Redundant Steering System for Highly Automated Driving of Trucks

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Vorwort

Die vorliegende Arbeit entstand während meiner Tätigkeit als wissenschaftlicher Mitarbeiter am Fachgebiet Fahrzeugtechnik (FZD) der Technischen Universität Darmstadt. Die Inhalte dieser Dissertation resultieren aus einem Forschungsprojekt, das in Kooperation mit der Hubei Henglong Automotive System Group Co. Ltd durchgeführt wurde.

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List of Abbreviations

Abbreviation	**Description**
ACC	**A**daptive **C**ruise **C**ontrol
ACSF	**A**utomatically **C**ommanded **S**teering **F**unction
AD	**A**utomated **D**riving
AD2-	**A**utomated **D**riving with level of automation **2** (SAE J3016) and lower
AD3+	**A**utomated **D**riving with level of automation **3** (SAE J3016) and higher
ADAS	**A**dvanced **D**river **A**ssistance **S**ystem
ADS	**A**utomated **D**riving **S**ystem
aFAS	**a**utomatisch **f**ahrerlos fahrendes **A**bsicherungsfahrzeug für Arbeits**s**tellen auf Bundesautobahnen, engl. Automated Unmanned Protective Vehicle for Highway Hard Shoulder Road Works
AHPS	**A**ctive **HPS**
AM	**A**djustment **M**echanism
ASIL	**A**utomotive **S**afety **I**ntegrity **L**evel
AV	**A**ctive **V**alve
BLDC	**B**rush**l**ess **D**irect **C**urrent Motor
CoG	**C**enter **o**f **G**ravity
CPU	**C**entral **P**rocessing **U**nits
CSF	**C**orrective **S**teering **F**unction
CV	**C**ommercial **V**ehicle
DC	**D**irect **C**urrent
DDT	**D**ynamic **D**riving **T**ask
DoF	**D**egree **o**f **F**reedom
E	**E**rror
ECE	**E**conomic **C**ommission for **E**urope
ECU	**E**lectronic **C**ontrol **U**nit
E/E	**E**lectrical and/or **E**lectronic
EM	**E**lectric **M**otor
EPS	**E**lectric **P**ower **S**teering
EPS²	**R**edundant **EPS**
ESC	**E**lectronic **S**tability **C**ontrol
ESF	**E**mergency **S**teering **F**unction
F	**F**ailure
FA	**F**eedback **A**ctuator
FBS	**F**allback **S**ystem
FMEA	**F**ailure **M**ode and **E**ffect **A**nalysis
FSR	**F**unctional **S**afety **R**equirement

FTA	**F**ault **T**ree **A**nalysis
HARA	**H**azard **A**nalysis and **R**isk **A**ssessment
HPS	**H**ydraulic **P**ower **S**teering
HPS²	**R**edundant **HPS**
ICE	**I**nternal **C**ombustion **E**ngine
ID	**Id**entity
IEC	**I**nternational **E**lectrotechnical **C**ommission
iHSA	**I**ntelligent **H**ydraulic **S**teering **A**ssist
ISO	**I**nternational **O**rganization for **S**tandardization
LKS	**L**ane **K**eeping **S**upport
Lkw	**L**ast**k**raft**w**agen
LoA	**L**evel **o**f **A**utomation
MA	**M**odulation **A**ctuator
Nfz	**N**utz**f**ahr**z**eug
OC	**O**pen-**C**enter
ODD	**O**perational **D**esign **D**omain
OEDR	**O**bject and **E**vent **D**etection and **R**esponse
PE	**P**ower **E**lectronic
Pkw	**P**ersonen**k**raft**w**agen
PS	**P**ower **S**teering **S**tate
PSP	**P**ower **S**teering **P**ump
RASS	**R**edundant **A**ctive **S**teering **S**ystem
RC	**R**oad **C**lass
RCB	**Rec**irculating **B**all Steering Gear
SAE	**S**ociety of **A**utomotive **E**ngineers
SbW	**S**teer-**b**y-**W**ire
SCD	**S**afety **C**oncept **D**iagram
SG	**S**afety **G**oal
SOTIF	**S**afety **o**f **t**he **I**ntended **F**unctionality
StG	**S**teering **G**ear
SysML	**S**ystem **M**odeling **L**anguage
ToR	**T**ake-**o**ver **R**equest
ToT	**T**ake-**o**ver **T**ime
UML	**U**nified **M**odeling **L**anguage

List of Symbols and Indices

Symbol	Unit	Description
A	m²	Area
a	m/s²	Acceleration
b	Nm²	Bore constant
C	Ah	Battery capacity
c	Nm/rad	Stiffness
E	J	Energy
F	N	Force
h	m	Height
I	A	Electric current
i	-	Transmission ratio
J	kgm²	Moment of inertia
K_{M}	Nm/A	Torque constant of electric motor
k_{Valve}	-	Valve constant
ℓ	m	Length
M	Nm	Torque
m	kg	Mass
P	W	Power
p	N/m²	Pressure
r_0	m	Scrub radius
r_τ	m	Castor offset
Q	m³/s	Volume flow
t	s	Time
U	V	Voltage
v	m/s	Velocity
w	m	Width
δ	rad	Steering angle
$\dot{\delta}$	rad/s	Steering angle velocity
$\ddot{\delta}$	rad/s²	Steering angle acceleration
κ	1/m	Curvature
λ	-	Slip
μ	-	Friction coefficient
φ	rad/s	Twisting angle of steering valve
ρ	kg/m³	Density
σ	rad	Kingpin angle
τ	rad	Castor angle
γ	rad	Camber angle

Index	Description
active	Active hybrid state
AV	Active valve
Batt	Battery
Bore	Bore torque during steering
C	Coupling mass
D	Damping
DI	Driver intervention
dry	Dry road surface
dyn	Dynamic
e	Electric
eff	Effective Inertia
EM2H	Electric motor to steering wheel
F	Friction
f	Front
fa	Front axle
FB	Feedback
fl	Front left
fr	Front right
H	Steering wheel, handle
h	Hydraulic
H2P	Steering wheel to pitman arm
H2W	Steering wheel to wheel
high	High
ICE	Internal Combustion Engine
J	Inertia
loss	Loss
low	Low
max	Maximum
min	Minimum
n	Nominal
OR	Overriding
OS	Operational steering requirements
P	Pitman arm
P2W	Pitman arm to wheel
Piston	Hydraulic piston of HPS
R	Rack
r	Rear
RR	Redundancy steering requirements
Servo	Servo steering
sb	Stand-by hybrid state

Switch	Threshold for state switch
Tread	Tire tread
V	Vehicle
VS	Valve sleeve
VTB	Valve torsion bar
W	Wheel
wet	Wet road surface
x	Surge DOF (corresponds to vehicle longitudinal direction)
y	Sway DOF (corresponds to vehicle lateral direction)
z	Heave DOF (corresponds to vehicle vertical direction)

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Kurzzusammenfassung

Die Entwicklung des automatisierten Fahrens (AD) vom teilautomatisierten Fahren (AD2-) hin zum hochautomatisierten (AD3+) ist nicht nur im Fokus der Personenkraftwagen (Pkw) – Industrie, sondern auch Schwerpunkt in der Nutzfahrzeug (Nfz) – Industrie, insbesondere bei der Entwicklung von Lastkraftwagen (Lkw). Während es für Pkw bereits eine Vielzahl an Forschungsarbeiten zu AD3+ gibt, besteht auf diesem Gebiet für Lkw besonders im Bereich der für AD3+ geeigneten Lenksystemen noch großer Forschungsbedarf, da die Anforderungen an diese, besonders hinsichtlich der geforderten maximalen Lenkkräfte und Lenkleistungen, bei AD3+ Lkw um ein Vielfaches höher sind als bei AD3+ Pkw.

Gegenstand dieser Dissertation ist daher, basierend auf den Rahmenanforderungen für Lkw-Lenksysteme hinsichtlich Bauraum, Schnittstellen, Energieversorgung und Achslasten sowie auf den in dieser Arbeit ermittelten Betriebs- und Redundanzanforderungen durch eine deduktive Methodik und einem systematischen Durchgehen des Lösungsraumes ein Konzept eines aktiven Lenksystems für AD3+ Lkw zu entwickeln. Eine redundante rein elektromechanische Servolenkung fällt aufgrund der zu geringen zur Verfügung stehenden elektrischen Leistung und eine redundante hydraulische Servolenkung aus Effizienzgründen aus dem Lösungsraum heraus. Dieser beschränkt sich mit heutigen Aktoren auf kombinierte elektromechanisch-hydraulische Servolenkungen, sogenannte hybride Lenkungen, wofür mögliche unterschiedliche Funktionsstrukturen hergeleitet und basierend auf Anforderungen aus einer Sicherheitsanalyse bewertet werden. So wird der Lösungsraum eingegrenzt.

Das erarbeitete Konzept, welches alle Anforderungen erfüllt, ist ein Redundantes Aktives Lenksystem (engl. redundant active steering system – RASS) mit einem elektromechanischen Subsystem und einem hydraulischen Subsystem, welches ein sowohl durch den Fahrer als auch durch ein elektrisches Signal regelbares hydraulisches Lenkungsventil besitzt. Das RASS stellt eine sogenannte „fail-degraded“ Funktionalität dar, deren Degradationsgrad durch die ermittelten Redundanzanforderungen bestimmt wurde. Das doppelt regelbare Lenkventil wird so konzipiert, dass eine Übersteuerbarkeit der Automatik durch den Fahrer zu jeder Zeit gewährleistet und eine innerhalb der Momenten- und Leistungsgrenzen des elektrischen Subsystems vollkommen freie Aufteilung des geforderten Lenkmomentes auf das elektrische und das hydraulische Teilsystem ermöglicht wird. Diese Funktionalität ist zur Effizienzsteigerung gegenüber herkömmlichen Lkw-Lenksystemen nutzbar.

Für die verschiedenen Systemzustände des RASS wird eine Betriebsstrategie entwickelt, welche unter Berücksichtigung des Fahrerzustandes, der geforderten Lenkmomente sowie möglicher Systemausfälle, den Servolenkungszustand so steuert, dass sich die Effizienz des Lenksystems steigert, die Wechsel zwischen manuellen und automatisierten Fahren regelt und im Fehlerfall Rückfallstrategien bereitstellt. Als Ergebnis liegt ein neuartiges Lenkkonzept vor, welches alle Anforderungen heutiger Lkw erfüllt und für AD3+ geeignet ist.

Summary

The development of automated driving (AD) from partially automated driving (AD2-) to highly automated driving (AD3+) is not only in the focus of the passenger car industry, but also in the commercial vehicle (CV) industry, especially in the development of trucks. There is already a lot of research work on AD3+ for passenger cars. However, in this area there is still a great need for research for trucks, particularly in the area of steering systems suitable for AD3+, since the requirements of these, especially with regard to the maximum required steering forces and steering powers, are much higher for AD3+ trucks than for AD3+ passenger cars.

Therefore, the subject of this thesis is to develop a concept of an active steering system for AD3+ trucks by means of a deductive methodology and a systematic analysis of the solution space. The development is based on the frame requirements for truck steering systems with regard to assembly space, interfaces, energy supply and axle loads as well as on the operational and redundancy requirements determined in this thesis. On the basis of these requirements, a redundant electric power steering system is excluded from the solution space due to the insufficient electrical power available on board and a redundant hydraulic power steering system for efficiency reasons. With today's actuators, the solution space is limited to combinations of electric and hydraulic power steering, the so-called hybrid steering systems, for which the possible different functional structures are derived. These are evaluated on the basis of requirements from a safety analysis, whereby the solution space is limited.

The developed concept, which meets all requirements, is a redundant active steering system (RASS) with an electric subsystem and a hydraulic subsystem, which is equipped with an active steering valve that can be controlled by the driver as well as by an electrical signal. The RASS provides a so-called "fail-degraded" functionality whose degree of degradation was determined by the determined redundancy requirements. The double controllable steering valve is designed in such a way that the driver is able to override the automatic system at any time and that the required steering torque can be distributed arbitrarily between the electric and the hydraulic subsystem within the torque and power limits of the electric subsystem. This functionality is usable to increase efficiency compared to conventional truck steering systems.

An operating strategy is developed for the various system states of the RASS which, taking into account the driver's state, the required steering torques and possible system faults, controls the power steering state in such a way as to increase the efficiency of the steering system, controls the transitions between manual and automated driving and provides fallback strategies in the event of a fault. The result is an innovative steering concept that meets all the requirements of today's trucks and is suitable for AD3+.

1 Introduction

1.1 Motivation

Driver assistance systems and automated driving (AD) gain more and more importance in the automotive development. This trend takes place not only in the development of passenger cars, but also in the development of commercial vehicles including trucks.

Advanced driver assistance systems (ADAS) are already available for trucks today. Examples of ADAS for trucks are Adaptive Cruise Control (ACC), Lane Keeping Support (LKS), Automated Emergency Braking (AEB) or side wind compensation. Whereas ACC only controls the longitudinal movement of the vehicle, the other two support the driver in lateral guidance.[1a,2]

Different partially automated driving systems have recently been demonstrated on public roads in prototypical trucks as well, such as exit-to-exit automation, traffic jam assist, Platooning or automated trailer backing.[3] Examples for exit-to-exit automation are the Highway Pilot and the Interstate Pilot. Both systems are a combination of ACC and LKS, but intervene far more intensely, hence the longitudinal and the lateral guidance are performed by the automation system during monotonous driving on highways or during a traffic jam. However, the driver is responsible and has to be available to take over the control at all times if a critical traffic situation occurs, e.g. construction zones.[4,5] An advancement of such systems is the so-called "platooning", where two or more trucks connect each other to a convoy with a 15 m gap between the vehicles via data communication. This use case aims to reduce the required traffic space, increase safety and reduce fuel consumption.[4] The Smart Truck Maneuvering is exemplary for an automated trailer backing system. It enables the driver to maneuver the truck with one or more trailers from outside with a remote tablet.[1b] The Freightliner Inspiration Truck, based on the Freightliner Cascadia, was the first partially automated

[1] Gaedke, A. et al.: Driver assistance for trucks (2015), (a) p.221 | (b) p.225.

[2] Brunner, G.; Negele, K.: Electrification of the steering (2008), p.12.

[3] Engström, J. et al.: Deployment of Automated Trucking (2019), p.150.

[4] Ballarin, C.: The truck on its way to an autonomous means of transport (2016), p.40.

[5] Flämig, H.: Autonomous vehicles and autonomous driving in freight transport (2016), p.373.

driving truck that achieved a road approval in the US state of Nevada in 2015.[6] Other prototypes of automated driven trucks were developed and tested by companies like Otto[7] or Embark[8].

These demonstrated applications already show the positive effects of automated driving, which are the reduction of accidents, of emissions, of costs and of the driver's stress as well as the improvement of traffic flow. Hence, the benefits of automated driving are significant for all stakeholders of trucking industry.[9,10] Therefore, the development is going further towards highly and fully automated trucks, which will increase the named benefits of automated driving even more and will release the driver from the driving task, so that he is able to fulfill other tasks, e.g. transport management.[11,12] The project aFAS (German for Automated Unmanned Protective Vehicle for Highway Hard Shoulder Road Works) has proven the technical feasibility of a driverless and fully automated driven vehicle on public roads for the first time. The result was a prototypical truck, which was able to drive fully automated up to 10 km/h on the hard shoulder of the German highways. Compared to ADAS and partially automated systems, the biggest distinction of highly and fully automated systems is the driver who is not available as a fallback level for the automated system. This increases especially the safety requirements for all necessary vehicle systems, which makes it necessary to develop new appropriate systems.[13]

The standard architecture of highly automated operated vehicles requires sensor technology for positioning, perception of the environment, sense of the vehicle's and of the driver's state, electronic control units (ECU) for data fusion and processing, and actuators for the realization of the intended driving maneuvers. The software interprets the sensor data and plans the trajectory which is controlled by the actuators to execute the correct maneuvers. The actuators include the actuators for longitudinal control such as brake, engine and gearbox control, but also the actuator for the lateral control, which is the steering system.[14] The electronic controllability of the different actuators is a mandatory property of the actuators for automated driving. Whereas the brakes as well as the engine and the gearbox are already electronically controllable in today's heavy-duty trucks, their steering systems are still almost solely hydraulic power steering (HPS) systems, without electronic controllability.

[6] Ballarin, C.: The truck on its way to an autonomous means of transport (2016), p.37.

[7] Freedman, D. H.: Self-Driving Trucks.

[8] Embark - Self-Driving Semi Trucks (2019).

[9] Flämig, H.: Autonomous vehicles and autonomous driving in freight transport (2016), pp. 373–374.

[10] Engström, J. et al.: Deployment of Automated Trucking (2019), p.150.

[11] Brockmann, S.; Schlott, S.: The long way to autonomous truck driving (2015), p.11.

[12] Kirschbaum, M.: Highly automated driving for commercial vehicles (2015), p.6.

[13] Stolte, T. et al.: Towards Automated Driving (2015), p.672.

[14] Cacilo, A. et al.: Highly automated driving on highways (2015), p.47.

Electric power steering (EPS) systems are ideally feasible for the use as automated steering systems, since the electric motor is easily controllable by the software of their ECU. However, due to the high demanded steering torque and steering power of heavy-duty trucks up to 6 kW, a EPS system on its own is not feasible with the design of current vehicle's power supply. Therefore, one approach of automated steering systems for heavy-duty trucks is to add an additional electrically controllable actuator, like an electric motor, to the hydraulic steering system, which is able to overlay steering torque to the steering system independently of the driver via a torque overlay gear. Those systems are called Hybrid Steering Systems because of their two different sources of steering torque.[15,16a]

An additional advantage of such hybrid steering systems is the potential to reduce the energy demand of the steering system. This is important especially for long haul trucks, since the fuel costs make up more than a quarter of overall costs of transportation. Only about 1 % of the applied power is used as mechanical steering power. The high losses are caused by the hydraulic power steering pump (PSP), which permanently produces an engine speed dependent volume flow, which has to run through the open-center (OC) steering valve.[16] Because solely electric steering is feasible during most of the occurring driving situations of trucks, the use of an additional EPS system combined with a PSP with a variable volume flow can make fuel savings up to 0.56 l/100km possible.[17,16b]

1.2 Particularities of Trucks Compared to Passenger Cars

Heavy-duty trucks significantly differ from passenger cars in several properties. For the development of a steering system, especially for the determination of the requirements for the steering system, it is important to know these differences. From now on, the name truck is used in the present thesis as a synonym for so-called class 7 trucks or higher classes with a vehicle weight over 11.8 t[18] respectively category N3 vehicles with a vehicle weight exceeding 12 t[19]. In the following sections, differences of trucks and passenger cars in general vehicle properties, e.g. chassis, dimensions and electrical and/or electronic (E/E)-systems, are discussed first. The different steering system of trucks is described in the second part.

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[15] Reimann, G. et al.: Steering Actuator Systems (2016), pp. 757–758.

[16] Wiesel, U. et al.: Hybrid steering system for reducing fuel consumption (2010), (a) pp. 63–64 | (b) p.68.

[17] Brunner, G.; Negele, K.: Electrification of the steering (2008), p.13.

[18] Hallenbeck, M. E. et al.: Vehicle Classification Rules (2014), p.13.

[19] European Union: Framework for the approval of motor vehicles (2007), p.62.

1.2.1 General Vehicle Properties

The different properties of trucks compared to passenger cars lead to different requirements for the truck's systems, such as for the steering system. The most obvious difference are the dimensions of a truck. The mass of a truck is up to 40 times the mass of a passenger car, they are up to four times longer and two times as wide as cars.[20]

Although trucks often have more axles than passenger cars, the higher mass also leads to up to 15 times higher wheel and axle loads and thus to much higher tire contact forces. Together with the higher tire pressure of trucks, usually between 6 and 8 bar, and a wear-optimized tire design this leads to lower maximum friction values and to lower maximum deceleration of approximately 7-8 m/s^2 in consequence. The bigger height of trucks of up to 4 m and the resulting higher center of gravity of 1.2 to 2.5 m leads to an earlier roll over of trucks, which is why the maximum possible lateral acceleration of trucks is in the range of 4 to 6 m/s^2.[21]

In addition to the differences concerning dimensions, there is also a big difference in the operating hours and the environmental conditions. The service life up to 1,500,000 km and up to 50,000 h of operation are three to five times higher compared to passenger cars, which leads to significantly higher requirements for the truck components. Among other things, typical truck chassis are therefore based on a ladder-type frame construction with rigid axles.[21]

The electrical power supply of trucks is a direct current (DC) voltage network with 24 V and a typical battery capacity is about 220 Ah for a truck with high electric energy demands. Due to the permanently increasing requirements to the power network, the introduction of a 48 V power network is discussed consistently. However, due to the widespread availability of the 24 V components and devices, which not exists for 48 V, a near-term switch to 48 V seems not realistic.[22]

All these differences in the general vehicle design and properties substantially effect the design of truck steering systems, as described in the following.

