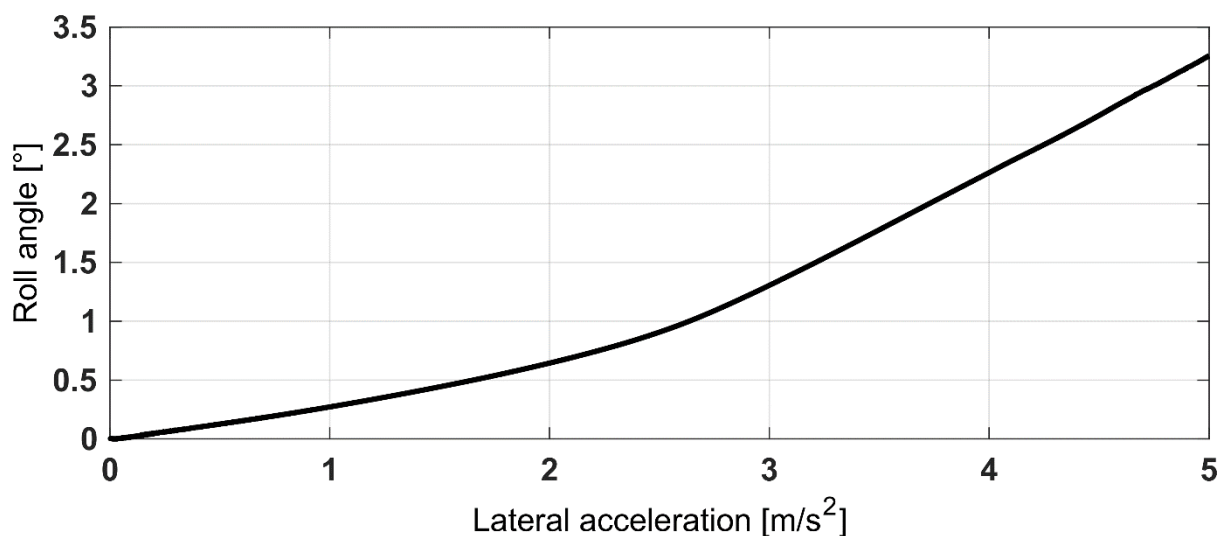


Fig. 9 Roll angle as a function of lateral acceleration from MBS

MBS is also used to determine dynamic wheel camber and dynamic toe angle depending on the wheel travel. The results for front and rear axle are shown in Fig. 10. Positive values of the wheel travel represent a jounce movement. From the compromise of the requirements for conventional axle kinematics and, at the same time, the use of a steer-by-wire system, a dynamic change of the toe angle on the front axle with a maximum deviation of less than 0.15° in toe-out is realized. On the rear axle, toe-in is achieved with a maximum deviation of approx. 0.1° during jounce movement. It is also accomplished, that the negative wheel camber increases during jounce movement to adjust the position of the tire contact patch during cornering with a maximum value of -2.5° .

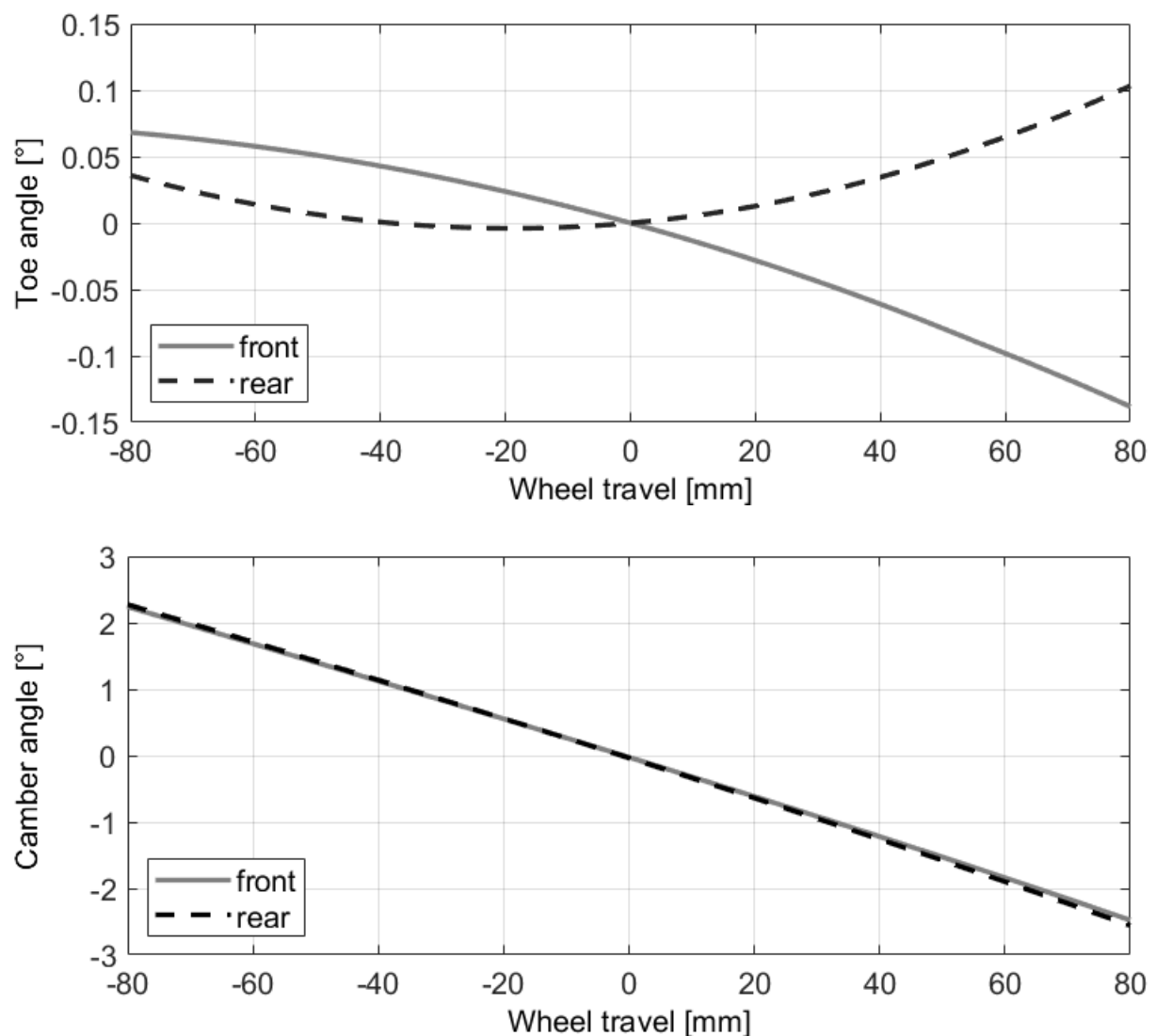


Fig. 10 Toe angle and camber angle depending on wheel travel from MBS

2.3 Electric Steering

When designing conventional or by-wire steering systems, the driver's steering feel is of great importance [1]. In the UNICARagil project, autonomous driverless vehicles are developed, which fundamentally changes the demands on the steering system.

For the design of the steering motor, additional simulations are carried out in the Software "Adams Car". The constraints in the simulations are derived from the overall vehicle requirements that have an influence on the steering motor. In addition to the wheel steering angle range of $[-66^\circ, 93^\circ]$, the total weight of 3200 kg for the heaviest vehicle, the bi-directionality with a preferred driving direction, the different vehicle lengths and especially the selected tire provide key data for the simulation. The aforementioned necessary tire width and wheel size led to the selection of a tire with the dimensions 195/55R20.

For the dimensioning of the steering motor, driving maneuvers are selected that provide relevant information about the torques around the steering axle. The first maneuver is standstill steering with applied brake. The combination of wheel load, friction coefficient and torsional stiffness of the tire generates forces around the steering axle due to constraints in the tire contact patch. Together with the different disturbance force lever arms, these forces result in a reaction torque around the steering axle that must be overcome by the steering motor [11]. Fig. 11 shows the steering torque requirement over the wheel steering angle of the front left wheel for the fully loaded shuttle at a friction coefficient of 1. This requirement represents the peak operation state of the steering motor. In order to execute this maneuver, a maximum torque of at least 1290 Nm at a minimum of 5 min^{-1} must be reached at the wheel. To protect the steering unit, it only needs to be possible to perform this maneuver 5 times in succession within 70 seconds. After reaching the critical temperature of the steering motor or the inverter, the maneuver can only be performed after a cooling phase of 3 minutes.