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[20] Shladover, S. E. et al.: Demonstration of automated heavy-duty vehicles (2006), 1.8.

[21] Hecker, F.: Brake-Based Stability Assistance Functions (2016), pp. 1023–1026.

[22] Hilgers, M.: Electrics and Mechatronics (2016), pp. 5–6.

1.2.2 Truck Steering System

This section describes the differences and characteristics of current truck steering systems. This includes the steering axle kinematics, the steering geometry and the actual hydraulic steering system, which is responsible for the steering torque generation to assist the driver.

Today's trucks are mostly equipped with a servo hydraulic recirculating ball (RCB) steering, which transmits the input steering torque of the driver amplified by the hydraulic assistance via a related linkage to the wheels.[23] A schematic sketch of a HPS for trucks is shown in Figure 1-1 and the corresponding structure is illustrated in Figure 1-2. The input of the steering system is the steering wheel (1), where the driver applies torque to the system. The steering column transmits this torque through the hydraulic valve (2) to the input shaft of the steering gear (3), which transforms it to the output torque at the pitman arm (4). The driver's torque applied at the hydraulic valve controls the amount of hydraulic steering assistance. The internal combustion engine (ICE, 7) drives the Power Steering Pump (PSP, 8), which supplies the valve with hydraulic oil from the tank (9). The steering gear is a RCB gear with an integrated hydraulic piston, whereby it adds the hydraulic power to the mechanical power from the driver. A more detailed description follows. The pitman arm transforms the output torque of the steering gear to a steering force at the push rod (5), which transmits it to the steering arm. The steering arm is connected to the left steering knuckle. The steering power is transferred via the knuckle arms and the tie rod (6) to the right steering knuckle.[24] The steering knuckles are connected to the axle beam of the rigid axle.

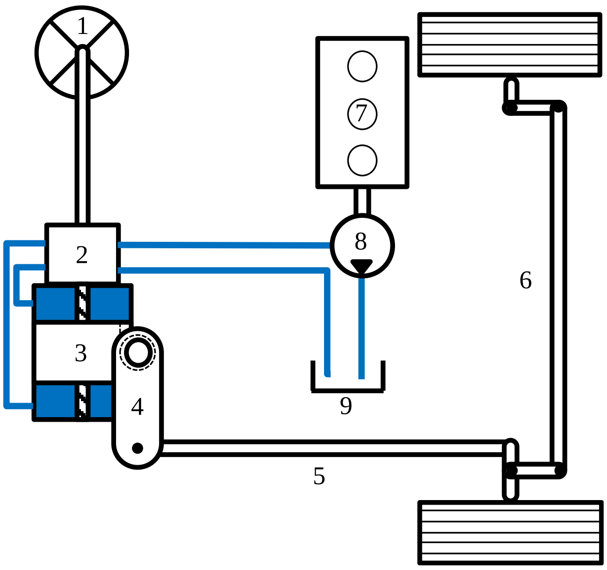

Figure 1-1: Schematic sketch of a HPS for trucks

[23] Hecker, F.: Brake-Based Stability Assistance Functions (2016), p.1023.

[24] For left-hand drive vehicles.

This steering kinematics in combination with a RCB steering gear are used in almost every vehicle with a rigid front axle. With this structure, it is possible to mount the steering gear to the frame and thus the steering system is able to follow the complex spatial movements of the rigid axle without much backlash on the steering system. Also, the connection to the steering wheel inside the truck's cab, which has an additional suspension relative to the frame, is easier with this steering layout.[25]

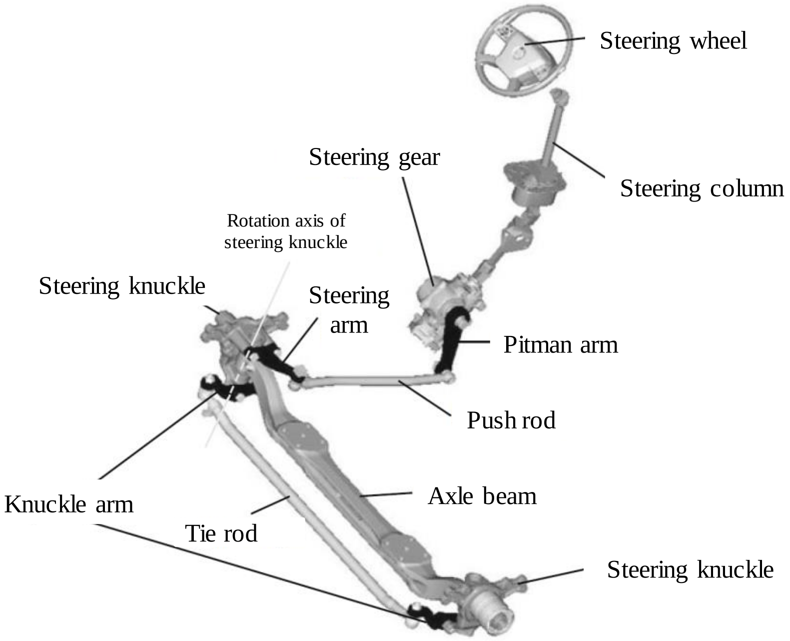

Figure 1-2: Exemplary truck steering system with steering kinematic and RCB steering gear[26]

In the following thesis, the start of the torque or energy flow is the steering wheel and the end are the wheels. The order of components in this flow is considered, when talking about torque before or behind a special component.

Figure 1-3 illustrates the functional structure of a HPS. The driver's input torque (M_{H}) and input angle (δ_{H}) at the steering wheel are transferred through the torsion bar (TB) to the steering gear (StG). The hydraulic force (F_{h}) is added to the driver's torque within the steering gear as described hereafter. The output torque and angle of the steering gear at the pitman arm ($M_{\mathrm{P}}, \delta_{\mathrm{P}}$) are transmitted to the steering torque and angle at the wheels ($M_{\mathrm{W}}, \delta_{\mathrm{W}}$) by the steering linkage. The amount of hydraulic force is controlled by the torsion angle (φ_{d}) of the torsion bar induced by the driver. The PSP driven by the truck's ICE generates the hydraulic volume flow (Q_{h}) and high pressure (p_{high}), which is applied on the hydraulic piston. The tank is the reservoir with a low hydraulic pressure (p_{low}).

[25] Hullmann, J. et al.: Mechanical and Hydraulic Gears (2017), p.334.

[26] Following Hilgers, M.: Chassis and Axles (2016), p.26.

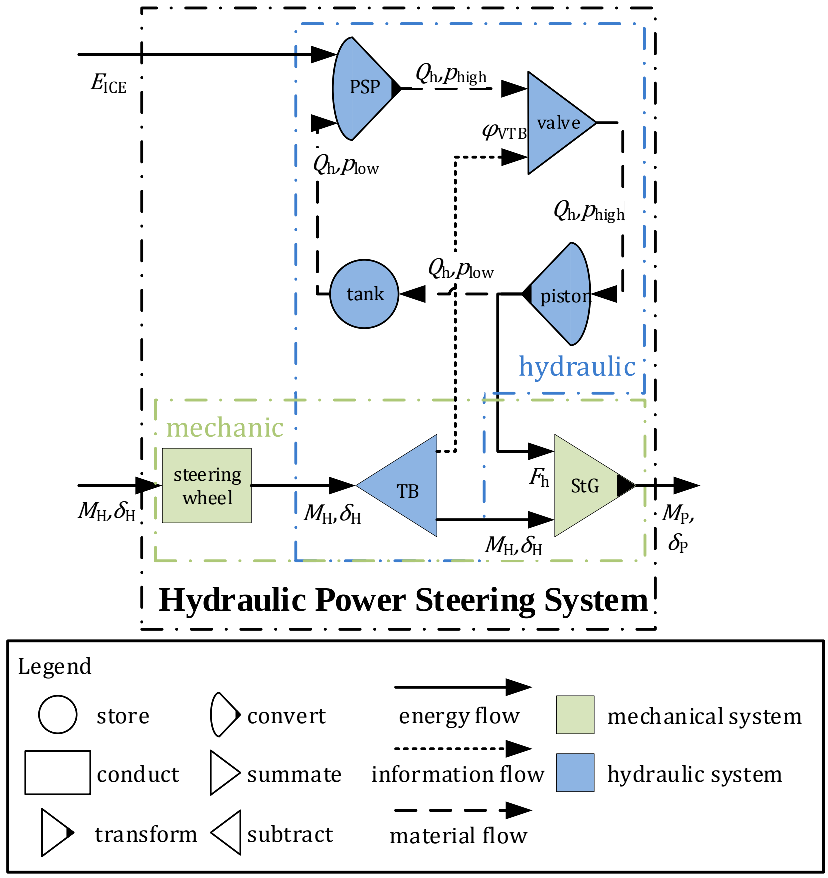

Figure 1-3: Functional structure of hydraulic power steering (HPS) system

The hydraulic system is combined with the RCB gear in a so-called integrated module. A cutaway view of such a module is shown in Figure 1-4. The input shaft of the steering gear and the connected rotary slide valve are similar to those of a hydraulic rack-and-pinion steering, but are designed for much higher volumetric flows rates and higher hydraulic pressures, compliant with the higher required power of the RCB gear. The torsion bar inside the rotary slide valve connects the input shaft with the spindle of the RCB gear. The balls transform the rotation of the spindle to a lateral movement of the ball nut, which also functions as the piston of the hydraulic cylinder. The two chambers of the cylinder are connected to the corresponding ring grooves of the valve. A dovetailing transfers the translation of the ball nut to the output shaft of the steering gear, which is connected to the pitman arm outside of the gear housing. This steering gear design offers several benefits for the use in trucks. It is more robust than rack-and-pinion steering systems and very reliable and thus it obtains a long service life of about one million kilometers. Moreover, it has a better damping against external impacts and is more compact compared to a rack-and-pinion steering. However, the drawbacks of the RCB steering gear are the high weight, the high costs and the indirect steering feel.[27]

[27] Hullmann, J. et al.: Mechanical and Hydraulic Gears (2017), pp. 333–334.

The core component of the truck's HPS system concerning the control of the hydraulic power and the steering feel is the rotary slide valve. Other types of valve designs have been fully replaced by this valve.[28a] Its structure and its function are described in more detail below.

Its mechanical function is to transmit the steering torque from the input shaft via the torsion bar to the valve's output shaft, which is connected to the spindle of the RCB gear. The hydraulic functions of rotary slide valve are to connect the two chambers of the hydraulic cylinder with the inlet from the PSP and the runback to the hydraulic tank as well as to control the hydraulic power according to the input torque from the driver at the steering wheel. The torsion bar fixes the input shaft and the output shaft axially, but allows a relative twist between them as a function of the input torque. Thereby, the amount and the direction of the hydraulic power assistance are controlled. The stiffness of the torsion bar mainly influences the ease and the feeling of the steering system.[28b]

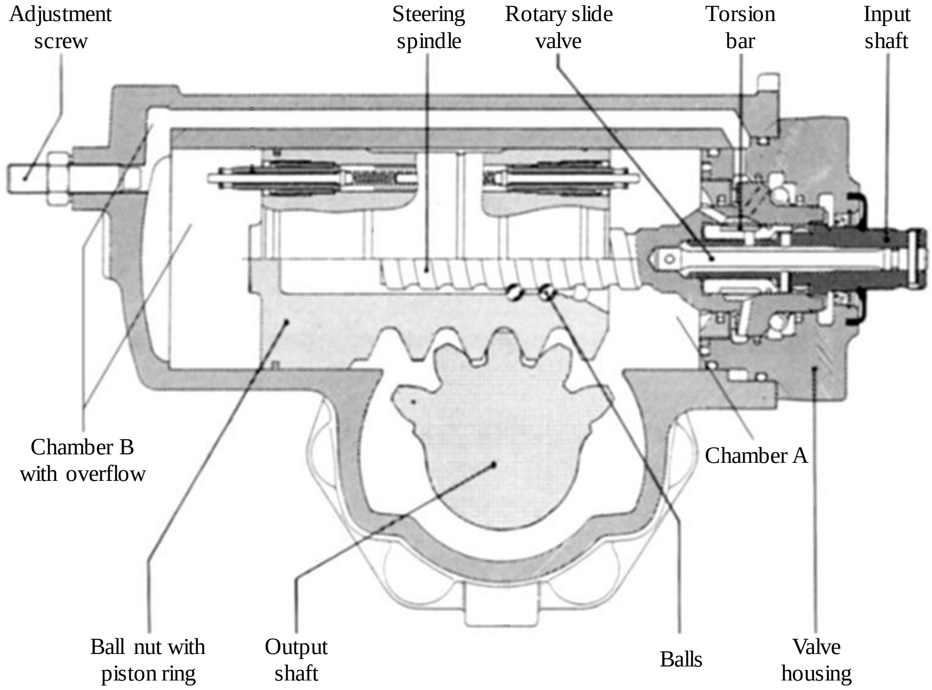

Figure 1-4: Cutaway view of RCB steering gear with integrated hydraulic piston[28c]

Figure 1-5 shows the hydraulic diagram and a cutaway view of a rotary slide valve. The two main function parts of the valve are the input shaft and the valve sleeve. The valve sleeve is solid connected with the spindle of the RCB gear, but can be rotated relatively to the input shaft within certain limits. The hydraulic outputs of the valve sleeve at its outside are hydraulically connected with three grooves in the valve housing, which creates three chambers. The axial grooves inside of the valve sleeve are spaced over the circumference and implemented at a certain distance from the sleeve's end. They interact with the grooves of the input shaft and thus adopt the valve function. The hydraulic counterpart is the input shaft of

[28] Hullmann, J. et al.: Mechanical and Hydraulic Gears (2017), (a) pp. 306–312 | (b) pp. 333–334 | (c) p. 332.

the valve. Besides by the torsion bar, it also transmits the input torque directly to the spindle if the valve is fully twisted and hits the mechanical end stop. There are axial grooves on the high-precision cylindrical outer diameter of the shaft with exact so-called control edges, which in combination with the grooves inside the valve sleeve influence the characteristics of the power assistance curve over the steering wheel torque. The runback from the hydraulic cylinder flows back through the hollow input shaft. The rotary slide valve hydraulically functions like a 4/3-way proportional valve with an open center (OC). OC means that even in the center position of the valve the PSP transports a continuous hydraulic volume flow through the valve. In neutral position the hydraulic resistance is least. An increasing twist of the valve causes an increasing resistance and an increasing dynamic pressure. Hence, the pressure inside of one cylinder chamber rises and causes a hydraulic steering assistance.[29a]

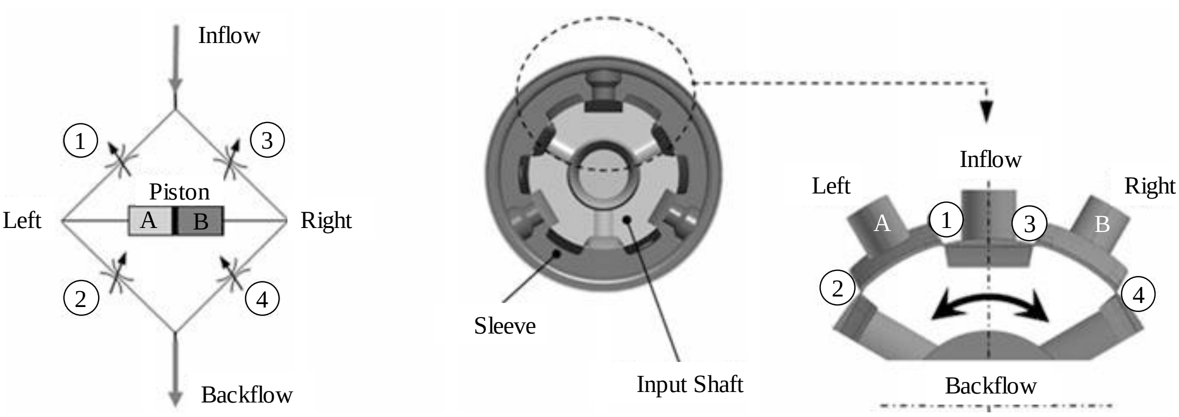

Figure 1-5: Hydraulic diagram and cutaway view of a rotary slide valve[29b]

For trucks, the hydraulic power is generated almost exclusively by a PSP driven by the engine. Whereas the radial piston pump was used in trucks as well earlier, today the vane pump is the mostly built-in pump type. The PSP is designed to generate the maximum required steering power already at idling speed of the engine, because the maximum steering power is required during steering at standstill. This design causes high losses, e.g. during highway driving with high engine speeds and low required steering power. Hence, different new designs of vane pumps are used today to decrease the hydraulic losses. Examples for such pumps are the vane pump with a bypass, which lowers the circulation pressure when the steering is passive, or the variable pump, which has a variable capacity and thus generates only as much volume flow as required for steering.[29c]

The PSP is not in focus of this thesis. The existence of pumps with a variable hydraulic volume flow is required, but their detailed functionality is not relevant here.

The description of the standard steering system in today's trucks clarifies the functionality of the HPS, but also the lack of a possibility to control the steering system independently of the driver. The HPS is called a passive power steering system, which requires the driver's steering wheel torque as input for the control of the steering valve.

[29] Hullmann, J. et al.: Mechanical and Hydraulic Gears (2017), (a) pp. 306–312 | (b) p. 331 | (c) pp. 362–364.

1.3 Scope of the Thesis

The initial subchapters present the motivation for automated trucks and their benefits for the different stakeholders. This thesis focuses on the target application of highly automated driving of trucks, which differs from partially automated driving primarily in the sense that the driver does not permanently observe the automated system and therefore is not available as an immediate fallback level. Partially automated driving is not considered here.

The scope of this thesis includes the hardware of the steering system, its functional safety assessment and strategies for its operation. The hardware is considered starting from the steering wheel as the driver’s input, over the steering gear with all its actuators and its electric power supply, to the pitman arm as the interface to the truck’s steering kinematics. Since E/E-systems are being introduced into the truck’s steering system to be able to generate steering torque independently of the driver for the application of automated driving, functional safety plays a decisive role in the development of the new steering concept. An increased functionality of the steering system is required for highly automated driving as well. Thereby, the development of operation strategies for this application is also necessary within this thesis.

Not in the scope of this thesis are the hydraulic power supply by the PSP, the steering kinematics of the truck and a detailed software development. Within this thesis, it is assumed that a PSP with a variable volume flow is available to supply the steering system with variable hydraulic power. The steering kinematics are left out because the target steering system should be designed in such a way that it can be integrated into current trucks without the need for completely new steering kinematics. The operation strategies for the steering system are only developed on a functional level. The detailed software implementation is not considered here.

It is important to define what is assumed in this work to be the input signal of the steering system during automated driving. An output steering torque requirement of the overall steering system is used as input signal for the target steering system. Similar to manual driving, where the driver’s torque at the steering wheel serves as the input, the steering system is torque-controlled during automated driving as well. The steering angle control is the task of the controller for lateral guidance, but is not within the scope of this thesis.

2 Analysis of the State of the Art and the Scientific Research

The current state of the art and the current state of scientific research concerning steering systems for highly automated driving, especially of trucks, is summarized in the following subchapters. In order to form a basis for the discussion of the requirements for highly automated driving, the different levels of driving automation and different fallback levels and redundancy strategies are explained at first. The second part gives a short overview of the functional safety standard, of the legal requirements concerning steering systems, especially for automated driving, and of different fallback levels and redundancy strategies. Descriptions of different approaches for active steering systems for trucks, which present the current state of the art of truck steering systems, follow.

2.1 Levels of Driving Automation

In order to be able to develop and discuss requirements for a steering system for highly automated driving and to design such a steering system it is necessary to know several terms, definitions and levels regarding the driving automation. SAE J3016[30a] addresses these topics and gives a taxonomy for driving automation.

The term driving automation includes all levels, shown in Table 2-1, and is defined as "the performance of part or all of the dynamic driving tasks (DDT) on a sustained basis."[30b] Systems that perform "the entire DDT on a sustained basis, regardless of whether it is limited to a specific operational design domain (ODD)"[30b] belong to the levels 3, 4 and 5 of driving automation and are summarized as highly automated and abbreviated with AD3+ in the following thesis.

The DDT consists of the sustained lateral and/or longitudinal control of the vehicle as well as the objective and event detection and response (OEDR). The latter task is not in focus of this project, but the DDT fallback is important for the development of a steering system for AD3+. The fallback is defined as the "response by the user or by an automated driving system (ADS) to either perform the DDT or achieve a minimal risk condition after occurrence of a DDT performance-relevant system failure(s) or upon ODD exit"[30b]. For level 1 and 2 systems, summarized with AD2-, the driver is the fallback level and has to be available to take over the control immediately at any time. To be able to do this, the driver always has to supervise the system. For level 3 systems, the driver becomes the "fallback-ready user" when the ADS is engaged, hence the driver doesn't have to supervise the ADS but has to be receptive to intervene if requested and in the case of evident system failures which compel him

[30] Society of Automotive Engineers: J3016 Terms related to driving automation (2016), (a) - | (b) pp. 3–6.

to take over the DDT. In case of a fault of the automated steering system, this means that it still has to work automated for a short time until the driver intervenes. For levels 4 and 5, the driver does not have to be available anymore for the fallback task of the DDT. Hence, the automated system itself has to fulfill this task with integrated fallback levels, such as a steering redundancy.