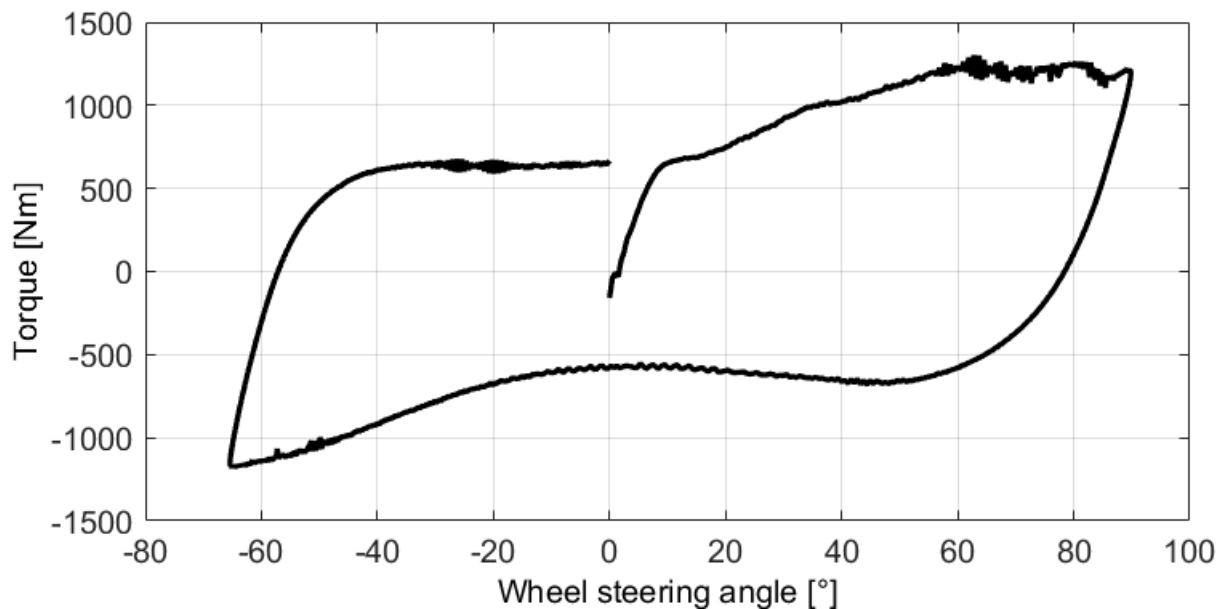


Fig. 11 Steering in standstill with brakes applied

The next maneuver is a quasi-static constant radius cornering. This simulates the continuous operation state of the steering motor. Fig. 12 shows the steering torque requirement of both wheels of the front axle for the shuttle at a friction coefficient of 1. Due to the larger steering angle at the inner wheels, the steering torque requirement is correspondingly larger at the beginning of the maneuver. As the lateral acceleration increases, the body's rolling motion creates a wheel load difference. This results in

increasing load on the outer and a decreasing load on the inner wheels. The torque demand thus increases for the outer wheels and decreases for the inner wheels. For moderate lateral accelerations of up to 2 m/s^2 , a continuous torque of only approx. 200 Nm is required. However, taking into account the use cases taxi and private vehicle, higher lateral accelerations should also be possible. Consequently, 4 m/s^2 is set as the target lateral acceleration, just as for the chassis design. This results in a continuously available torque requirement of approx. 340 Nm at a minimum angular speed of 15 min^{-1} .

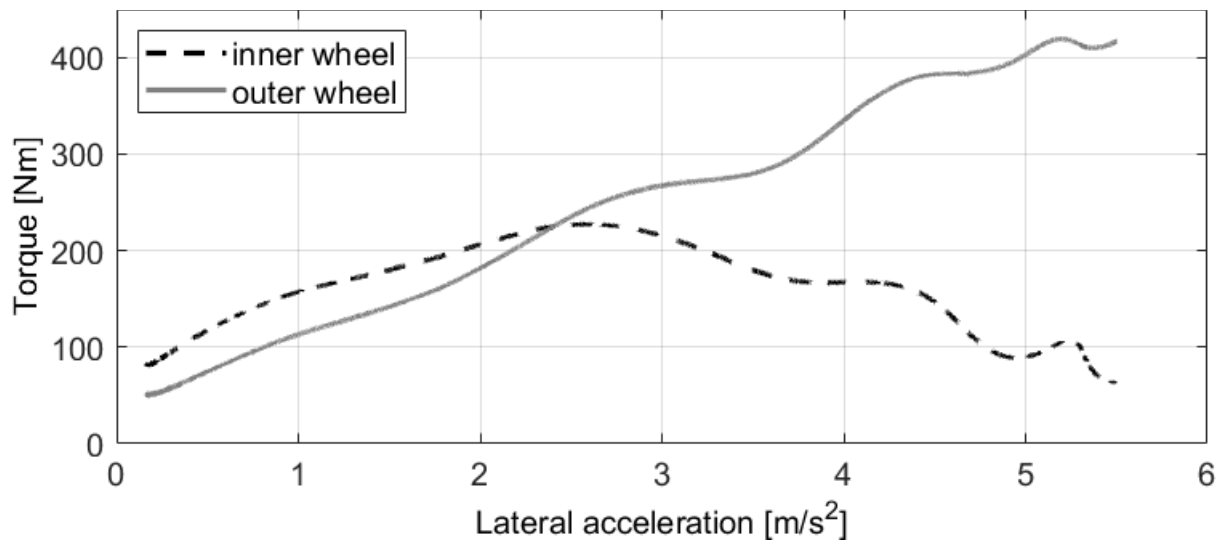


Fig. 12 Quasi-static cornering at constant radius

The last maneuver is a full deceleration. Here the steering torque requirement is determined, which the steering motor must provide in order to maintain a given steering angle during braking. For this purpose, the vehicle is first accelerated to 70 km/h at a friction coefficient of 1 and then brakes to a standstill with -1 g maintaining a steering angle of 0° . As in the first maneuver, this requirement also represents a peak operation state of the steering motor. Fig. 13 shows the steering torque requirement. For this maneuver, the steering motor needs to provide a torque of at least 620 Nm at the wheel. The number of repetitions is limited to 10 consecutive full decelerations. As in the first maneuver, after reaching the critical temperature of the steering motor or the inverter, the maneuver can be performed again after a cooling phase of 3 minutes.