Table 2-1: Levels of driving automation[31]

	Level	**Name**	**Narrative definition**	**DDT**		**DDT fallback**	**ODD**
				Sustained lateral and longitudinal vehicle motion control	**OEDR**		
	Driver performs part or all of the DDT						
	0	No Driving Automation	Performance by driver of entire DDT, even when enhanced by active safety systems.	Driver	Driver	Driver	n/a
AD2-	1	Driver Assistance	Sustained and ODD-specific execution by a driving automation system of either lateral or longitudinal vehicle motion control subtask of DDT (but not both simultaneously) with expectation that driver performs the remainder of DDT.	Driver and system	Driver	Driver	Limited
AD2-	2	Partial Driving Automation	Sustained and ODD-specific execution by a driving automation system of both lateral and longitudinal vehicle motion control subtasks of DDT with expectation that driver completes OEDR subtask and supervises driving automation system.	System	Driver	Driver	Limited
	ADS ("System") performs the entire DDT (while engaged)						
AD3+	3	Conditional Driving Automation	Sustained and ODD-specific performance by ADS of entire DDT with expectation that DDT fallback-ready user is receptive to ADS-issued requests to intervene, as well as to DDT performance-relevant system failures in other vehicle systems, and will respond appropriately.	System	System	Fallback-ready user (becomes the driver during fallback)	Limited
AD3+	4	High Driving Automation	Sustained and ODD-specific performance by ADS of entire DDT and DDT fallback without any expectation that a user will respond to a request to intervene.	System	System	System	Limited
AD3+	5	Full Driving Automation	Sustained and unconditional (i.e., not ODD-specific) performance by ADS of entire DDT and DDT fallback without any expectation that a user will respond to a request to intervene.	System	System	System	Unlimited

[31] Society of Automotive Engineers: J3016 Terms related to driving automation (2016), p.17.

2.2 Functional Safety

The development of a concept of a redundant active steering system for AD3+ of trucks is the key target of this thesis. As already discussed in the previous subchapters, the present standard HPS is not suitable for this use case, since it is not able to generate steering torque independently of the driver. Hence, additional electrical and/or electronic (E/E) systems are necessary in addition. This emphasizes the functional safety according to the ISO 26262[32] as a key topic to consider during the development process. Since its latest version, the ISO 26262 has been adapted for E/E systems of vehicles with a maximum total weight of more than 3.5 t[33], thus it can be used for the development of E/E systems in trucks today as well.

This subchapter starts with an overview of the ISO 26262 and its definitions of the important terms, which are required for the understanding of this thesis. The focus concerning functional safety is on the concept phase of the ISO 26262, since the scope of this thesis is limited to the development of a comprehensive concept of a steering system for AD3+ of trucks. Therefore, a short summary of this concept phase and its application to automated driving and to steering systems is given below.

The new Public Available Specification (PAS) ISO/PAS 21448 on the safety of the intended functionality (SOTIF)[34] is not considered within this thesis, because it only covers the correctness of the intended functionality, but not the functional safety of the system technologies, which include steering systems.

2.2.1 ISO 26262

The ISO 26262 is the international standard from the International Standardization Organization (ISO) about functional safety of E/E systems especially for use in the automobile industry. It was developed based on the IEC 61508 (International Electrotechnical Commission) because more and more safety-relevant E/E systems are implemented in modern vehicles. The ISO 26262 provides guidelines for the development of such systems, without dictating certain methods for the single development steps. Figure 2-1 gives an overview of the single development steps described in the ISO 26262.

Because the key target of this thesis is to determine the requirements for a steering system for AD3+ of trucks and implement them into a concept of a redundant active steering system, only the following parts of the ISO 26262 are considered within this thesis:

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[32] International Organization for Standardization: ISO 26262 (2018).

[33] International Organization for Standardization: ISO 26262-1 (2018), p.1.

[34] International Organization for Standardization: ISO/PAS 21448 SOTIF (2019).

- ISO 26262-3: Concept phase[35]
- ISO 26262-4: Product development at the system level[36]
- ISO 26262-5: Product development at the hardware level[37]
- ISO 26262-6: Product development at the software level[38]

The safety analysis in subchapter 5.2 presents the results from the application of the whole content described in the concept phase in ISO 26262-3. The used methodology for the concept phase is described in section 2.2.1.2. The content of the other three parts is not depicted separately, but parts of ISO 26262-4 are applied in subchapter 5.3, parts of ISO 26262-5 in chapter 7 and parts of ISO 26262-6 in chapter 6.

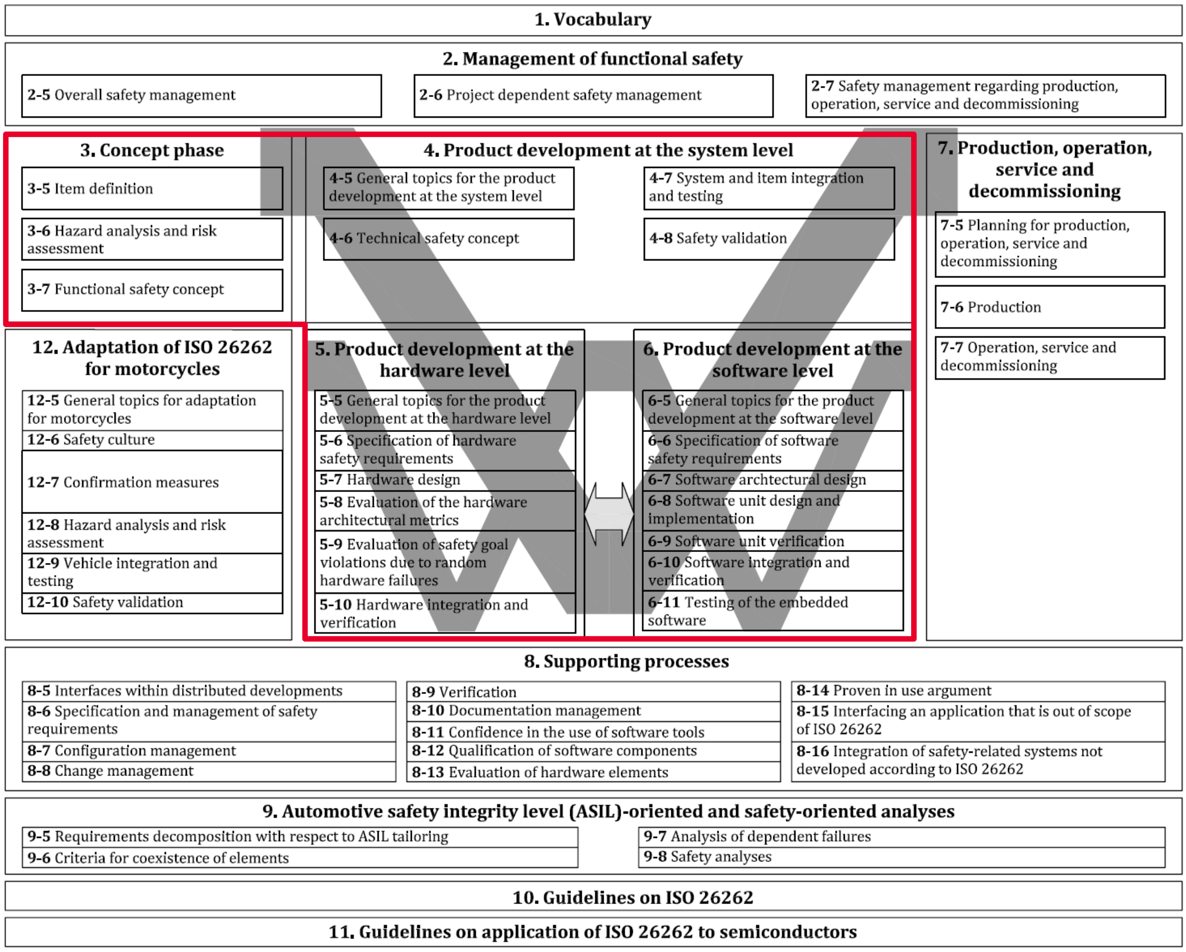

Figure 2-1: Overview of ISO 26262 methodology[39]

[35] International Organization for Standardization: ISO 26262-3 (2018).

[36] International Organization for Standardization: ISO 26262-4 (2018).

[37] International Organization for Standardization: ISO 26262-5 (2018).

[38] International Organization for Standardization: ISO 26262-6 (2018).

[39] International Organization for Standardization: ISO 26262-1 (2018), vii.

2.2.1.1 Definitions

First, to understand the approach in this thesis concerning functional safety, the most important vocabulary from ISO 26262-1 are defined in Table 2-2. A more detailed list is attached in appendix A.3.1.

Table 2-2: Definition of important term of ISO 26262[40]

Term	Description
Automotive safety integrity level (ASIL)	"one of four levels to specify the item's or element's necessary ISO 26262 requirements and safety measures to apply for avoiding an unreasonable residual risk, with D representing the most stringent and A the least stringent level"[41]
Error	"discrepancy between a computed, observed or measured value or condition, and the true, specified or theoretically correct value or condition"
Failure	"termination of an intended behavior of an element or an item due to a fault manifestation"
Fault	"abnormal condition that can cause an element or an item to fail"
Functional safety	"absence of unreasonable risk due to hazards caused by malfunctioning behaviour of E/E systems"
Functional safety concept	"specification of the functional safety requirements, with associated information, their allocation to elements within the architecture, and their interaction necessary to achieve the safety goals"
Functional safety requirement	"specification of implementation-independent safety behaviour or implementation-independent safety measure including its safety-related attributes"
Redundancy	"existence of means in addition to the means that would be sufficient to perform a required function or to represent information"
Safety goal	"top-level safety requirement as a result of the hazard analysis and risk assessment"

2.2.1.2 Concept Phase

The ISO 26262 provides a framework for the development of safety-relevant E/E-systems. The third phase of its process is the concept phase, which includes the following steps[42]:

- Item definition
- Hazard analysis and risk assessment
- Functional safety concept

[40] International Organization for Standardization: ISO 26262-1 (2018).

[41] In addition to the four ASIL, QM (quality management) denotes no requirement to comply with ISO 26262

[42] International Organization for Standardization: ISO 26262-3 (2018), p.1.

The item definition targets to provide comprehensive information about the item's functionality, its dependencies on and interaction with its environment and its interfaces with other items, but also legal requirements and potential hazards, to get an appropriate understanding of the item and its behavior.[43]

The final results of the hazard analysis and risk assessment (HARA) are the safety goals, which are formulated to prevent or mitigate hazardous events and thus to avoid unreasonable risk. To devise the safety goals, hazardous events are identified in the HARA by analyzing and categorizing potential hazardous situations. The categorization is based on the so-called automotive safety integrity level (ASIL), which assesses the potential hazardous situations.[43]

The last step of the concept phase is the development of the functional safety concept according to the ISO 26262, with the objective to derive the functional safety requirements based on the functional safety goals from the previous step. Furthermore, the functional safety requirements are allocated to elements of the preliminary system architecture or to external safety measures. The developed functional safety concept contains fault detection and failure mitigation. This includes measures to transmit the vehicle into a safe state with fault tolerance mechanisms, fault detections and driver warnings.[43]

The ISO 26262 provides the framework and methodology for the development of E/E systems of vehicles, but it leaves the applied methods open. The methodology used in this thesis for the concept phase as well as the steps of the ISO 26262 and the methods for the development of the functional safety concept are described later during application in subchapter 5.2.

2.2.2 Fault Tolerance Levels and Redundancy Strategies

AD2- systems rely on the driver as a fallback level in case of a malfunction of the automated system or if the system exceeds its limits. However, the driver is not available as an immediate fallback level for AD3+. The automated system has to provide internal fallback levels instead. For the development and the discussion of a redundant active steering system, it is important to know the different levels of fault tolerance and possible redundancy strategies.

One fault tolerance level is called fail-silent and is used in today's EPS systems. As implicated in its name, such a system keeps passive after one or more faults occur, i.e. it is shut off and thus does not influence other components in a wrong way. For the example of a fail-silent EPS, this means that the electric motor does not lock the steering in case of a fault. Hence the driver is still able to steer the vehicle via the mechanical linkage. A so-called fail-safe system immediately switches to a safe state as soon as one or more faults arise. If no energy is required for the switch, it is called passive fail-safe and it is called active fail-safe if energy is required. If a transition to a safe state is not possible immediately, a so-called

[43] International Organization for Standardization: ISO 26262-3 (2018), pp. 4–17.

fail-operational system is necessary, i.e. the system tolerates the fault and stays operational. Furthermore, it is distinguished if the system stays fully operational or if it stays operational with a degraded functionality. The latter is called fail-degraded.[44a]

In order to achieve a fail-operational or fail-degraded functionality, the system requires redundancies. There are mainly two different types of redundancy defined in the ISO 26262. The multiple identical implementation of a function is called homogeneous redundancy, whereas the implementation of the same function with different solutions is called diversity or diverse redundancy.[45] An exemplary fail-operational EPS system with a homogeneous redundancy for AD3+ of passenger cars is shown in Figure 2-2. All components of the system are available twice. The electric motor contains two independent windings for the implementation of the fail-operational functionality. The two power stages ensure the electric power supply and the two central processing units (CPU) serve as a redundant control of the steering system.

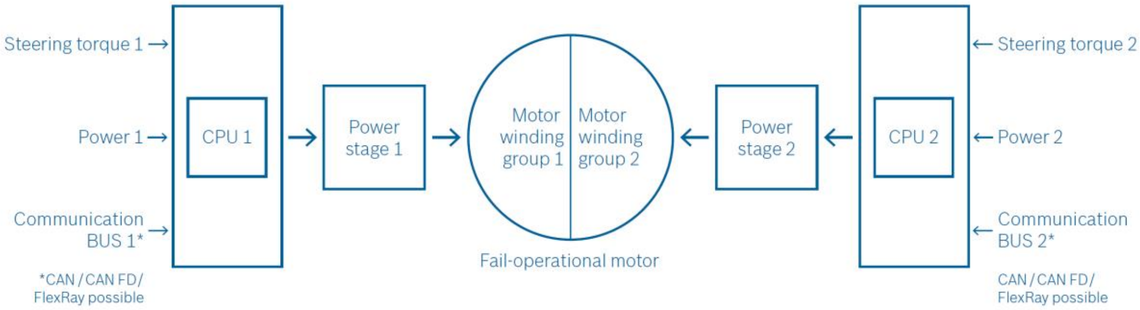

Figure 2-2: Structure of a fail-operational EPS[46]

2.2.3 Functional Safety Concepts of Current Steering Systems

Functional safety concepts of current steering systems suitable only for AD2- or even only for power steering are structured relatively simple. The reliability of sole mechanic and/or hydraulic steering systems, e.g. today’s standard truck HPS, is primarily ensured by an oversizing design. Because they contain no E/E systems, functional safety according to ISO 26262 is not an issue. In contrast, the functional safety of E/E systems of an EPS including their hardware and software components is ensured using safety measures and redundancies.[44b]

Today’s EPS systems for passenger cars are designed in a way that faults can be largely excluded. However, a suitable safety concept has to be developed, which detects occurring

[44] Isermann, R.: Fault tolerance in mechatronic systems (2016), (a) pp. 45–46 | (b) p. 43..

[45] International Organization for Standardization: ISO 26262-1 (2018), p.20.

[46] Robert Bosch GmbH: New steering systems for tomorrow's mobility (2017).

faults and minimizes the fault consequences by transmitting the system into a safe state in a sufficient time. Hence, safety goals for an EPS only for manual driving are:[47]

- EPS has to detect faults that cause an undesirable actuator function and switch into a safe state.
- EPS has to detect faults that cause a heavy-running steering system and switch into a safe state.
- EPS has to prevent unintended reset of the power assistance.

Therefore, the following requirements exemplary define the safe state of such an EPS system:[47]

- The EPS is shut off and does not generate steering assistance.
- For leaving the safe state, the EPS has to be switched off and its ECU has to be successfully reset.
- The electric actuator must not apply any torques to the steering system, above a safety-critical value.
- The steerability of the vehicle by the driver via the mechanical linkage of the steering wheel and the wheels has to be maintained at any time according to ECE R79[48].

From these definitions of the safety goals and the safe state, the functional and furthermore the technical requirements of the EPS's subsystems, such as the ECU, the steering actuator and the steering wheel torque sensor, are derived to prevent any violation of the safety goals.

The previously mentioned driverless research vehicle aFAS has a so-called hybrid steering system, which is a combination of an EPS and a HPS. Thus it is able to steer the truck driverlessly. Its steering system has a fail-safe architecture considering faults inside the steering systems, the power supply and the communication to other systems. It requires no redundancy, because the vehicle's safe state is an immediate stop, which is sufficient for velocities below 10 km/h. Thus, the safe state of the steering system is a passive steering system, which means a deactivated power steering function similar to the fail-silent state of the EPS.[49]

Although systems for AD3+ have much higher complexity, the ISO 262626 is also used for their development. The importance of the concept phase increases because the functional safety concept has to change from fail-safe to fail-operational or fail-degraded and the definition of safety goals and the safe state according to the intended level of automation is more complex.[50] The fail-operational or fail-degraded safety concept is required to transfer the vehicle into a safe state. The high requirements regarding availability of the actuators and

[47] Gaedke, A. et al.: Electric Power Steering Systems (2017), pp. 448–453.

[48] United Nations: ECE R79 r4 (2018), p.30.

[49] Christian Rief: aFAS Steering System (2018), p.5.

[50] Martin, H. et al.: Functional Safety of ADS (2016), p.413.

their control are achieved by suitable redundancy concepts.[51] The structure of the system has to be designed in a way that in case of a fault a minimum set of the system's components stay operational and thus are available as a fallback level. The power supply of the steering system has to provide at least enough energy for the fallback level to reach the defined safe state. The required minimum set of steering components and the required minimum steering energy, also described as level of degradation, are not described in the current research yet.[52]

Safety goals and functional safety concepts for the control of steering system suitable for AD3+ as well as for the communication between the ADS and the steering system are already developed exemplarily for the research vehicles "MOBILE" of TU Braunschweig[53,54] and "SpeedE" of RWTH Aachen.[55] Both approaches have been developed specifically for those research vehicles with steer-by-wire (SbW) four-wheel steering systems and are not applicable to truck steering systems, which are in scope of this thesis. However, those research results are used as input for the safety analysis in this thesis.

2.3 Legal Requirements for Steering Systems

This subchapter gives an overview of the legal requirements for steering systems, especially concerning automated driving. The two main legal requirements valid for steering systems for the automated driving of trucks are the "Vienna's Convention on Road Traffic"[56a] and the "Uniform Provisions Concerning the Approval of Vehicles with Regard to Steering Systems"[57] (ECE R79). Both documents contain many regulations, of which only those that are crucial for the steering system of automated trucks are described hereafter.

The convention on road traffic dictates that "every motor vehicle shall be equipped with a strong steering mechanism which will allow the driver to change the direction of the vehicle, easily, quickly and surely"[56b], whereas "every driver shall at all times be able to control his vehicle"[56c].

Since the original convention on road traffic originates from 1968 and was designed for manual driving only, some extensions were made considering automated driving, among other things. Hereby, an automated system is allowed to perform the vehicle's driving task instead of the driver, who can do other activities meanwhile. However, this is only valid if

[51] Becker, J. et al.: System architecture and safety requirements for AD (2017), p.275.

[52] Becker, J.; Helmle, M.: Architecture and System Safety Requirements for AD (2015), p.43.

[53] Stolte, T. et al.: Safety goals and functional safety requirements (2016).

[54] Stolte, T. et al.: On Functional Safety of Vehicle Actuation Systems (2016).

[55] Gillen, C.: Dissertation, Development of efficient safety concepts for steering systems (2015).

[56] United Nations: Convention on Road Traffic (1968), (a) - | (b) p.59 | (c) p.11.

[57] United Nations: ECE R79 r4 (2018).

the driver's further activities do not prevent the driver from taking over the driving task when required. This condition makes it clear that only AD2- is enabled by the last extensions. For the introduction of AD3+, further extensions are necessary in the future.[58]

The ECE R79 is the valid regulation for the steering systems for vehicles of the categories M, N and O, including the category N3, which is on behalf of this thesis. The ECE R79 was also adapted to enable ADAS functions and to support the development of automated driving. It differs between "Autonomous Steering Systems", which are controlled from outside the vehicle, "automatically commanded steering functions" (ACSF), which are steering functions controlled automatically by the vehicle, "corrective steering functions" (CSF), which are functions that support the driver only for a limited duration, and the "emergency steering functions" (ESF), which intervene in case of an emergency to avoid a collision. The target steering system of this thesis belongs to ACSF, which do not require commands of the driver for controlling the vehicle. Those systems are not included in the current version of the ECE R79, but appropriate amendments are already provided. Definitions that are more detailed are listed in appendix A.1.1.[59a]

This differentiation in the ECE R79 is based on the automated lateral guidance control of the vehicles, but not on the functionality of the actual steering system. In addition, it does not fit with the SAE levels of automation. Regardless of the categorization defined in the ECE R79, a suitable automated steering system requires the ability to generate steering torque independently of the driver. In addition, the testing procedure dictated by the ECE R79 is focused on the lateral guidance functionality and not on the actual steering system's functionality. It seems to be questionable if such requirements are placed in the correct regulation here.