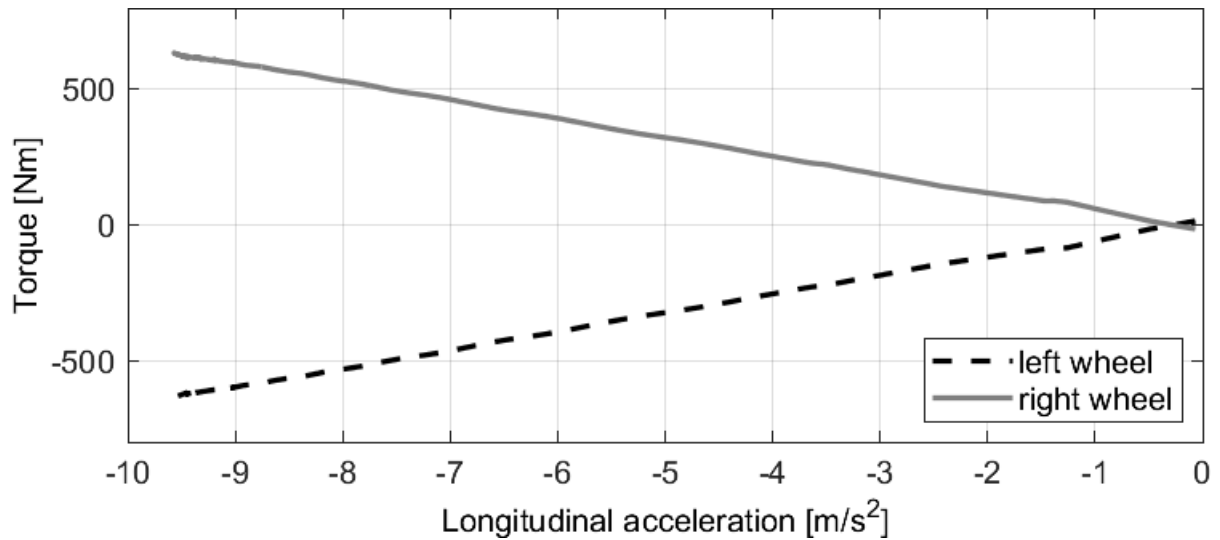


Fig. 13 Full deceleration

In addition to these maneuvers, the additional dynamic torque required to accelerate the entire steering system must be determined. Based on a single lane change with the research vehicle SpeedE, the wheel acceleration was calculated using the measured wheel steering angle (see Fig. 14). With this estimation, the wheel acceleration provides an estimate for the dynamic steering torque requirement of the UNICARagil vehicle as a function of the moment of inertia. For this maneuver, an upper limit of 10 repetitions and a cooling phase of 3 minutes was also defined.

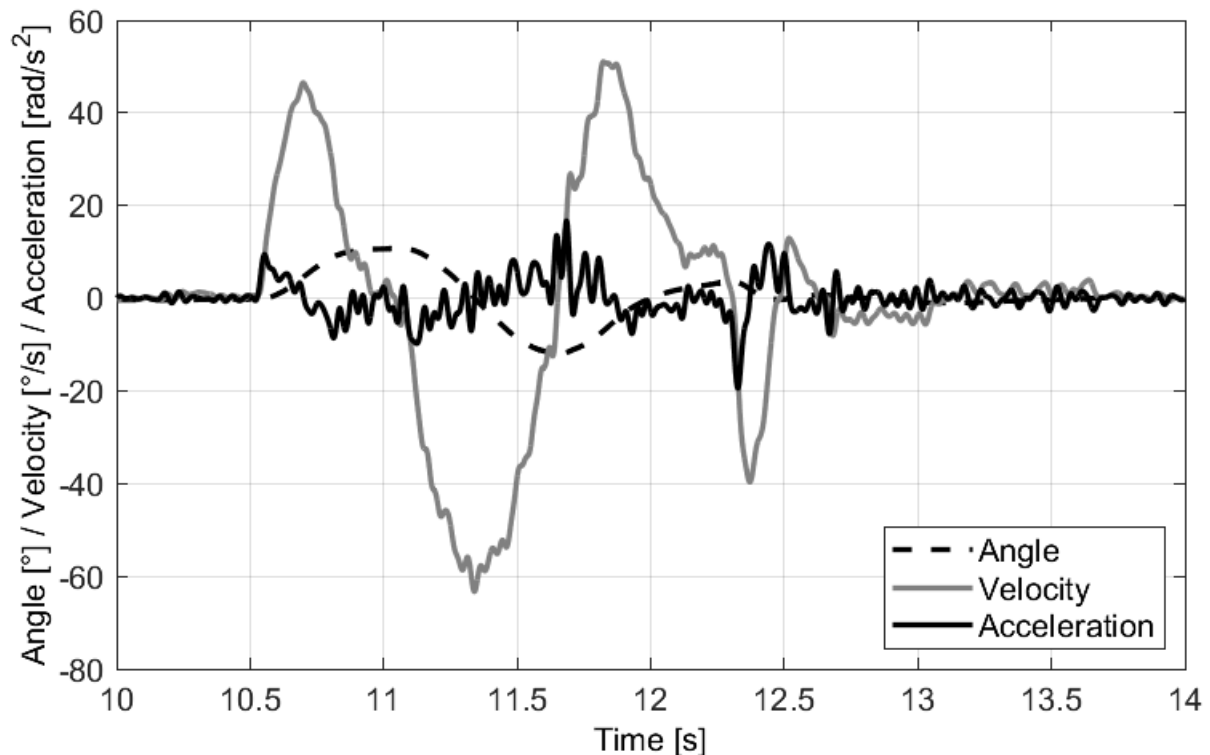


Fig. 14 Wheel steering angle, velocity, acceleration during single lane change

Further requirements for the dynamics module that directly affect the steering system are the self-diagnosis capability and the fail-silent safety requirement. To meet these requirements, the following components are integrated:

- Brake: A brake that can be controlled separately via a 24 V supply is integrated in each steering system. The brake blocks the steering in case of a detected malfunction in order to fulfill the fail-silent requirement.
- Temperature sensors: The sensors are used to measure the critical temperature relevant for derating. This value gives information about the condition of the steering motor.
- Torque sensor: The sensor is used to measure the torque on the output of the steering system. It helps to detect disturbance torque occurring on the wheel and damages within the strain wave gearing.

Based on these requirements, as well as the constructive and electrical ones, the overall specifications for the steering system are derived.

2.4 Wheel Design

The usage of an electric steering system in combination with a wheel hub motor does not only have a strong influence on the chassis design but also on the wheel. The wheel must be designed to meet the following requirements.

- Tire dimensions: 195/55R20
- Connection bolts: 4 x M12, pitch-circle diameter = 100 mm
- Static wheel load: 800 kg
- Sufficient package space for the wheel hub motor
- Mechanical interface for the brake disc
- Simulative validation based on industry standards

The first draft is based on a standard design for low-pressure casted aluminum wheels to show suitability in general and for mounting the brake disc. Since the finite element analysis (FEA) and the manufacturability can be evaluated positively, it enables an iteration, which is optimized for mechanical strength and design. With regard to the assessment of mechanical strength, rotary bending, rim rolling and impact tests are the industry standard. Rotary bending simulates cornering with high lateral loads, rim rolling covers straight driving and the impact test shall represent unwanted situations such as hitting a curb or a pothole. Fig. 15 shows as an example the results of the FEA for the back of the wheel. The highest stress in the simulations of the rotary bending is 136 MPa. As castable aluminum alloys can achieve yield strengths of more than 200 MPa, this results in a sufficient safety factor [12]. In addition, only moderate accelerations shall occur during regular operation of autonomous driving (see 2.1). In the impact tests, material deformations are acceptable, as these are no regular loads. However, no critical component failure should occur, as these situations cannot be completely avoided. The simulations were carried out for an impact between two spokes as well as for the impact directly on one spoke and show sufficient strength of the wheel. With the completed mechanical validation, the wheels can be manufactured in a low-pressure die casting process. They will be finished with a multi-color paint application, used at the project partner Maxion Wheels, to match the corresponding vehicle design.