However, there are a few regulations that are addressed to the actual automated steering system. Similar to the Vienna's convention on road traffic, the ECE R79 also requires that the driver is able to override the automated system at any time by deliberate intervention.[59b] For all categories of automated steering functionalities, it dictates a maximum required force at the steering wheel to override the system by the driver of 50 N[59c], i.e. a steering wheel torque of 12.5 Nm at a truck steering wheel with 0.25 m radius.

Furthermore, the ECE R79 requires the development and description of a safety concept for the automated steering system, which shall ensure a safe operation of the steering system even in case of a system fault. A fallback level inside the steering system, a separate back-up system or the reduction of the available automated steering functionality are proposed for adequate safety measures to guarantee the safe operation.[59d]

Important for truck steering systems are the regulations that all requirements should be met with a fully loaded truck and the steering effort at a steering wheel with 0.25 m radius shall not exceed 50 Nm for an intact steering system respectively 112.5 Nm for a steering system

[58] United Nations: ECE WP.1/159 (2017), p.5.

[59] United Nations: ECE R79 r4 (2018), (a) pp. 5–7 | (b) p.4 | (c) p.14 | (d) pp. 42–45.

with a failure.[60] The latter requires a special dual-circuit HPS for trucks with very high front axle loads, because otherwise in the event of a failure of the operation steering system, the driver would not be able to steer those vehicles exclusively mechanically.[61] The target steering system has the claim to replace this dual-circuit system.

2.4 Active Steering Systems

This subchapter summarizes the current state of the art and of the scientific research concerning active steering systems, especially for trucks. The term active describes systems in which the quantity of the output variable is controlled by an actuator and auxiliary power, such as an EPS. In contrast, passive systems are not able to control the transferred quantity with additional auxiliary energy, such as standard HPS. With standard passive HPS, the hydraulics assist the driver with auxiliary power, but the assistance rate is determined by the design of the steering valve and the PSP and is not controllable during operation. They are comparable to a fixed-ratio gear box.[62,63]

In this thesis, active steering systems describe steering systems that are able to generate steering torque and thereby change the steering angle independently of the driver, but by means of electronic signals instead. Passive steering systems require the input torque of the driver as mechanical input signal to generate and to control the power steering torque.

In the following sections, EPS systems like used in most of today's passenger cars and a prototype EPS for commercial vehicles are described first. Two different types of active HPS for trucks are presented as well, which are the hydraulic power steering systems with active valves and the so-called hybrid steering systems. To complete this summary, steer-by-wire concepts are described as well. With all these active steering systems, the functionality concerning the implementation of ADAS and automated driving is similar to today's EPS of passenger cars, which means that the assistance torque is controllable independently of the driver's input torque and steering torque is producible even without any input of the driver.

[60] United Nations: ECE R79 r4 (2018), p.30.

[61] Hullmann, J. et al.: Mechanical and Hydraulic Gears (2017), p.335.

[62] Isermann, R.: Mechatronic Systems (2008), p.56.

[63] Brunner, S.; Harrer, M.: Steering Requirements (2017), p.59.

2.4.1 Electric Power Steering

Electric power steering (EPS) systems are mainly used in today's passenger cars with rack and pinion steering. The disadvantage of the lower power density compared to a HPS is negligible here, but the benefits of an EPS are the higher efficiency and the higher functionality compared to a HPS. The former is achieved by the power-on-demand feature of the EPS, which means that it is only activated when required, whereby its energy consumption is decreased. Since an EPS is an active steering system, it is able to generate torque independently of the driver, which can be used for comfort or ADAS functions.[64]

Figure 2-3 shows a schematic sketch of an EPS, to explain its structure. In contrast to the previously described HPS, the EPS does not use the driver's input torque as a mechanical input signal. The input torque of the driver at the steering wheel (1) is measured by a sensor (10), which transfers this information to an electric control unit (ECU, 11). The ECU calculates the required assistance torque depending on the current driving state and on demands of ADAS or AD functions. Using the electric power stored in a battery (12), supplied by a generator (13) driven by the ICE (7), the ECU controls an electric motor (14), which overlays its torque to the driver's torque at the shaft between the steering wheel and the steering gear (3) via a torque overlay gear (15). The kinematics from the steering gear to the steered wheels vary depending on the vehicle and are shown here exemplary for a truck. The electrical recording and processing of the driver's torque and, above all, the electrical control of the steering actuator make the EPS an active system. However, ensuring its functional safety becomes a central issue as a result.

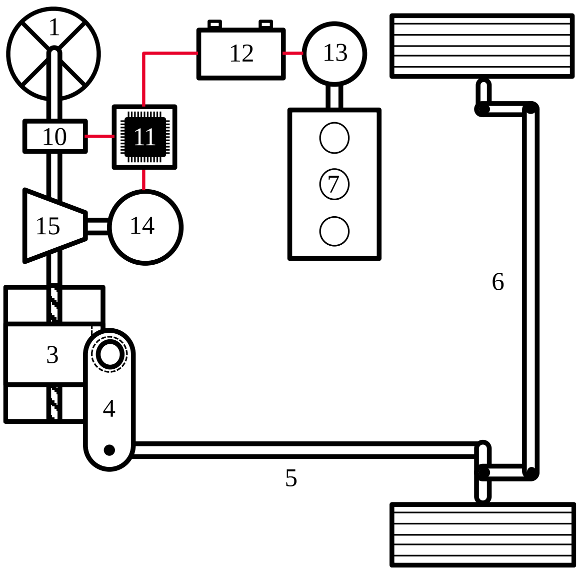

Figure 2-3: Schematic sketch of an EPS for trucks

There are different types of EPS steering systems on the market, which differ in the mounting position of the electric motor. Since there is no EPS for trucks yet, an exemplary functional

[64] Gaedke, A. et al.: Electric Power Steering Systems (2017), pp. 403–404.

structure for a so-called rack-EPS for passenger cars is shown in Figure 2-4. One of the crucial components of an EPS is the steering wheel sensor (sensor), which senses the driver's input torque and angle ($M_{\mathrm{H}}, \delta_{\mathrm{H}}$) at the steering wheel. The electric control unit (ECU) processes this information, connects it with the torque or angle requirements from the ADAS ($M_{\mathrm{ADAS}}, \delta_{\mathrm{ADAS}}$) and transmits it to the power electronic (PE), where the signal is connected with electric energy from the battery (E_{Batt}). This energy is used to supply the electric motor (EM) to generate additional steering torque (M_{e}), which is added to the steering power of the driver by a torque overlay gear (gear). The steering torque and angle of the driver are converted into a steering force (F_{H}) and a translation (y_{H}) in the steering gear (StG). The result is the overall sum of steering forces (F_{R}) and rack translation (y_{R}). Those are transformed to the steering torque and angle at the wheels ($M_{\mathrm{w}}, \delta_{\mathrm{w}}$).

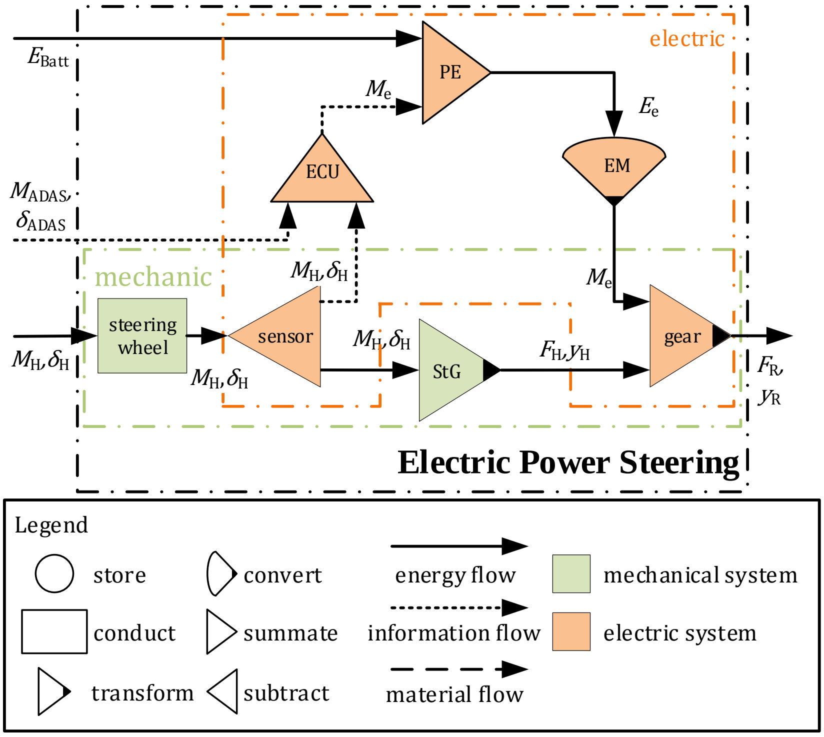

Figure 2-4: Functional structure of electric power steering (EPS) system

Standard EPS are already suitable for AD2-, since they are active steering systems and require no fail-operational system design. The driver is available as an immediate fallback level. Hence, the described fail-silent design is sufficient.

The only one example of an EPS for commercial vehicles (CV) is ZF's ReAX EPS, which is a prototype published in June 2018 and shown in Figure 2-5. According to ZF, all SAE Levels of Driving Automation can be realized with this system. A powerful EM with a maximum torque of 70 Nm in combination with a 48 V on-board vehicle power supply system generates all required steering torque without any hydraulics. This system can also be used for a steer-by-wire application. So far, this system is a very early prototype and only this little information is known about it. Its maximum output steering torque and power are not published yet. However, it is no suitable solution for the scope of this thesis, since it requires a 48 V electric power supply and is not realizable with the available 24 V on-board network.[65]

AD3+ also requires an EPS to have a fail-operational system design. An approach for such a fail-operational EPS for passenger cars is already presented in Figure 2-2. Such a system structure allows AD3+ and offers high safety due to its redundant system design. In case of a fault, the second independent electronic circuit provides at least half of the steering assistance and thus the functionality to guarantee a safe stop of the vehicle.[66]

Figure 2-5: ZF ReAX EPS[65]

[65] ZF Friedrichshafen AG: ZF Press Release (2018).

[66] Robert Bosch GmbH: New steering systems for tomorrow's mobility (2017), p.5.

2.4.2 Hydraulic Power Steering Systems with Active Valve

A more powerful solution for an active steering system is a standard HPS, which becomes an active steering system by an integrated active valve. As shown in a schematic sketch of such systems in Figure 2-6, a sensor (10) is used to record the input torque of the driver at the steering wheel (1) similar to EPS.

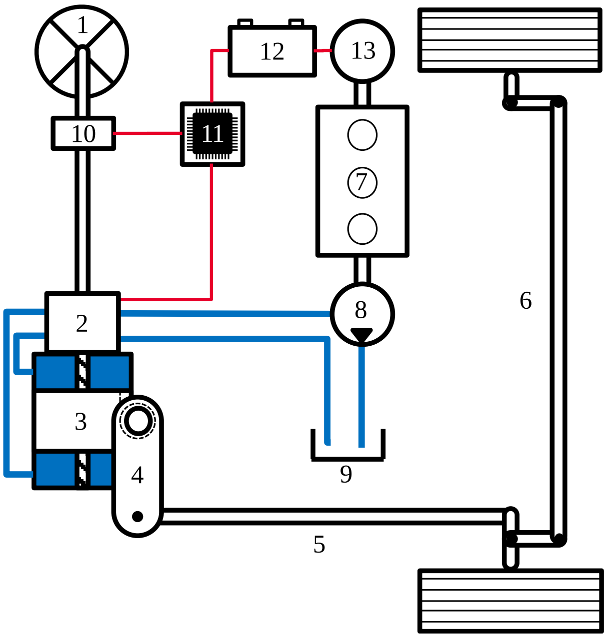

Figure 2-6: Schematic sketch of a HPS for trucks with active valve

The ECU also uses electric energy from the battery (12) and the generator (13) to control the active valve (2) and thereby to control the hydraulic assistance torque. The hydraulic energy is supplied by a PSP (8) driven by an ICE (7), which pumps the hydraulic oil to the active valve and the hydraulic piston inside the steering gear (3). The kinematics are similar to those of a standard HPS of trucks. With this system, the driver is still able to control the valve by his input torque. The active valve serves as an additional actuator, which also controls the quantity of the systems output torque independently of the driver.

An exemplary function structure of such a steering system is illustrated in Figure 2-7. The hydraulic components are similar to those of a standard HPS except for the active valve (AV) and the variable power steering pump (vPSP), which is able to adjust its output volume flow. Similar to EPS systems, a sensor detects the driver's input torque and angle ($M_{\mathrm{H}}, \delta_{\mathrm{H}}$) at the steering wheel, an ECU processes and combines them with information from the ADAS and transmits them to the PE. The PE supplies the modulation actuator (MA), which is able to overlay an additional driver-independent rotation angle (φ_{VS}) to the driver-induced valve torsion angle (φ_{VTB}) via a gear. Thereby, the HPS is controllable independently of the driver by the modulation mechanism also called modulator, but still by the driver as well.

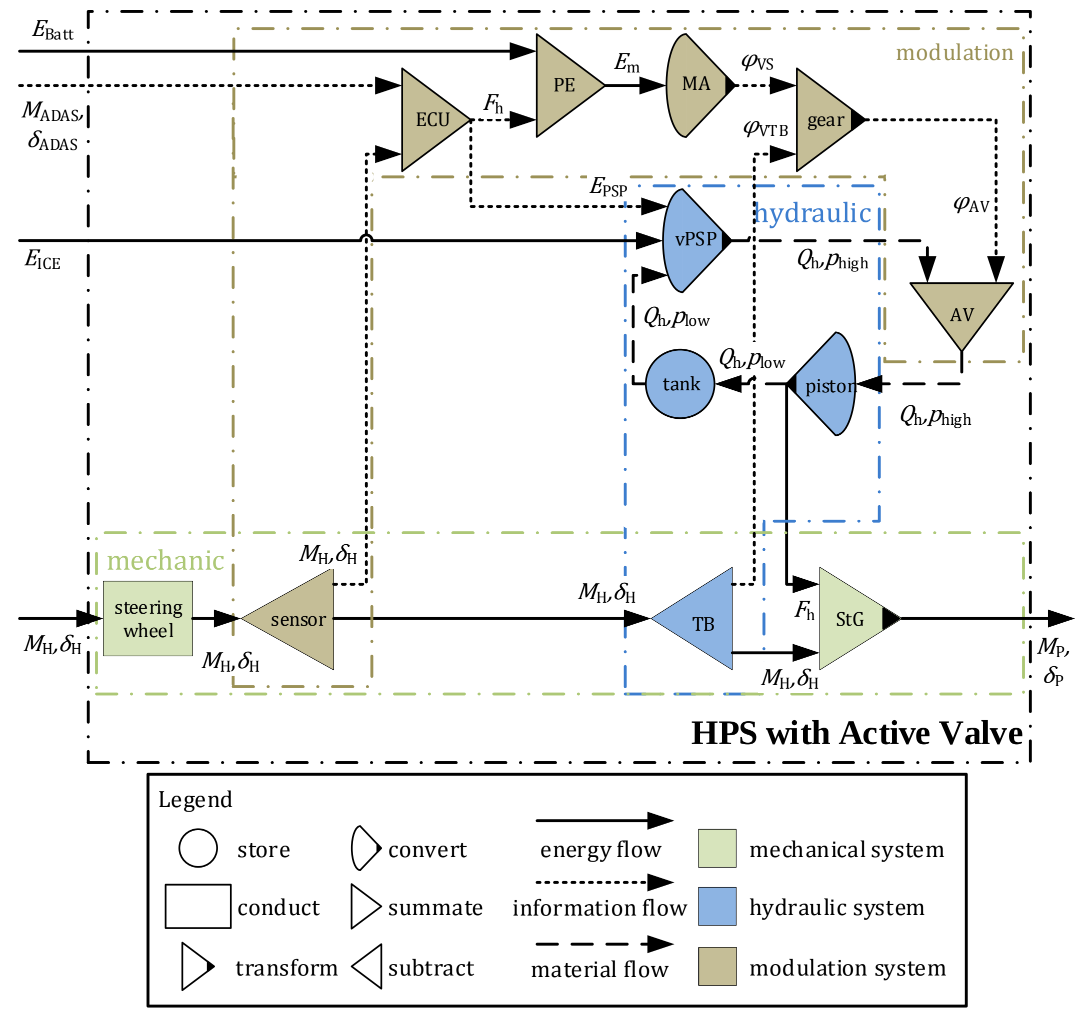

Figure 2-7: Functional structure of HPS system with active valve

There are different ways to implement the overlay mechanism of the modulator, but all existing solutions are based on an adjustable valve sleeve.[67,68] The sleeve is twisted independently of the driver by an additional actuator, e.g. an electric motor or an electro-magnet, to control the steering torque of the HPS independently of the driver. Because the actuator is only responsible for the modulation of the hydraulics, but not for the actual generation of steering torque, a low actuator power of maximum 100 W is sufficient for that task. The steering power is still provided by the PSP similar to standard HPS. Hence, such systems are not able to shut-off the hydraulics and steer solely electrical, but by using a PSP with adjustable power demand an increase in efficiency is also possible.[69]

One example for a hydraulic power steering system with an active valve is Knorr-Bremse's "intelligent Hydraulic Steering Assist" (iHSA) with a recirculating ball (RCB) gear. The active valve module iHSA is shown on the left side of Figure 2-8 alone and in combination

[67] Janz, B. et al.: Steering valve with planetary gear (2009).

[68] Birsching, J. E.: HPS with magnetic torque overlay (2014).

[69] Müller, J.-H.: Torque overlay for hydraulic steering (2010), p.558.

with a RCB gear on the right side. The actuator of the iHSA is a small electric motor for controlling the hydraulic valve.[69]

Another example for an active valve for the combination with a hydraulic RCB gear is Wabco's "Magnasteer". It has the same functionality as the iHSA, but it uses an integrated electromagnetic valve actuator.[70]

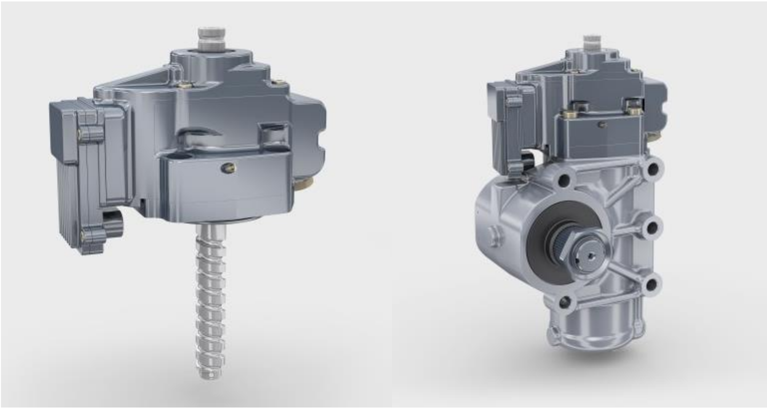

Figure 2-8: iHSA Module (left)[71] and iHSA Module with hydraulic RCB Gear (right)[72]

Both active valve systems are designed as a module, which is mounted at the top of the RCB and replaces the standard passive steering valve including the input shaft of the RCB. Compared to passive valves, the active valve modules have bigger dimensions, whereas the height is crucial because the space above the RCB is limited in the truck.

2.4.3 Hybrid Steering Systems

Another solution of an active steering system for trucks is a so-called hybrid steering system. It is called hybrid because it uses two different energy sources to produce steering power and steering torque by combining an EPS system with a standard HPS system, as shown in the schematic sketch in Figure 2-9. The electric subsystem is similar to the described EPS and includes the sensor (10), the ECU (11), the battery (12), the generator (13), the electric motor (14) and the torque overlay gear (15). The hydraulic subsystem contains the passive valve (2), the steering gear (3) with an integrated hydraulic piston, the PSP (8) and the tank (9). The kinematics are standard truck steering kinematics. By connecting an EPS between the steering wheel and the HPS, the hybrid system receives the same functionality as an EPS. The latter overlays torque to the driver's torque or replaces the input torque of the driver at the HPS during automated driving.

[70] WABCO: Active Steering Technology (2018).

[71] Knorr-Bremse: iHSA Control Module (2018).

[72] Knorr-Bremse: Truck RCB Steering Gear with iHSA Control Module (2018).