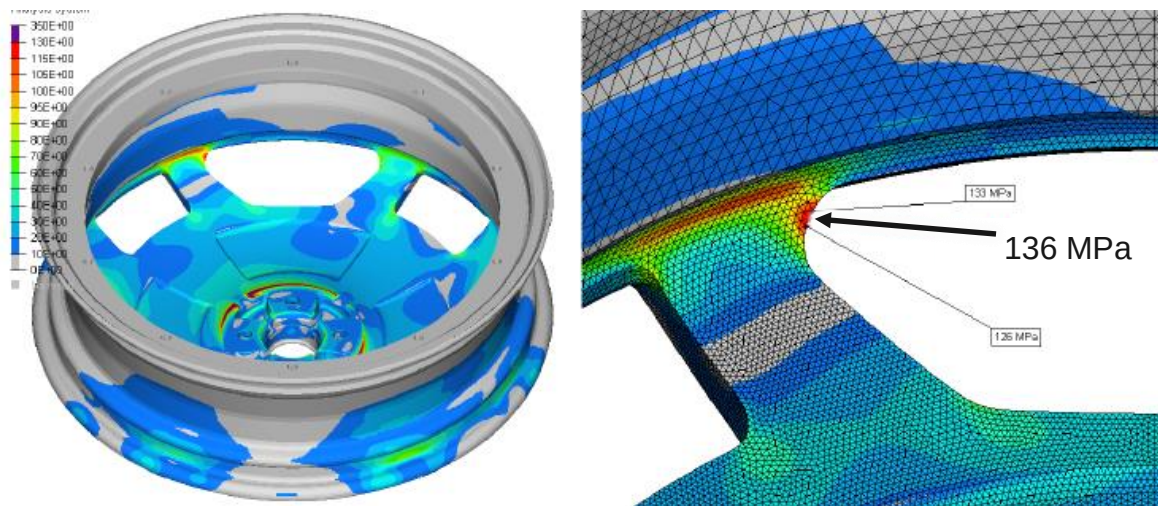


Fig. 15 Finite elements analysis for the iteration of the wheel

3 Control Unit and Inverter

A Modular concept is also used for the hardware design of the dynamics module. As illustrated in Fig. 16, the control devices of the dynamics modules are connected to form the spinal cord (SC). Each SC-module contains one control-unit, one interface-unit and one power-unit or inverter. Thanks to this modular concept, the SC-module provides a high flexibility regarding x-by-wire or advanced driver assistance systems. For instance, the same control-unit can be utilized to control different types of electric motors by modifying the interface-unit.

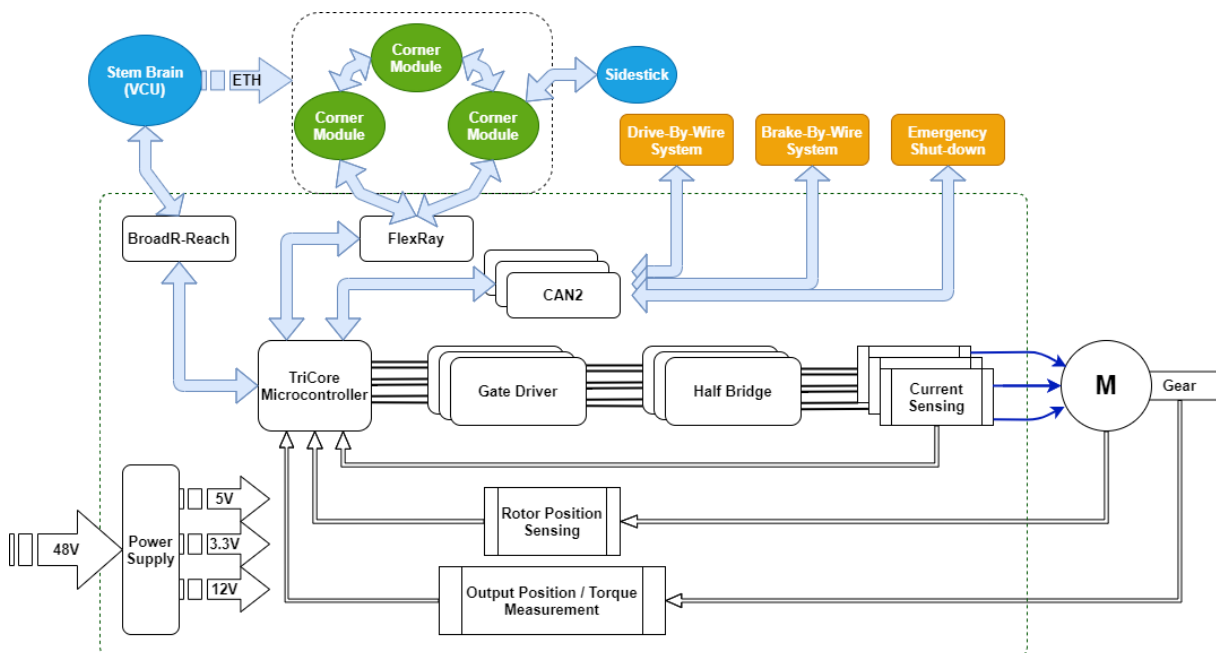


Fig. 16 Schematic layout of the dynamics module control device incl. interfaces

The control-unit is the central computing and communication unit of this modular concept. The use of an Aurix™ TriCore™ enables the simultaneous control of electric motors and safety applications with a single microcontroller. The whole control-unit

consists of one Aurix™ TriCore™, two FlexRay channels as well as two CAN channels. BroadR Reach is also included as a special feature. BroadR-Reach is an Ethernet physical layer standard designed for use in automotive applications.