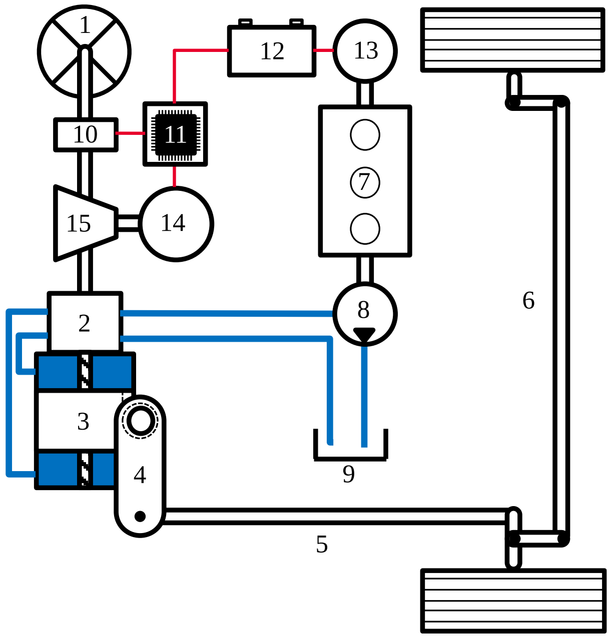

Figure 2-9: Schematic sketch of a hybrid steering system for trucks

An exemplary functional structure of a hybrid steering system is shown in Figure 2-10. The EPS is mounted between the steering wheel and the torsion bar (TB) of the hydraulic valve at the input shaft of the steering gear (StG). The functionality of the EPS is similar to a standalone EPS, same for the HPS. The position of the EPS in front of the valve allows it to control the valve in addition to the driver. This enables a hybrid steering system to actively generate steering torque not only electrically but also hydraulically. With the additional electric motor (EM) of the EPS, those systems are able to generate steering torque purely electrically and combined electrically and hydraulically. The hydraulics are shut off when only low steering power is required, which the EPS is able to provide alone. Thereby, the power required by the hybrid steering system is significantly reduced by 75 % compared to standard HPS.[73]

[73] Wiesel, U. et al.: Hybrid steering system for reducing fuel consumption (2010), p.68.

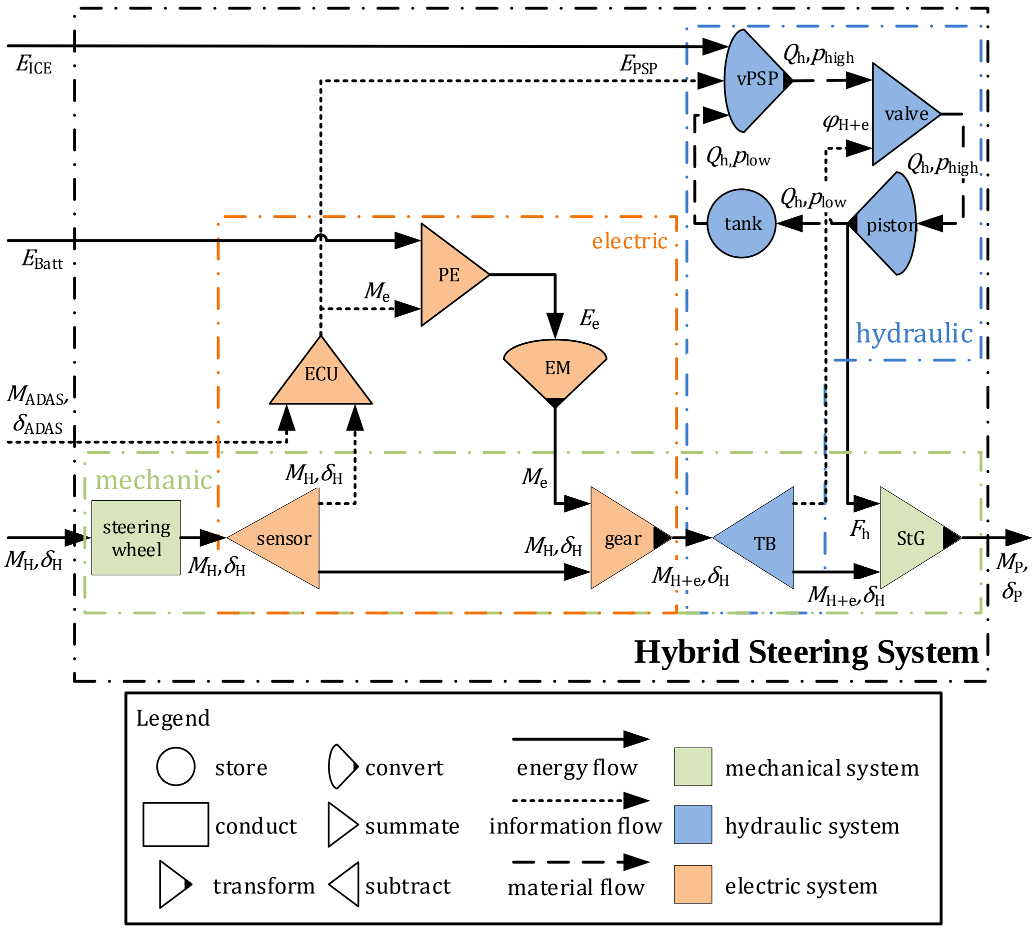

Figure 2-10: Functional structure of a hybrid steering system

There are several approaches known to implement a hybrid steering system. A selection of these is described in the following. Two similar examples for hydraulic power steering with electric torque overlay are ZF's "ReAX Gear Mounted" and Bosch's "Servotwin", shown in Figure 2-11. Both use an electric motor and a worm gear to overlay electrically generated torque to the input shaft of the passive hydraulic valve of the HPS. The Servotwin delivers over 8 kNm of hydraulic torque at the pitman arm and 65 Nm of electric torque at the input shaft of the passive valve.[74] Both systems are available on the market.

[74] Bosch Mobility Solutions: Automated and efficient for the future (2018), p.5.

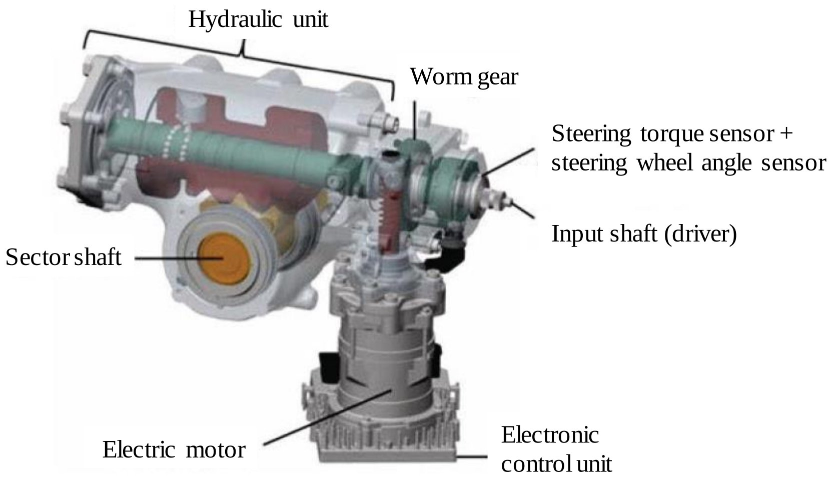

Figure 2-11: Structure of the Servotwin[75]

Other hybrid steering systems are Volvo's "Dynamic Steering System"[76], which is already established on the market and functions similar to the first two examples, or MAN's "EcoSteering" as a result of a research project[77a]. The latter distinguishes itself from the other systems by the mounting position of the EPS. As shown in Figure 2-12, the electric motor and the overlay gear are attached at the lower end of the RCB's input shaft. Thereby, the EPS is also able to generate steering torque electrically, but cannot actively control the HPS via the hydraulic valve. This limits the functionality of the system, but it simplifies its control as the EPS is controllable independently of the driver's input torque and the hydraulic valve.

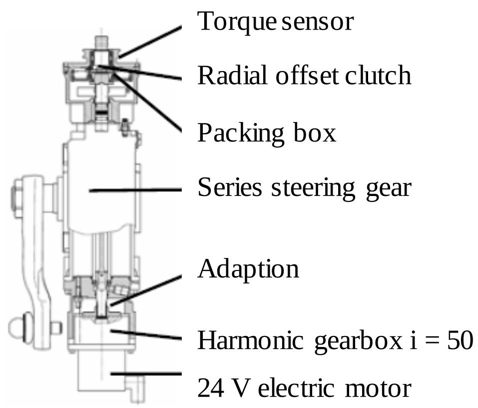

Figure 2-12: Structure of MAN's EcoSteering[77b]

[75] Gaedke, A. et al.: Driver assistance for trucks (2015), p.217.

[76] Volvo Trucks Magazine: Benefits of Volvo Dynamic Steering (2017).

[77] Brunner, G.; Negele, K.: Electrification of the steering (2008), (a) - | (b) p.7.

2.4.4 Steer-by-Wire Systems

Steer-by-wire (SbW) systems are innovative steering systems characterized by the mechanical decoupling of the linkage between the steering wheel and the steered wheels. The structure is similar to those of an EPS, as shown in Figure 2-3, but without the mechanical linkage between the steering wheel and the steering gear. An additional actuator to generate feedback torque at the steering wheel is installed instead.

An exemplary function structure is depicted in Figure 2-13. A sensor detects the driver's intended steering torque and angle (M_H, δ_H) at the steering wheel. An ECU transmits these signals via the PE to the EM, which is connected to the wheels by a gear and the steering linkage and executes the driver's intention at the steered wheels. An additional sensor detects the torque and angle at the output of the steering system at the pitman arm (M_P, δ_P) and transmits them to a so-called feedback actuator (FA) via the ECU to provide haptic response to the driver by a feedback torque (M_{FB}) and feedback angle (δ_{FB}).

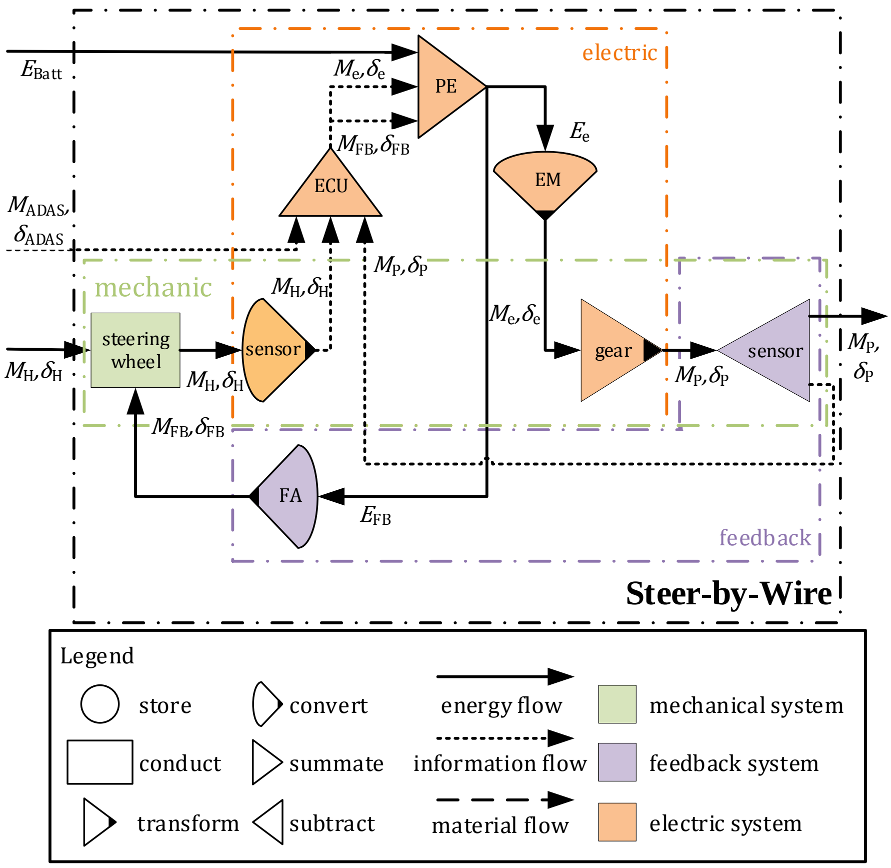

Figure 2-13: Functional structure of a steer-by-wire (SbW) system

The disadvantages of this type of steering systems are higher costs, weight and complexity compared to the previous presented steering systems, which are caused by the high redundancy requirements of SbW systems. In case of a fault, a switch to a fail-silent mode is not sufficient because of the missing mechanical linkage between the steering wheel and the steered wheels. Hence, as for AD3+, a fail-operational mode is required with multiple implementations of all components. Thereby, it is guaranteed that the driver is able to control the steering even in the case of any fault.[78]

A functional safety concept for a SbW system includes an operation strategy, fault management with a degradation plan and a suitable warning plan for the driver. The sensors have to be three times redundant and the actuators at least twice.[79a]

In contrast, the main benefits of a SbW are the ability to control the steering angle and steering torque at the wheels completely independent of the driver's input at the steering wheel. In the other direction, the haptic feedback for the driver can be designed independently of the forces at the wheels as well. Thereby, a SbW is predestined for the implementation of ADAS and automated driving. Besides the functional benefits, a SbW also has some constructive advantages.[79b]

Because current active steering systems have almost similar functionality, but much lower safety requirements, the development of SbW systems is not particularly focused on today and no SbW system is used in any production car or truck for public roads until now. However, with the increasing electrification of vehicles and the introduction of AD3+ vehicles, which also require fault-tolerant steering systems, it is possible to use technical synergies and thus to reduce costs. This could be a change for SbW systems to establish on the market.[79c,80]

2.5 Conclusion of the Analysis of the State of the Art and the Scientific Research

The previous sections give an overview on the current state of the art and of the scientific research of highly automated driving (AD3+), especially of trucks, and its requirements to the steering system. The categorization and description of the different levels of driving automation according to the SAE J3016 defines that the driver is not available as a fallback for the dynamic driving task for AD3+ systems. Thus, the automated vehicle has to provide an internal fallback level.

Furthermore, the importance of the functional safety for active steering systems for AD3+, especially the importance of the concept phase, is clarified here. To ensure a safe operation of an active steering system during any conditions, even in case of a steering system fault, adequate safety goals have to be defined and a functional safety concept that fulfills these

[78] Winner, H. et al.: When does By-Wire arrive brakes and steering? (2004), p.63.

[79] Huang, P.-S.; Pruckner, A.: Steer by Wire (2017), (a) pp. 523–524 | (b) p.513 | (c) pp. 524–526.

[80] Harrer, M.; Pfeffer, P.: The Future of Steering Systems (2017), p.546.

safety goals has to be developed. This results in the need of a redundant active steering system or a redundant active lateral guidance functionality for AD3+ that is fail-operational or at least fail-degraded by using a homogenous or diverse redundancy. To avoid a system failure by a systematic fault, a diverse redundancy is preferable. A simple homogenous redundancy by implementing all components multiple times is also inefficient concerning costs and assembly space. By determining the minimum required degradation level of a fail-degraded redundant active steering system, which is sufficient for AD3+, an efficient functional safety concept is derivable. Current safety concepts for active steering systems are mostly only fail-silent, since they are only designed for AD2- and either the driver is available for immediate fallback or an immediate safe stop is possible at any time. The presented fail-operational EPS in Figure 2-2 is also not suitable for AD3+ of trucks due to its power limitations.

Although the legal requirements are not valid for AD3+ yet, they already include the following requirements, which are also reasonable for AD3+. The Vienna's Convention on Road Traffic and the ECE R79 both dictate that the driver of an automated vehicle has to be able to override the automated steering function at any time with less than 12.5 Nm of steering wheel torque. In addition, they require the development of a safety concept for the steering system including a fallback level and enough energy reserve to transfer the vehicle to a safe state at any time.

There are already several active steering systems available. A road map of steering systems for trucks is shown in Figure 2-14. Although the EPS has several benefits and there is also a prototype for an EPS for CV with 48 V on-board power supply, the high power and torque requirements for a truck and the prevalence of the 24 V on-board power supply still require the high power and torques of HPS systems.[81,82] A SbW system has the same power issue as the EPS and in addition it requires big effort to guarantee the ability to override the automated system by the driver.

However, standard HPS is not usable for ADAS or automated driving, because it is no active steering system.[83] An electrification of the HPS is required to transform it to an active steering system. Therefore, the HPS with an active valve (AV) and the hybrid steering system are two different approaches for active steering systems for trucks. Both systems are sufficient for AD2-, but not for AD3+, since they are fail-silent systems and have no integrated steering redundancy.

For AD3+, the development of a fail-operational or at least fail-degraded steering system is required. This can be a redundant active steering system (RASS) for trucks with a 24 V on-

[81] Wiesel, U. et al.: Hybrid steering system for reducing fuel consumption (2010), p.64.

[82] Müller, J.-H.: Torque overlay for hydraulic steering (2010), p.557.

[83] Brosig, S.; Lienkamp, M.: Driver Assistance System Functions (2017), p.532.

board power supply or a redundant EPS (EPS²) for trucks with a 48 V on-board power supply. Since the 24 V on-board net will remain the standard voltage system in trucks for at least the next decade[84], this thesis focuses on the development of a RASS for AD3+ of trucks that draws its steering power from both the engine and the on-board net.

Since there is no RASS for trucks known in the current state of the art (SoA), there are also no strategies for the operation of such a system, including system state transitions and fallback strategies, or a proper functional safety concept available.

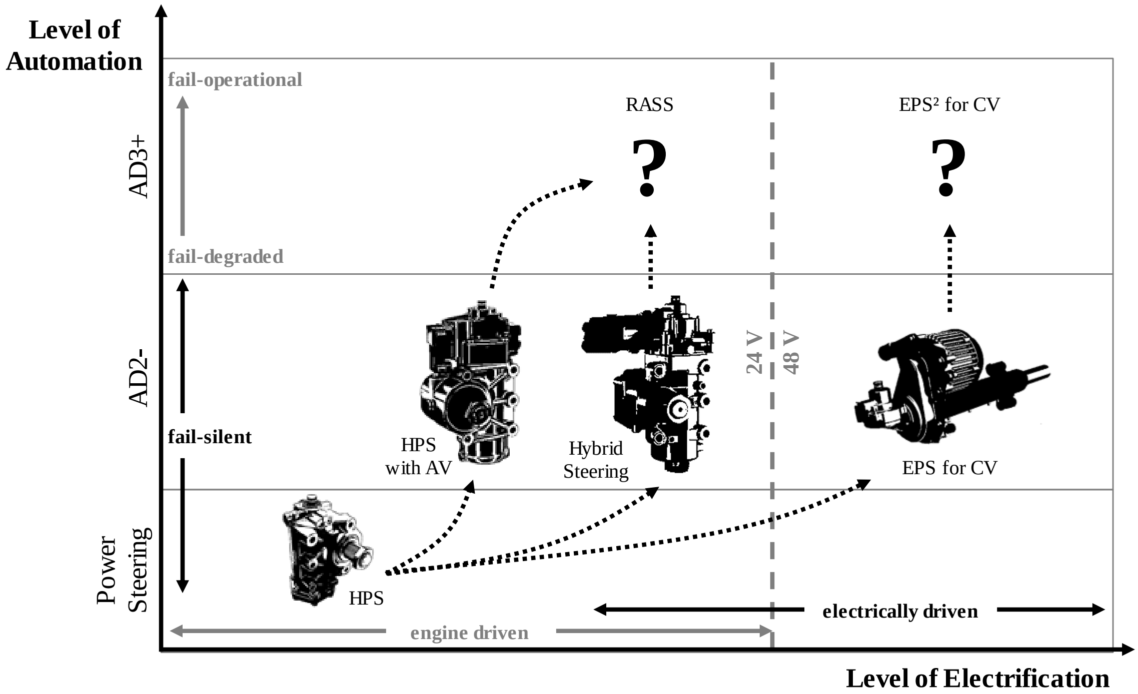

Figure 2-14: Roadmap of steering systems for trucks[85]

[84] Expert discussion with development engineer from MAN Truck & Bus SE.

[85] Following Bosch Mobility Solutions: Automated and efficient for the future (2018), p.3.

3 Objectives and Methodology of the Thesis

3.1 Objectives

Considering the intended target application and the results from the analysis of the state of the art, the superior objective of this thesis is the development of a steering system for AD3+ of trucks. Since it is not available in the current state of art, a redundant automated lateral guidance system is required for AD3+, which includes an active steering system with an appropriate fallback system. The active steering system should be able to overlay torque to the driver's torque and to generate steering torque without any driver input. A steering angle overlay or a steer-by-wire functionality is no target functionality of this thesis. In order to achieve this superior objective, it is necessary to meet the research need outlined below, which has not yet been resolved by the current state of research.

First of all, there is a research need on the requirements of the steering fallback system, which is necessary due to the absence of the driver as an immediate fallback during AD3+. So far there is no investigation known that answers the question whether the steering system has to be fully fail operational or if a fail degraded system is sufficient. If it is clear which fallback requirements are demanded of the steering function, it is examined whether an external fallback system using the brakes is sufficient for this or whether a redundant steering system is necessary.

With the determined fallback requirements and the definition of the redundancy requirements, the required functionalities and sufficient functional structures of the steering system have to be identified.

The required increased functionality of the steering system demands the consideration of the functional safety, whereby there is an open issue of a functional safety concept of a steering system for AD3+ of trucks. For the further research progress, it is important to determine the functional safety requirements that have to be implemented in a proper highly automated steering system.

Since there is no steering system for AD3+ of trucks in the current SoA, a proper system architecture has to be determined for the implementation of the defined functional safety requirements. Based on the already identified suitable functional structures, more detailed structures have to be developed which not only focus on the system's functionality but also on the safety-related issues including the fallback requirements.

The increased requirements of the functionality of the steering system also require the provision of an increased amount of system states. There is a research need to examine which states the steering system must provide in order to be suitable for AD3+. Not only the different system states play a decisive role, but also the transitions between them. This requires the development of special operation and state transition strategies.

The final step for the development of the new steering system concept is the determination of the design specifications of the crucial system components. The crucial components of the steering system responsible for the implementation of all the requirements identified in advance has to be defined first, before the design specifications of those components are determined.

The desired result of this research is a comprehensive concept of a steering system for AD3+, which includes a safety concept as well as specifications of the crucial subsystems and components. The derivation of functions that are already integrated in current truck steering systems and the derivation of the resulting specifications are expressly not part of this thesis.

3.2 Overall Methodology and Structure

Based on the defined scope of this thesis and the analysis of the current state of the art, the described objectives and the research methodology, illustrated in Figure 3-1, are derived.

As described, the scope (subchapter 1.3) combined with the analysis of the current state of the art (chapter 2) disclose the research needs within this field. The objectives of this thesis (chapter 3) address the determined research needs.

The applied deductive methodology aims at narrowing the solution space by analyzing all functional and safety-relevant aspects, which are described below. The focus in this thesis is not on the aspects that are also relevant for today's standard HPS truck steering systems, but on the aspects that are necessary to consider due to the new functionalities of the new steering system for highly automated driving.

The first step is to analyze the initial conditions of a steering system for highly automated trucks and results in the frame requirements for the system (subchapter 4.1). The investigation of the requirements for such a steering system determines on the one hand the operational requirements (subchapter 4.3) and on the other hand the fallback requirements (subchapter 4.4). The operational requirements describe the overall required maximum steering torque and steering power of a truck steering system during correct operation. In contrast, the driving maneuvers that are relevant to reach a safe state in case of incorrect operation, e.g. due to a fault in the steering system, are evaluated to define the fallback requirements for the steering system suitable for highly automated driving. It is checked whether the fallback requirements can be met by differential braking, whereby brake steering would become a sufficient steering redundancy. The redundancy concept suitable for highly automated driving of trucks is determined with the help of those results.

Based on the established requirements for a steering system for highly automated trucks, different functional structures, which meet all the requirements, are set up and discussed (subchapter 5.1). A safety analysis investigates the functional safety of such a steering system and results in several safety goals and functional safety requirements for the steering system (subchapter 5.2). In addition, the safety goals and the functional safety requirements are used to develop overall system architectures (subchapter 5.3), which fulfill all previous determined requirements.

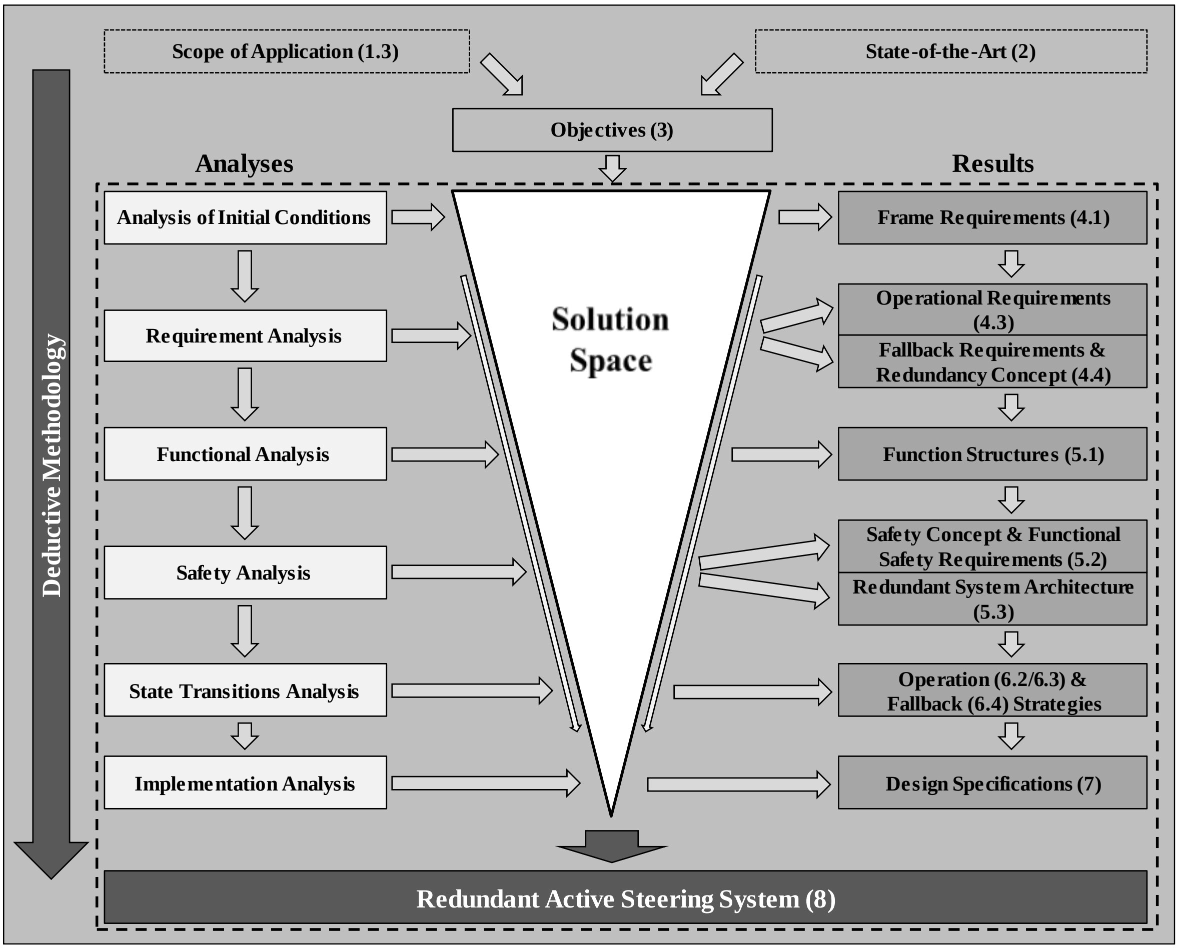

Figure 3-1: Research methodology

The increased functionality of the new steering system architecture is enabled by the steering system's different operation states. The transitions between those states are analyzed and strategies for operational transitions (subchapters 6.2 & 6.3) and for fallback transitions (subchapter 6.4) are developed.

The implementation of the derived requirements in the system design is developed to determine the specifications of the system's crucial components (chapter 7).

The final concept for a redundant active steering system (RASS) is compared with the current state of the art of truck steering systems and the innovations and advantages of the RASS are discussed (chapter 8).

4 Requirement Analysis

The requirements of a steering system for AD3+ of trucks, which is in focus of this thesis, are determined in this chapter. The frame requirements define the primary requirements of a truck steering system. With the help of a simulation model of the steering system and several test drives, the steering torque and steering power demanded in different driving maneuvers during correct operation are determined. For the use of AD3+, not only the correct operation is relevant and therefore must be considered, but also the event of a fault of the steering system. Therefore, redundancy is necessary. The fallback requirements and redundancy requirements are developed using the steering model and test drives as well. It is verified whether steering by differential braking is suitable as fallback steering. The overall determined requirements of the steering system for AD3+ of trucks are summarized at the end of this chapter.

4.1 Frame Requirements

The frame requirements describe the requirements that are independent of the type of steering system, but are determined by the vehicle class that the steering system is developed for, and by legal regulations. All relevant frame requirements are listed in Table 4-1. More frame requirements exist, such as requirements regarding life-time or environmental impacts, but those are not considered during the concept development, which is in focus of this thesis.

The first frame requirement is the ability of the steering system to drive automatically according to SAE level 3 of automation or higher (AD3+).

The legal regulations define the maximum force that the driver has to apply at the steering wheel during correct as well as during incorrect operation of the steering system. Furthermore, they dictate that the driver has to be in control of the vehicle at any time and able to override an automated steering system with a limited effort. For ensuring the functional safety of a mechatronic steering system, the development process has to follow the ISO 26262.

The type of mechanical interfaces of a truck's steering system is similar in most of road-legal trucks worldwide and is depicted in Figure 1-2. The steering wheel in combination with the steering column is the input of the steering gear, the pitman arm and a push rod connect the output of the steering gear with the steering kinematic of the steering axle. As already mentioned before, the on-board network of most of today's trucks is a 24 V network. The target steering system of this thesis should have the same interfaces to enable its use in the majority of today's trucks.

The dimensions of the steering system are limited by the available assembly space inside the truck. Since the steering system is mounted underneath the cab, the space is mainly restricted by the height and the width of the cab. To minimize the influences of the suspension on the steering, the mounting position is also limited.[86]

Another important frame requirement is the weight of the vehicle, which is applied on the vehicle's steering axle. This weight mainly influences the required steering torques, power and energy, as shown in the following steering model.

Table 4-1: Frame requirements of automated truck steering system

Requirement name	**Values, data**	**Description**
Level of driving automation	AD3+	Steering system suitable for SAE level of automation 3 and higher
Maximum steering wheel force/torque[87a]	200 N/50 Nm for correct operation, 450 N/112.5 Nm for incorrect operation	Maximum force required by the driver at the steering wheel;
Override ability[88]/ Maximum force/torque for overriding[87b]	50 N/12.5 Nm	Automated control of the steering is overrideable by the driver at any time with limited effort
Functional safety	ISO 26262 conforming	Development of mechatronic steering system according to latest version of ISO 26262
Output interface	Pitman arm with push rod	Interface between output of steering system and truck's steering kinematic
Electric power supply	24 V DC	Electric power supply by vehicle's power network
Dimensions[89]	550 x 400 x 550 mm^3 (length x height x width)	Dimensions of steering gearbox
Maximum weight at steering axle load[89]	8,500 kg	Maximum static weight at steered axle

[86] Brunner, G.; Negele, K.: Electrification of the steering (2008), p.4.

[87] United Nations: ECE R79 r4 (2018), (a) p.30 | (b) p.14.

[88] United Nations: Convention on Road Traffic (1968), p.11.

[89] Datafor MAN TGS 26:440

4.2 Steering Model

The simulation model, which is used to determine the torque, power and energy requirements of the steering system, is developed in the following subchapter. The overall model structure and the model equations are described first, followed by the evaluation of the simulation model with the help of equivalent driving tests and the transfer of the simulation model to trucks with higher front axle loads.

For the development of the simulation model as well as for the evaluation of the test drives, a horizontal coordinate system is used whose axes are oriented as shown in Figure 4-1. The origin of this coordinate system is the same as the origin of the vehicle-fixed coordinate system, its *x*- and *y*-axis are the projections of the *x*- and *y*-axes of the vehicle-fixed axes to the horizontal ground. The *z*-axis is perpendicular to the other two axes.

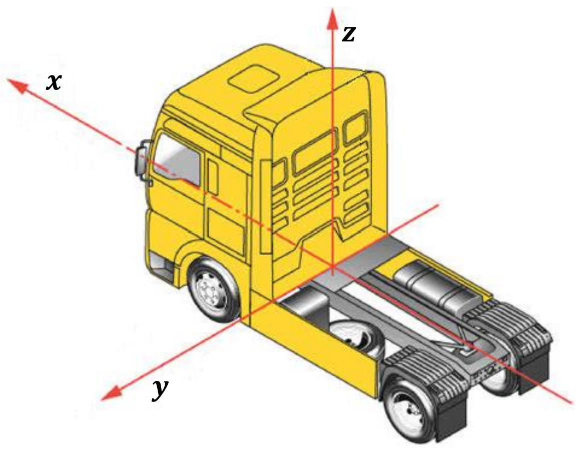

Figure 4-1: Coordinate system[90]

4.2.1 Model Structure

The overall model structure of the simulation model of the truck's steering system is illustrated in Figure 4-2. Besides the different specified parameters of the truck, several parameters measured during defined driving maneuvers are used as input data for the simulation of the corresponding driving maneuvers.

These parameters are the vehicle's velocity (v_V), the torque at the steering wheel (M_H), the accelerations at the vehicle's front axle (fa) ($a_{x,\mathrm{fa}}, a_{y,\mathrm{fa}}, a_{z,\mathrm{fa}}$) and the angle ($\delta_\mathrm{P}$), the angular velocity ($\dot{\delta}_\mathrm{P}$) and the angular acceleration ($\ddot{\delta}_\mathrm{P}$) at the vehicle's pitman arm. These parameters are used to calculate the steering ratios ($i_\mathrm{H2W}, i_\mathrm{H2P}, i_\mathrm{P2W}$), the steering angles ($\delta_\mathrm{W}, \dot{\delta}_\mathrm{W}, \ddot{\delta}_\mathrm{W}$) and the forces at the front axle ($F_{x,\mathrm{fa}}, F_{y,\mathrm{fa}}, F_{z,\mathrm{fa}}$) at the first step.

[90] Hoepke, E.; Breuer, S.: Commercial Vehicle Technology (2016), p.38.

The second step of the simulation is the calculation of the different steering torques and the overall torque at the pitman arm (M_P). The pitman arm torque and the steering angle at the pitman arm are the target results of the simulation model, because the pitman arm is the mechanical interface between the steering system and the vehicle and they describe the overall output torque, power and energy of the steering system. Therefore, the torque, power and energy requirements of the steering system are referred to the pitman arm. The model's first calculation step is described in section 4.2.2 and the second step in section 4.2.3.

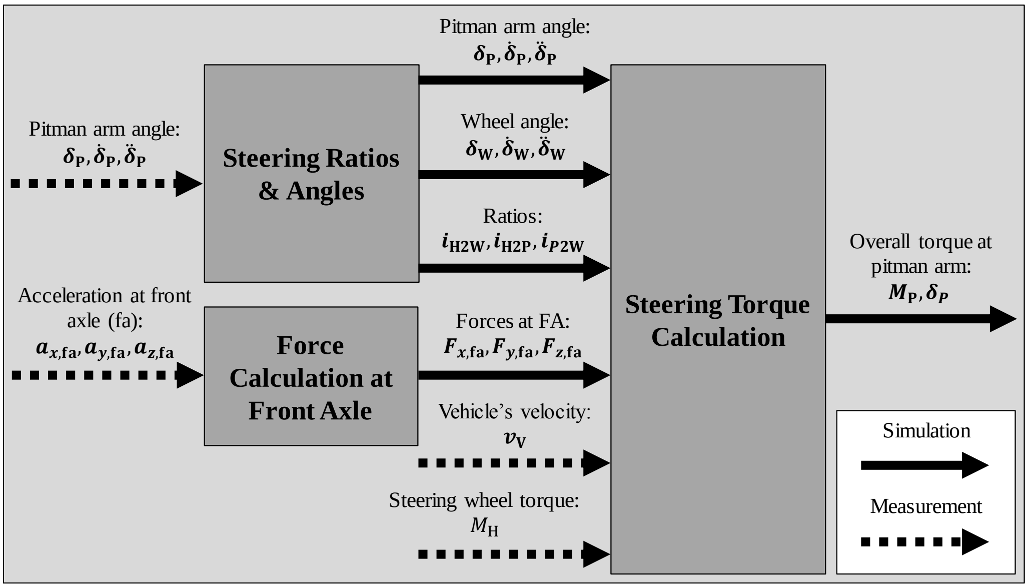

Figure 4-2: Simulation model structure

4.2.2 Calculation of Steering Ratios and Forces at Front Wheels

The calculation of the transmission ratios of the steering system belongs to the first step of the steering model. There are the ratio between the steering wheel and the front wheels (i_H2W), the ratio between the steering wheel and the pitman arm (i_H2P) and the ratio between the pitman arm and the front wheels (i_P2W). The steering ratios are defined in equations (4-1) to (4-5):

$$i_\mathrm{H2W} = \frac{\delta_\mathrm{H}}{\delta_\mathrm{W}} \tag{4-1}$$

$$i_\mathrm{H2P} = \frac{\delta_\mathrm{H}}{\delta_\mathrm{P}} \tag{4-2}$$

$$i_\mathrm{P2W} = \frac{\delta_\mathrm{P}}{\delta_\mathrm{W}} \tag{4-3}$$

The other part of the first calculation step determines the forces at the wheels of the front axle, which strongly influence the required torque, power and energy of the steering system. To be allowed to simplify the calculation of the dynamic forces at the front wheels, the condition in equation (4-4) that the coupling mass (m_C) is zero, has to be fulfilled. The coupling mass is defined by the vehicle mass (m_{V}), the vehicle's yaw inertia (J_z) and the distance from the vehicle's center of gravity to the front axle (ℓ_{f}) and to the rear axle (ℓ_{r}).[91]

$$m_{\mathrm{C}} = m_{\mathrm{V}} - \frac{J_z}{\ell_{\mathrm{f}}\ell_{\mathrm{r}}} = 0 \tag{4-4}$$

Since the considered vehicle[92] fulfills this condition, the movement of the mass at the front axle ($m_{\mathrm{V,fa}}$) is independent of the movement of the mass at the rear axle.[93] This allows to calculate the forces at the wheels of the front axle ($F_{x,\mathrm{fl/fr}}, F_{y,\mathrm{fl/fr}}, F_{z,\mathrm{fl/fr}}$) by the vehicle's mass at the front axle ($m_{\mathrm{V,fa}}$) and the accelerations at the front axle in each direction ($a_{x,\mathrm{fa}}, a_{y,\mathrm{fa}}, a_{z,\mathrm{fa}}$). The following three equations are used to calculate the forces:

$$F_{x,\mathrm{fa}} = F_{x,\mathrm{fl}} + F_{x,\mathrm{fr}} = m_{\mathrm{V,fa}} \cdot a_{x,\mathrm{fa}} \tag{4-5}$$

$$F_{y,\mathrm{fa}} = F_{y,\mathrm{fl}} + F_{y,\mathrm{fr}} = m_{\mathrm{V,fa}} \cdot a_{y,\mathrm{fa}} \tag{4-6}$$

$$F_{z,\mathrm{fa}} = F_{z,\mathrm{fl}} + F_{z,\mathrm{fr}} = m_{\mathrm{V,FA}} \cdot a_{z,\mathrm{fa}} \tag{4-7}$$

4.2.3 Steering Torque Calculation

The second part of the steering system simulation model is the calculation of the different steering torques and of the overall torque at the pitman arm. A schematic illustration of the steering system with the applied torques and the different angles is shown in Figure 4-3.

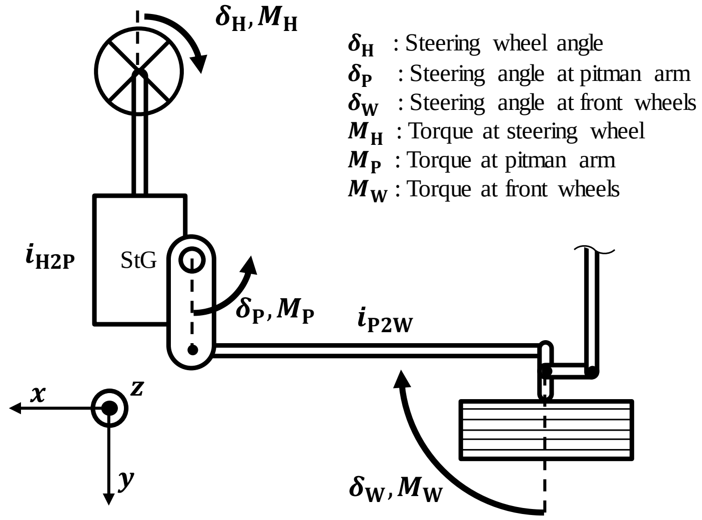

Figure 4-3: Angles and torques at steering system

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[91] Mitschke, M.; Wallentowitz, H.: Dynamics of motorvehicles (2014), p.298.

[92] Vehicle data see in appendix A.2.1.

[93] Mitschke, M.; Wallentowitz, H.: Dynamics of motorvehicles (2014), p.298.