As the name indicates, the interface-unit contains all interfaces of the internal sensors and the additional external communication interfaces as well as the entire power supply for itself and the control-unit. The gate drivers for the inverter-unit are also located on the interface-unit. A bidirectional serial synchronous (BiSS) interface is used to transfer the rotor position signal of the steering motor. BiSS is used because it offers high resolution, transfer rate, safety and flexibility. For redundancy and error detection, the angular position and torque at the gear output are also transmitted via BiSS.

The task of the power-unit or inverter, which is supplied with 48 V, is to provide current to the phases of the steering motor. The 48 V voltage level is already established in the automotive industry as a second voltage level or in electric vehicles as a third voltage level. It is intended to supply electrical components of more than 3 kW power, such as steering systems [13]. Compared to a 12 V system with the same power the necessary currents for a 48 V system are approx. 4 times lower. Thus, the losses are lower or thinner cables can be used. An aluminum printed circuit board (PCB) is selected for the inverter because of its good thermal properties. Due to the high thermal conductivity of aluminum, the high amount of locally generated waste heat of the components can be effectively dissipated, resulting in a homogeneous heat distribution. The thermal conductivity of aluminum cores with FR4 isolation varies between 1.0 and 8.0 W/mK depending on the design. Individually, the FR4-prepregs have a thermal conductivity of only 0.3 W/mK. In addition to the aluminum core in the PCB, an aluminum heat sink with cooling fins is used. Since the housing is also made of aluminum, as shown in Fig. 17, and cooling fins are also integrated in the sidewalls, high heat dissipation is ensured. The SC-modules are identical for each dynamics module and for every variant of the UNICARagil vehicles, so that they can be replaced independently from other modules in the vehicle.

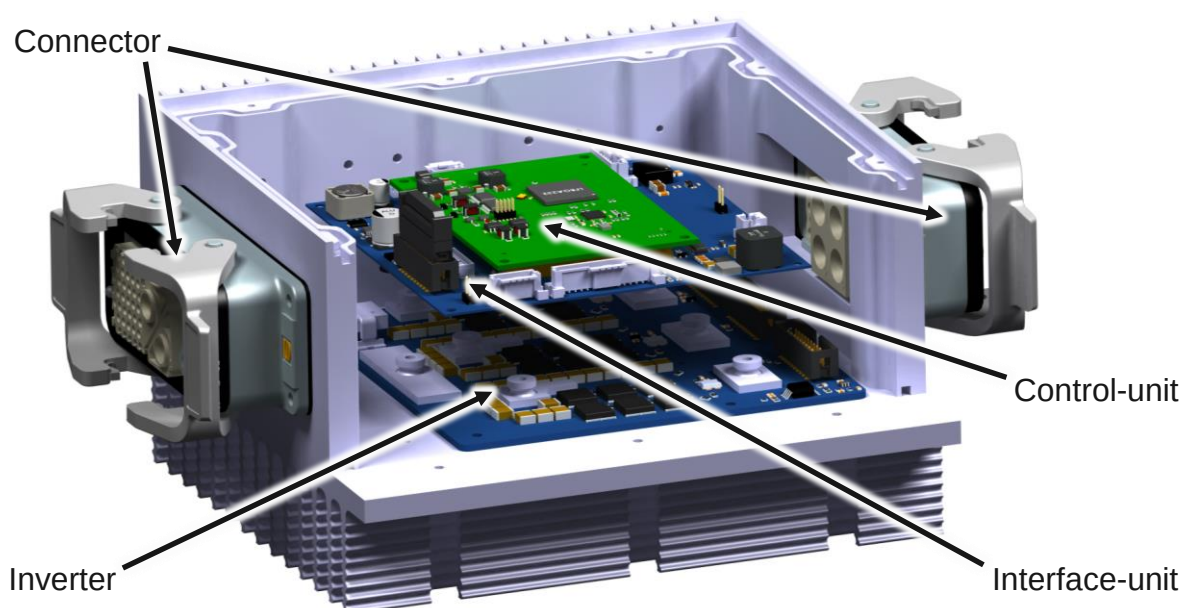


Fig. 17 Spinal cord module

4 Software and Functions

The modular character of the project also extends into the software structure. In this case, the software is divided into two parts. The first part is the signal conditioning together with the longitudinal and lateral controller implemented in Simulink. In signal conditioning, the input signals for the lateral and longitudinal controller are checked e.g. for plausibility and filtered if necessary. The controller code is compiled using Simulink code generation and can thus be integrated into the frame software. The frame software on the other hand is implemented in C code. As shown in Fig. 18, it provides the interface to the other control units and actuators of the dynamics module. In addition, monitoring and safety functions as well as the motor control of the steering actuator are implemented in C. Looking at the system as a whole, the following sequence results. The command variables of the vehicle dynamics control are pre-processed via signal conditioning and transferred to the lateral and longitudinal controllers. The specific values to be adjusted are then transmitted either to the steering motor control and thus to the steering actuator or via CAN bus to the wheel hub motor and the brake. As described in the previous chapter, the monitoring and safety functions are executed on a separate core of the Aurix™ TriCore™ to provide a higher level of safety. The overall vehicle safety concept is presented in [14].