Only the front left wheel is depicted here, but the torque at the wheel (M_{W}) is the sum of the torques at both front wheels and the steering angle at the front left and front right wheel are assumed to be equal (δ_{W}). The values of the steering angle and torque at the steering wheel ($\delta_{\text{H}}, M_{\text{H}}$), at the pitman arm ($\delta_{\text{P}}, M_{\text{P}}$) and at the wheels ($\delta_{\text{W}}, M_{\text{W}}$) are assumed to be positive as shown in Figure 4-3. The ratio between the steering wheel and the pitman arm (i_{H2P}) as well as the ratio between the pitman arm and the front wheels (i_{P2W}) are marked, too.

With the help of Figure 4-3, equation (4-8) is set up, which describes the sum of steering torques at the pitman arm. The torque at the steering wheel (M_{H}) and the torque at the front wheels (M_{W}) are both applied to the pitman arm via a transmission ratio, whereas the inertia torque (M_{J}), the friction torque (M_{F}) and the damping torque (M_{D}) are modeled to be applied directly at the pitman arm. Apart from the steering wheel torque, all torques counteract the steering movement.

$$M_{\text{P}} = M_J + M_{\text{F}} + M_{\text{D}} + \frac{M_{\text{W}}}{i_{\text{P2W}}} + M_{\text{H}} \cdot i_{\text{H2P}} \tag{4-8}$$

Since the steering wheel torque (M_{H}) is an input of the simulation model, a calculation of this parameter is not required.

The torque caused by the inertia of the steering system (M_J) is calculated with equation (4-9) by the inertia of the overall steering system reduced to the pitman arm ($J_{\text{eff,P}}$) and the angular acceleration of the pitman arm ($\ddot{\delta}_{\text{P}}$).

$$M_J = -J_{\text{eff,P}} \cdot \ddot{\delta}_{\text{P}} \tag{4-9}$$

The torque at the pitman arm induced due to the friction inside the steering system (M_{F}) is described with equation (4-10) assuming Coulomb friction only. It depends on an empirical friction constant ($\widehat{M}_{\text{F}}$) and the sign of the angular velocity of the pitman arm ($\dot{\delta}_{\text{P}}$).

$$M_{\text{F}} = -\widehat{M}_{\text{F}} \cdot \operatorname{sgn}\left(\dot{\delta}_{\text{P}}\right) \tag{4-10}$$

The damping-induced torque (M_{D}) also depends on an empirical damping value (d_{D}) and on the angular velocity of the pitman arm ($\dot{\delta}_{\text{P}}$). It is calculated by equation (4-11).

$$M_{\text{D}} = -d_{\text{D}} \cdot \dot{\delta}_{\text{P}} \tag{4-11}$$

The last parameter of equation (4-8) is the torque applied at the steered front wheels of the truck (M_{W}). It is the sum of the torques induced by differential braking (M_{F_x}), the torque due to the lateral forces (M_{F_y}), the lifting torque (M_{F_z}) and the bore torque (M_{Bore}). Equation (4-12) describes this correlation.[94]

$$M_{\text{W}} = -M_{F_x} - M_{F_y} - M_{F_z} - M_{\text{Bore}} \tag{4-12}$$

[94] Reimpell, J.; Betzler, J. W.: Basics Chassis (2005), pp. 229–251.

All four parts of the torque at the front wheels (M_W) strongly depend on the geometry of the truck's steered front axle. The geometric parameters, which are crucial for the calculation of the different torques, are described in Figure 4-4. The inclination of the wheel in the *y*-*z*-plane is called camber angle (γ), whereas the inclination of the kingpin in this plane is named kingpin angle (σ). The scrub radius (r_0) defines the distance in *y*-direction between the wheel contact patch and the point where the kingpin axis crosses the road surface. The distance between those two points in the *x*-direction is called the castor offset (r_τ) and the inclination of the kingpin in the *x*-*z*-plane is the castor angle (τ).

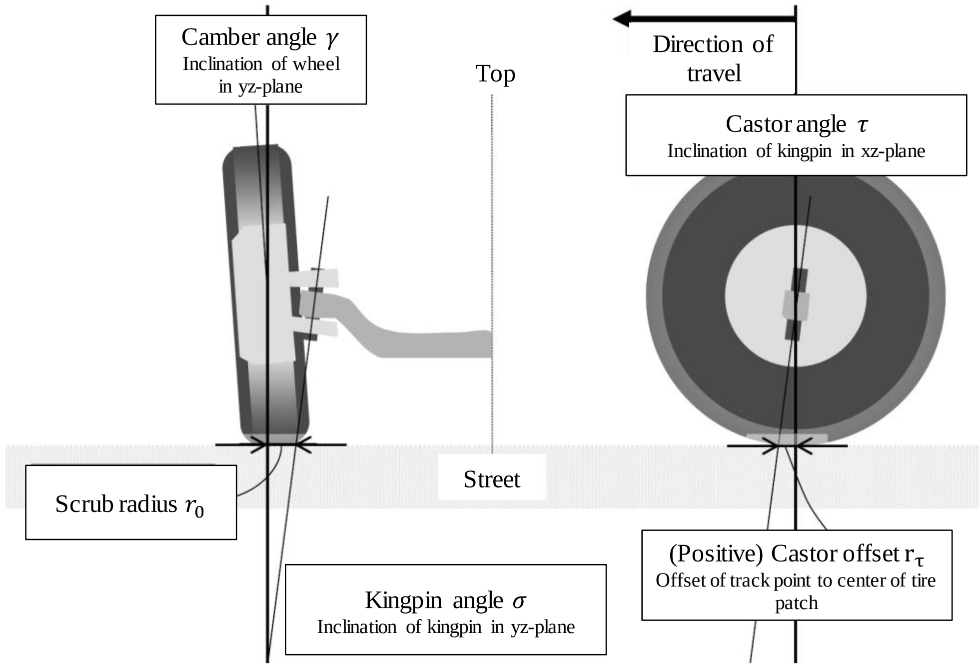


Figure 4-4: Illustration of the crucial geometric parameters at the steered front axle[95]

The steering toque induced by differential braking (M_{F_x}) is described by equation (4-13) and relies on the difference between the braking force at the left ($F_{x,\mathrm{fl}}$) and at the right wheel ($F_{x,\mathrm{fr}}$), the scrub radius (r_0), the kingpin angle (σ) and the castor angle (τ).[96a]

$$M_{F_x} = \left(F_{x,\mathrm{fl}} - F_{x,\mathrm{fr}}\right) \cdot r_0 \cdot \cos\sigma \cdot \cos\tau \tag{4-13}$$

The sum of the lateral forces at the front left ($F_{y,\mathrm{fl}}$) and right wheel ($F_{y,\mathrm{fl}}$) causes a torque at the wheels (M_{F_y}), which is calculated by equation (4-14). The lever arm of the lateral forces depends on the tire's tread length (ℓ_{Tread}), the kingpin angle (σ) and the castor angle (τ).[96b]

$$M_{F_y} = \left(F_{y,\mathrm{fl}} + F_{y,\mathrm{fr}}\right) \cdot \left(\frac{\ell_{\mathrm{Tread}}}{6} + r_\tau\right) \cdot \cos\sigma \cdot \cos\tau \tag{4-14}$$

[95] Following Hilgers, M.: Chassis and Axles (2016), p.22.

[96] Reimpell, J.; Betzler, J. W.: Basics Chassis (2005), (a) p.236 | (b) p.246.

The lifting toque (M_{F_z}) is caused by the wheel load at the front left ($F_{z,\mathrm{fl}}$) and right wheel ($F_{z,\mathrm{fr}}$), which counteract a deviation from the center position of the steering. It is calculated by equation (4-15). The kingpin angle (σ) and the castor angle (τ) influence how far the truck's front axle is lifted due to the steering angle (δ_W). The lever (ℓ_{F_z}), where the counter-force is applied, depends on the scrub radius (r_0), the dynamic wheel radius (r_dyn) and the kingpin angle (σ).[97]

$$M_{F_z} = \left(F_{z,\mathrm{fl}} + F_{z,\mathrm{fr}}\right) \sin\sigma \cdot \cos\tau \cdot \sin\delta_\mathrm{W} \cdot \ell_{F_z} \tag{4-15}$$

$$\ell_{F_z} = \left[\left(r_0 + r_\mathrm{dyn} \cdot \tan\sigma\right) \cdot \cos\sigma\right] \tag{4-16}$$

Equation (4-17) describes the bore torque (M_Bore) at the wheels, which is caused by the friction between the tire and the road surface and the tire's elasticity. Modeling the bore torque is more complex than for the other torques, because there is no steady function to describe it. Hence, several equations are demanded to model the bore torque. At standstill and at very low velocities, the bore torque acts like an in-line connection of a spring and a Coulomb friction element, as calculated with the lower equation of (4-17) and (4-19). As soon as the vehicle moves, the bore torque decreases significantly and depends on the tire's tread width (w_Tread) and the longitudinal tire stiffness ($(dF_x/d\lambda_x)|_{\lambda_x=0}$) with the longitudinal slip (λ_x) and is calculated by the upper equation of (4-17) and (4-18). The bore torque at standstill depends on the tire's bore stiffness (c_Bore) and the bore angle (δ_Bore), which itself describes the torsion angle of the tire tread compared to the wheel. The bore angle is calculated with equation (4-19) by integration of the wheel angular velocity ($\dot{\delta}_\mathrm{W}$), but it is limited to a maximum value ($\delta_\mathrm{Bore,max}$). The integral is reset to zero if the bore torque defined by (4-17) is zero.[98]

$$M_\mathrm{Bore} = \begin{cases} b \cdot \dfrac{\dot{\delta}_\mathrm{W}}{|v_\mathrm{V}|} & \text{if } \left| b \cdot \dfrac{\dot{\delta}_\mathrm{W}}{v_\mathrm{V}} \right| \leq |\, c_\mathrm{Bore} \cdot \delta_\mathrm{Bore}| \\ c_\mathrm{Bore} \cdot \delta_\mathrm{Bore} & \text{else} \end{cases} \tag{4-17}$$

$$b = \frac{1}{12} w_\mathrm{Tread}^2 \left.\frac{dF_x}{d\lambda_x}\right|_{\lambda_x=0} \tag{4-18}$$

$$\delta_\mathrm{Bore} = \begin{cases} \displaystyle\int_{t(M_\mathrm{Bore}=0)}^{t} \dot{\delta}_\mathrm{W}\, dt & \text{if } \left| \displaystyle\int_{t_0}^{t} \dot{\delta}_\mathrm{W}\, dt \right| \leq \delta_\mathrm{Bore,max} \\ \operatorname{sgn} \dot{\delta}_\mathrm{W} \left|\delta_\mathrm{Bore,max}\right| & \text{else} \end{cases} \tag{4-19}$$

The overall torque at the pitman arm is used to calculate the required steering energy (E_P) and power (P_P) in the same way for the measured data and for the simulated data. The steering energy (E_P) is calculated with equation (4-20) by integration of the steering torque at the

[97] Reimpell, J.; Betzler, J. W.: Basics Chassis (2005), p.246.

[98] Hesse, B.: Dissertation, Interferences between driving dynamics and power network (2011), pp. 47–49.

pitman arm (M_P) over the steering angle at the pitman arm (δ_P). The required steering power (P_P) is the time derivative of the steering energy (E_P), as described by equation (4-21).

$$E_\mathrm{P} = \int M_\mathrm{P}\, d\delta_\mathrm{P} \tag{4-20}$$

$$P_\mathrm{P} = \frac{dE_\mathrm{P}}{dt} = M_\mathrm{P}\dot{\delta}_\mathrm{P} \tag{4-21}$$

4.2.4 Evaluation of Steering Model

For the evaluation of the previously described steering model, the model is parameterized with the values of a fully loaded 12-t test truck[99]. This test truck was set up with measurement equipment to measure all the parameters listed in Figure 4-2. The steering torque and angle at the steering wheel were recorded by a measurement steering wheel, whereas the output torque of the steering system was measured with strain gauges at the pitman arm. The steering angle at the pitman arm was calculated from the steering wheel angle and the measured ratio (i_H2P). The accelerations in all three directions, the truck's velocity and its GPS position were recorded additionally.

The model equations and parameters for the steering torque caused by the friction inside the steering system (M_F) or by the lifting of the vehicle (M_{F_z}) are determined or evaluated experimentally in advance. The friction torque (M_F) is measured during steering movements with the front axle jacked up. To measure the lifting torque (M_{F_z}), the front wheels are positioned on so-called turntables so that no bore torque is superimposed on the lifting torque during the steering at standstill. The model equations for the bore torque (M_Bore), in turn, are evaluated during steering at standstill by subtracting the friction torque (M_F) and the lifting torque (M_{F_z}). The torques due to the damping (M_D) and the inertia (M_J) of the steering system are evaluated by test drives with a sinusoidal steering angle input. The model equations of the steering torques induced by the longitudinal forces (M_{F_x}) and by the lateral forces at the wheels (M_{F_y}) are evaluated by different driving maneuvers.

The performed driving maneuvers, which are listed in Table 4-2, cover the whole range of possible steering torques, steering angles and steering angular velocities, whereby they also cover all possible steering powers. Hence, these driving maneuvers are suitable for the evaluation of the developed steering system simulation model. For each driving maneuver, several test drives were carried out. The values in Table 4-2 correspond to the maximum values that occurred during these test drives.

The first three driving maneuvers are artificial driving maneuvers, which are used only for covering the whole range of steering torque and steering power. The driving maneuver

[99] Vehicle data see Table A-2 in the Appendix.

Standstill describes steering at standstill with the highest possible steering velocity over the entire steering angle range. The Figure 8 is a driving maneuver at low velocity, at which an eight is driven with maximum steering angle. The sine with 2 Hz describes a driving maneuver in which a frequency of 2 Hz at an amplitude of approximately 60° is applied at the steering wheel.

The highway driving, the country road, the mountain pass and the city driving all summarize driving maneuvers on a defined type of road. For each of those scenarios, several test drives are performed on different routes in and around Darmstadt in order to cover as much different potential maneuvers as possible. Of course, there is no completeness claimed. The same approach is valid for the highway access/exit and the highway interchange.

Table 4-2: Representative driving maneuvers conducted with by fully loaded 12-t truck

Driving maneuver	$\boldsymbol{v}_{\mathbf{max}}$ in $\frac{\mathrm{m}}{\mathrm{s}}$	$\boldsymbol{a}_{\boldsymbol{y},\mathbf{max}}$ in $\frac{\mathrm{m}}{\mathrm{s}^2}$	$\boldsymbol{\kappa}_{\mathbf{max}}$ in $\frac{1}{\mathrm{m}}$
Standstill	0	0	0
Figure 8	5	3.01	0.20
Sine with 2 Hz	17	2.96	-
Highway driving	25	2.45	$2.13 \cdot 10^{-3}$
Country road	20	3.43	$7.14 \cdot 10^{-3}$
Mountain pass	17	3.53	$2.22 \cdot 10^{-2}$
City driving	14	3.24	$2.00 \cdot 10^{-2}$
Turn-off (out of city)	-	5.00	$1.43 \cdot 10^{-2}$
Lane change	25	2.16	-
Avoidance maneuver	17	5.00	-
Emergency stopping bay[100]	17	1.57	-
Highway access/exit	-	3.14	$6.67 \cdot 10^{-3}$
Highway interchange	25	3.14	$2.50 \cdot 10^{-3}$

The maneuvers turn-off, lane change, avoidance maneuver and emergency stopping bay were all performed on a test track at different initial velocities and different steering angle velocities, with the aim of determining the highest requirements for those driving maneuvers.

The simulation model is used for dimensioning the steering system in the following subchapters. Therefore, the evaluation of the simulation model compares the maximum steering torque, energy and power measured during each driving maneuver with the simulated values. For the dimensioning of the system, an accuracy of the model of ± 10 % seems to be sufficient. The simulation model is considered as evaluated if the deviation of a simulated value to its corresponding measured value is less than 10 % of the measured value.

[100] Forschungsgesellschaft für Straßen- und Verkehrswesen: Guideline for rural roads (2012), p.91.

The comparison of the maximum steering torque at the pitman arm ($M_{\mathrm{P,max}}$) is shown in Figure 4-5. The deviation of the simulated steering torque at the pitman arm from the measured is within the 10 % range for all relevant maneuvers. The biggest steering torque occurs during steering in standstill and the lowest steering torque is required for a lane change and for steering into an emergency stopping bay.

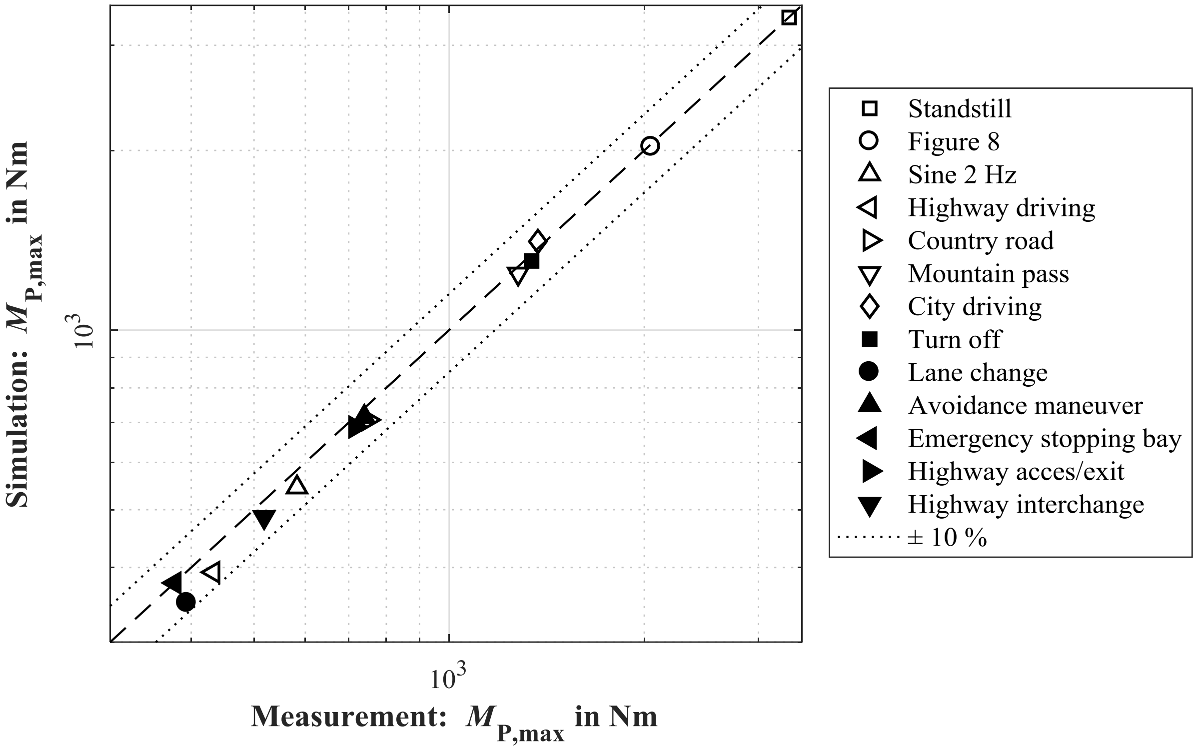


Figure 4-5: Comparison of measured and simulated maximum torque at pitman arm ($M_{\mathrm{P,max}}$)

A major portion of the required steering torque in all driving maneuvers except for steering at standstill is the torque due to the lateral forces at the front wheels (M_{F_y}). The torque caused by the inertia (M_J) has a main influence during the sine maneuver and the avoidance maneuver, whereas the lifting torque (M_{F_z}) makes a major contribution of the required steering torque during the maneuvers with a high maximum steering angle, which are the turn-off maneuver, the city driving, the Figure 8 and the steering at standstill. The bore torque (M_{Bore}) predominates in the maneuvers with extremely low velocities up to standstill. Those are the steering at standstill and the turn-off maneuver.

The maximum required steering energies ($E_{\mathrm{P,max}}$) during the relevant driving maneuvers are compared in Figure 4-6. The steering energy is calculated with equation (4-20). The discrete driving maneuvers, which are finished after a defined period, such as the turn-off maneuver, are distinguished from the continuous driving maneuvers without a defined end, such as highway driving. The required steering energy during the continuous maneuvers is referred to the driving distance. The maximum steering energy required during a driving distance of 1,000 m is used for the comparison. The maneuvers standstill and sine 2 Hz are not considered here, because only the steering torque and the steering power during those maneuvers are relevant for the design of the steering system. The maneuver with the most required steering energy is the figure 8, whereas during highway driving the least energy is necessary for steering. All simulated results deviate less than 10 % from the measured values.