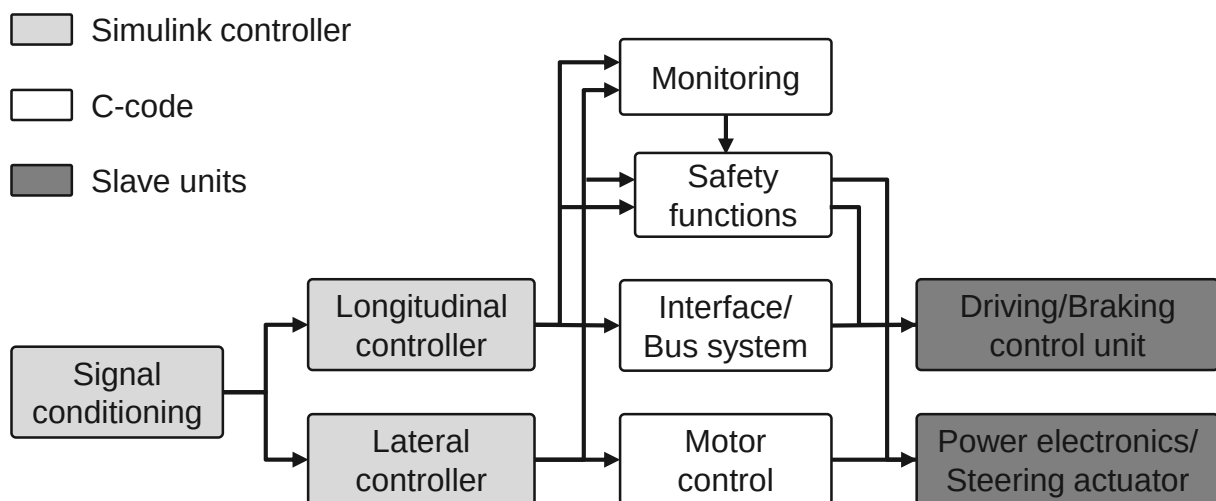


Fig. 18 Structure of the spinal cord software

5 Conclusions and Outlook

As described in this paper, the goal of the development of the dynamics module is to combine the functions of accelerating, braking and steering in one module. Agile methods should be used throughout the entire development process of the modular vehicles. In this context, the definition of requirements is based on the actual loads to be expected during testing and operation.

For the electric wheel hub motor, typical as well as maximum accelerations for passenger cars and in passenger transport are used for this purpose. In addition, gradients on possible driving routes are included in the dimensioning. The maximum acceleration

is the determining factor for the peak torque, whereas the permanently available torque is defined by the climbing ability. The executed simulations have shown that even the heaviest vehicle - the shuttle - can be operated with a 48 V drive system.

The presented chassis design enables the installation of all necessary components while at the same time realizing the desired steering angle range of 66° in one and 93° in the other direction. By adapting the design, it is furthermore possible to implement sufficient roll stabilization as well as wheel kinematics suitable for a steer-by-wire system.

Based on the chassis design the dimensioning of the steering motor is also done by considering realistic driving maneuvers. For this purpose, simulations of steering in standstill, full deceleration, steady-state cornering and single lane change are performed and discussed. It is found that steering in standstill and the full deceleration are decisive for the peak torque of the steering actuator, whereas the permanently available torque is defined by the steady-state cornering.

Similar to the chassis design, the main criterion for the wheel design also is to include all necessary components. Especially the connection of the brake disc of the perimeter brake to a passenger car wheel is a novelty. Here it is shown that a wheel produced in a low-pressure die casting process meets the requirements.

The final part of the paper explains the hardware and software structure of the SC-module, which is, indicated by the name, also modular. The more likely to be modified blocks signal conditioning as well as lateral and longitudinal controller are implemented in Simulink. In addition, the use of a multi-core microcontroller makes it possible to run safety and control algorithms on different cores, thus ensuring a high level of safety.

Based on the presented concept of the dynamics module, in the next phase the implementation of all functions and the application is carried out. In the first step, this includes the commissioning of the lateral and longitudinal controller and, on mechanical level, the assembly of the dynamics modules and the subsequent chassis measurement for the electric wheel alignment.

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