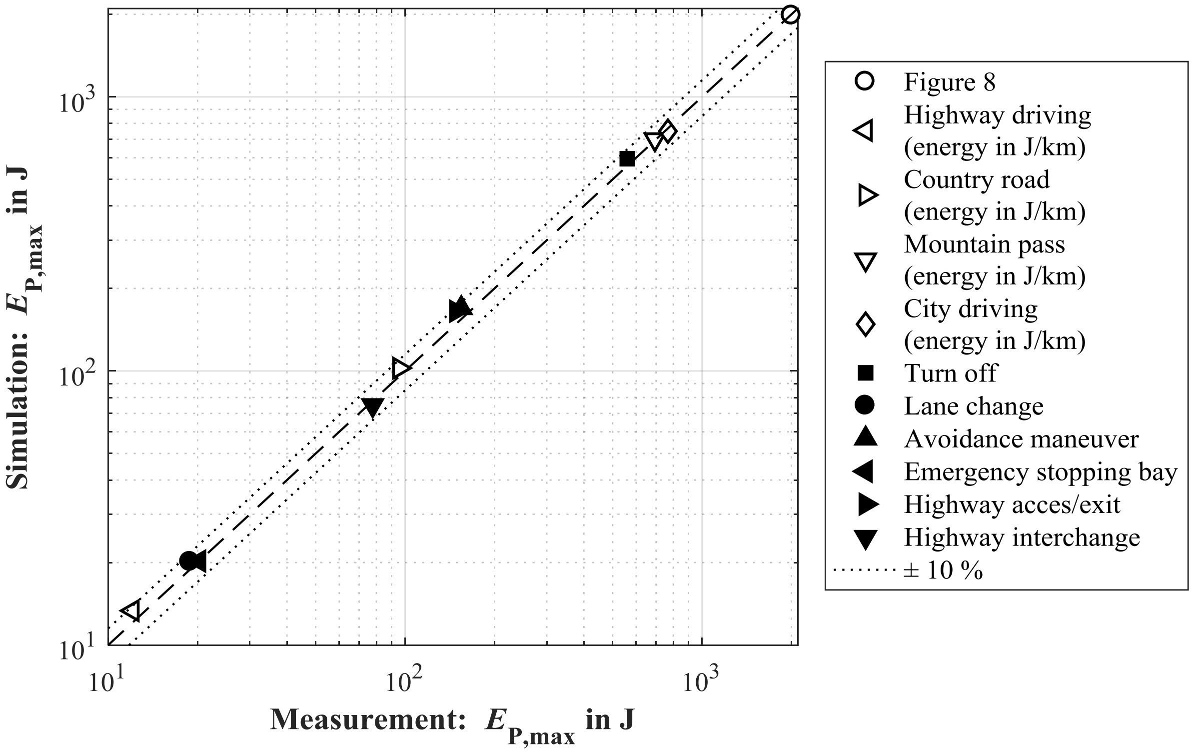


Figure 4-6: Comparison of measured and simulated maximum steering energy ($E_{\mathrm{P,max}}$)

The evaluation of the steering power, calculated with equation (4-21) of the simulation model, is illustrated in Figure 4-7. Because the maximum required steering power ($P_{\mathrm{P,max}}$) is independent of the duration of a driving maneuver, all relevant driving maneuvers are analyzed equally. All steering power results from the simulation deviate less than 10 % from the measured steering power for each relevant driving maneuver. Highway driving requires the least steering power, whereas steering in standstill necessitates the most steering power.

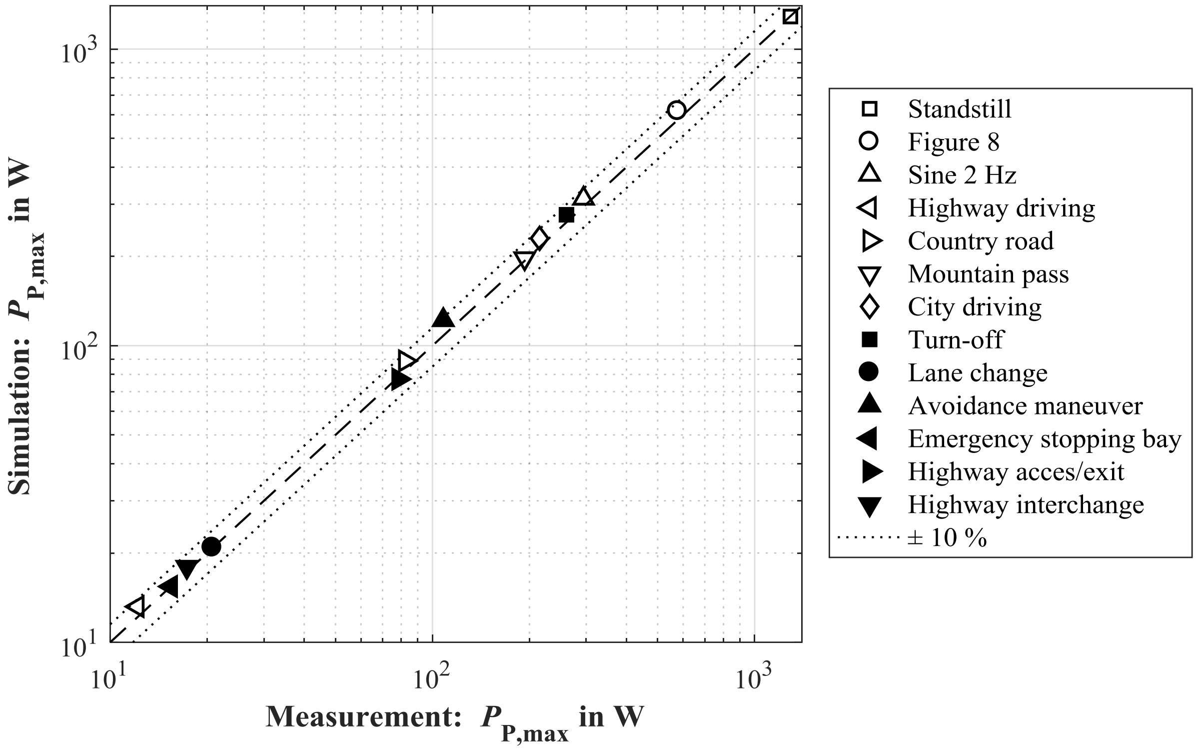


Figure 4-7: Comparison of measured and simulated maximum steering power ($P_{\mathrm{P,max}}$)

Because the simulated steering torques, energies and powers are all within the defined deviation range for the defined driving maneuvers, which covers all possible values of steering torque, energy and power, the simulation model is regarded as valid for the purpose of the further investigations.

4.2.5 Transfer to Higher Front Axle Loads

Because the truck used has a lower front axle load than the target range, the simulation model is applied in order to determine the requirements for the target 26-t truck. Thereby, the steering torque, energy and power requirements are derived for a truck with a maximum weight at the front axle of 8,500 kg as defined in the frame requirements before. The values of the other vehicle parameters of the considered 26-t truck are listed in the appendix[101].

[101] Vehicle data see Table A-3 in the Appendix.

The input data of the simulation model, which were recorded during test drives, are independent of the vehicle parameters. Since the test truck and the target truck have the same steering kinematics, it is assumed that the steering angles at the front wheels (δ_{W}) applied during the driving maneuvers and the steering ratios are the same for the 12-t test truck and for the 26-t truck. Thereby, the measured parameters of the angle at the pitman arm (δ_{P}), the vehicle's accelerations ($a_{x,\mathrm{fa}}, a_{y,\mathrm{fa}}, a_{z,\mathrm{fa}}$) and velocity ($v_{\mathrm{V}}$) as well as the steering wheel torque (M_{H}) are used unmodified as model input for the 26-t truck.

The vehicle parameters that have changed during the switch to a 26-t truck and influence the required steering torque, energy and power, are listed in Table 4-3. The geometric dimensions and the mass of the truck change as well as the yaw inertia and thus influence the steering requirements. As described in section 4.2.3, the parameters of the wheel and tire influence the steering torque as well as the suspension kinematics of the front axle. Both of them as well as the inertia of the steering system change between a 12-t and a 26-t truck, the latter due to the more massive components required in the steering system.

Table 4-3: Vehicle parameters influencing the required steering torque, energy and power

Parameter	Symbol
Vehicle	
Mass	m_{V}
Mass at front axle	$m_{\mathrm{V,fa}}$
Wheelbase	ℓ_{W}
Distance from front axle to CoG	ℓ_{f}
Track width	w_{W}
Height of CoG	h_{CoG}
Yaw inertia	J_z
Wheel/Tire	
Tire dimensions	-
Dynamic wheel radius	r_{dyn}
Tire pressure	p_{Tire}
Tire tread length	ℓ_{tread}
Tire bore stiffness	c_{Bore}
Maximum bore angle	$\delta_{\mathrm{Bore,max}}$
Bore constant	b
Suspension	
Castor angle	τ
Kingpin angle	σ
Scrub radius	r_0
Castor offset	r_τ
Steering system	
Inertia of steering system at pitman arm	$J_{\mathrm{eff,P}}$

4.3 Operational Requirements

The torque and power requirements of the steering system during correct operation without any fault inside the steering system are calculated based on representative driving maneuvers. Besides the driving on the different types of roads, such as highway, country roads, mountain pass and city roads, different maneuvers have been carried out. The maneuver Sine 2 Hz is representative for a hectic steering maneuver and Figure 8 for maneuvering at low velocities e.g. on a freight yard. The maneuver Standstill describes the steering at standstill.

Figure 4-8 shows the steering torque and power requirements for the described driving maneuvers. The maximum steering torque of $8{,}250\ \mathrm{Nm}$ and the maximum steering power of $3{,}410\ \mathrm{W}$ are required for the steering at standstill, whereas the maximum pitman arm angular velocity of $0.86\ \mathrm{rad/s}$ occurs during the sine steering maneuver with a steering frequency of 2 Hz. Highway driving requires the lowest steering torque and power.

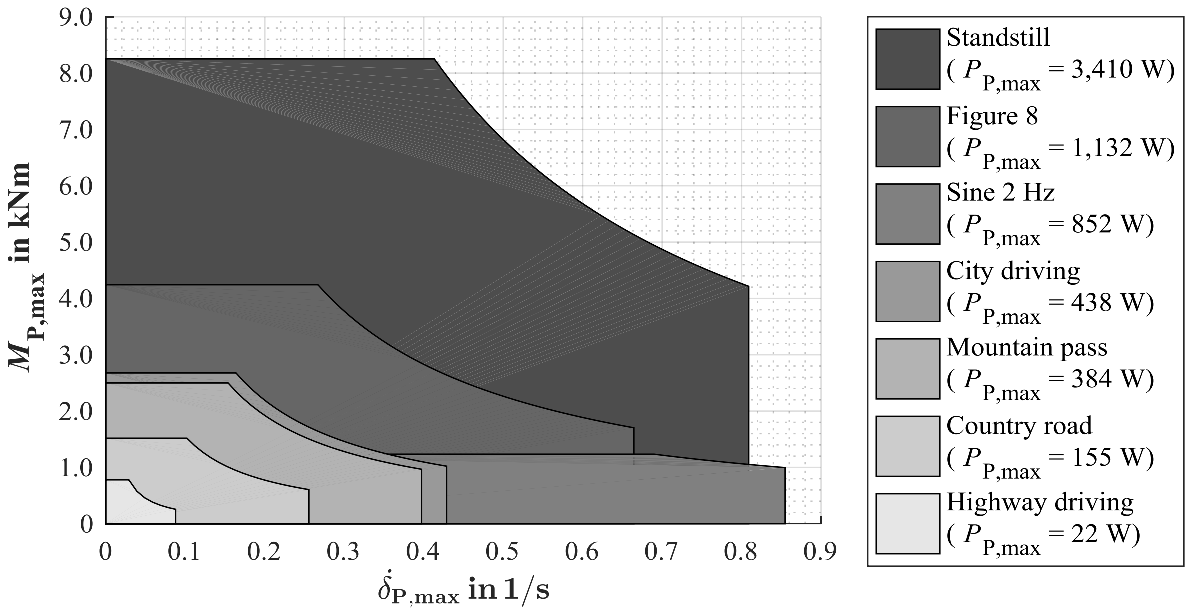


Figure 4-8: Required steering torque and power for 26-t truck

The steering energy required for the different maneuvers is not relevant during the correct operation of the steering system, because without any system fault, the power steering pump and the generator, which both are powered by the truck’s engine, provide the required steering power continuously. No energy storage is necessary here.

4.4 Redundancy Requirements

Besides the steering torque, energy and power requirements of the steering system during correct operation, the requirements for the incorrect operation due to a system fault are important to be considered for a steering system for AD3+ of trucks as well. Whereas a fail-silent steering system is sufficient for AD2-, a driver-independent fallback level is required for the steering functionality for AD3+, because in the event of a steering system fault the driver is available as immediate fallback for AD2-, but not for AD3+.[102]

The requirements to this steering redundancy are determined in the following. Therefore, safe states for different road classes are defined in section 4.4.1 first. The fallback steering system has to be able to transfer the vehicle to the defined safe states in case of a system fault. The maneuvers that are necessary to transfer the vehicle to the specific safe states for the different road classes are derived in section 4.4.2. Based on these relevant driving maneuvers, the fallback steering torque, energy and power requirements for the steering redundancy are determined with the previously described steering model in subchapter 4.2.

The content and results of the following sections 4.4.1, 4.4.2, 4.2.3 and 4.4.4 were published previously.[103]

4.4.1 Definition of Safe States

Since an automated vehicle contains a lot of E/E systems (electric/electronic), the definition of its safe states base on the ISO 26262 on functional safety of E/E systems of vehicles. A safe state is defined as an “operating mode of (a vehicle) without an unreasonable level of risk”[104], whereas risk is defined as the “combination of the probability of occurrence of harm and the severity of that harm” and harm is the “physical injury or damage to the health of persons”[104]. In the context of automated driving, the safe state is described as an operating mode, in which no unreasonable risk occurs for all persons involved in road traffic.[105] Thus, the internal system state as well as the environment of the vehicle are important for the safe state.

Based on the definitions from the ISO 26262, eight requirements of the safe state are determined here and listed in Table 4-4. The first requirement is the most important and a high-level requirement. All the other requirements serve to fulfill this superior requirement. The requirements no. 2 to no. 6 are the important requirements for the determination of the redundancy requirements. The safe state in each specific driving situation depends on those

[102] Matthaei, R. et al.: Autonomous Driving (2016), p.1544.

[103] Herold, M. et al.: Power Requirements for RASS (2019).

[104] International Organization for Standardization: ISO 26262-1 (2018), p.21.

[105] Reschka, A.: Dissertation, Safe Operation of Automated Vehicle, p.54.

five requirements. Therefore, we separate the safe state for city roads, country roads, and highways.

Table 4-4: Requirements to the safe state of an automated driving vehicle[106]

No.	Requirement
1	No hazard for passengers, other road users, pedestrians or for the environment
2	Vehicle stands still
3	Visibility to other road users bigger than required stopping visibility
4	Relative velocity to other road users less than 70 km/h
5	No blocking of rescue routes
6	No blocking of bridges, tunnels, intersections or roundabouts
7	Protection of the stopping place and warning of other road users
8	Emergency call (if necessary)

In order to define a safe state for each different type of road, it is important to know all the specific properties of each type, especially according to the possibilities for a safe stop of the vehicle. The German guideline for the construction of city roads[107a], the guideline for the construction of country roads[108] and the guideline for the construction of highways[109] describe the properties of the roads and are used to define three different road classes (RC) according to the possibility for a safe stop.

The different types of roads are characterized into three road classes, according to the criteria in Table 4-5. The number of lanes, counting both directions, the availability of a hard shoulder or an emergency stopping bay, the speed limit, the maximum curvature ($\kappa_{\max}$) and the maximum required stopping visibility are used for the classification. The first road class (RC1) contains the roads with a permanent hard shoulder, i.e. highways except urban highways. The second road class (RC2) describes the roads with emergency stopping bays instead of a permanent hard shoulder, i.e. urban highways and big country roads with at least two lanes in one direction. The last road class (RC1), which requires steering maneuvers to get to a safe state, contains the roads without a hard shoulder and without any emergency stopping bays. City roads are not part of these three classes, because there are always speed limits below 70 km/h[107b]. A relative velocity between the stationary vehicle and other road users of more than 70 km/h is not possible. Hence, the transition to the safe state is always an immediate braking maneuver and no steering is required for this class. The claim is that the automated vehicle knows what the safe states and the maximum distances between the single safe states are and how it gets there, depending on the road class.

[106] Reschka, A.: Safety Concept for Autonomous Vehicles (2016), pp. 473–484.

[107] Forschungsgesellschaft für Straßen- und Verkehrswesen: Guideline for urban roads (2006), (a) - | (b) p.13.

[108] Forschungsgesellschaft für Straßen- und Verkehrswesen: Guideline for rural roads (2012).

[109] Forschungsgesellschaft für Straßen- und Verkehrswesen: Guideline for highways (2008).

Table 4-5: Characterization of road classes[110,111,112a]

Road class	1	2	3
Number of lanes (both directions)	≥ 4	≥ 3	2
Hard shoulder available?	yes	partially	none
Emergency stopping bay available?	-	at least every 1,000 m	none
Speed limit	none	≥ 100 km/h	≤ 100 km/h
Maximum curvature ($\kappa_{\max}$)	$2.13 \cdot 10^{-3} \mathrm{m}^{-1}$	$3.57 \cdot 10^{-3} \mathrm{m}^{-1}$	$5.00 \cdot 10^{-3} \mathrm{m}^{-1}$
Maximum required stopping visibility	250 m	190 m	160 m

The safe state of RC1 with a hard shoulder is the standstill on the hard shoulder. Usually, no other road user drives on the hard shoulder, thus there is no relative velocity to others. In addition, the stopping visibility has no influence on this safe state and no bridge, tunnel or rescue route is blocked here. The relevant maneuvers to reach the safe state depends on the number of lanes of the road and on which road the vehicle currently drives when a fault potentially occurs and thus the transition to the safe state is required. If the vehicle is not on the lane next to the hard shoulder, one or more lane changes and the change to the hard shoulder are the relevant maneuvers to reach the safe state. Because there are no hard shoulders in areas of highway accesses or exits, a change to the hard shoulder could be temporarily not possible. In that case, the vehicle needs to drive on for a defined distance until the hard shoulder is available again.

Driving into and stopping inside an emergency stopping bay is the safe state of RC2, where such a stopping bay is intended to be every 1,000 m. The emergency stopping bays are at least 84 m long and 3 m wide.[112b] Because the emergency stopping bay is no continuous lane, no other road user is able to drive on it, whereby the risk seems to be lower standing inside an emergency stopping bay as standing on a hard shoulder. The relevant maneuvers to reach the safe state inside a stopping bay contain one or more lane changes as well, the drive and stopping maneuver into the stooping bay and the required drive on to the next available stopping bay. It is also possible that there are hard shoulders temporarily available on this road class, but for the design requirements of the steering system, the highest fallback requirements are used, which occur for the drive into an emergency stopping bay for this road class.

Because of the missing hard shoulders and the missing emergency stopping bays, the safe state of RC3 is not obvious. A safe stop at the side of the road is usually not possible due to a relative velocity to the other road users above 70 km/h. However, in practice junctions frequently appear on this road class, whereby a turn-off to a side road with a speed limit

[110] Forschungsgesellschaft für Straßen- und Verkehrswesen: Guideline for urban roads (2006).

[111] Forschungsgesellschaft für Straßen- und Verkehrswesen: Guideline for highways (2008).

[112] Forschungsgesellschaft für Straßen- und Verkehrswesen: Guideline for rural roads (2012) , (a) - | (b) p.91.

lower than 70 km/h or to a road with a hard shoulder or emergency stopping bays is possible. A safe stop on such a side road, e.g. a city road, represents the safe state in this road class. Hence, the relevant maneuvers are the drive on until the next turn-off to a side road and the turn-off maneuver itself. Of course, it is possible that there is a stopping bay or a parking lot at this road class as well, but this is an exception and the fallback requirements for the steering system are higher for the turn-off maneuver and thus are the crucial design requirements.

4.4.2 Relevant Driving Maneuvers

Based on the previously defined safe states for the three different road classes, the different relevant driving maneuvers, listed in Table 4-6, are simulated for a 26-t truck to calculate the required steering torque, steering power and steering energy.

Besides the previously mentioned relevant driving maneuvers, the "avoidance maneuver" is also considered as relevant, because it is possible at any time and at any road class that an avoidance maneuver becomes necessary. The different types of maneuvers are simulated several times with input data from driving tests on several roads, i.e. different routes of country roads, different highways and different city routes were tested. For each type of maneuver, the biggest occurring values for torque, power and energy are used to determine the fallback requirements. This approach is supposed to cover the worst case of each type of maneuver. The measured data of the several maneuvers are combined according to the definition of the safe state in the previous section to determine the final fallback requirements for each road class. Hereby, the most critical moment for the failure of the steering system is assumed. However, a complete cover of all possible situations is not guaranteed of course.

Table 4-6: Driving maneuvers relevant to reach the safe state

Maneuver type	$\boldsymbol{v}_{\mathbf{max}}$ in $\frac{\mathrm{m}}{\mathrm{s}}$	$\boldsymbol{a}_{\boldsymbol{y},\mathbf{max}}$ in $\frac{\mathrm{m}}{\mathrm{s}^2}$	$\boldsymbol{\kappa}_{\mathbf{max}}$ in $\frac{1}{\mathrm{m}}$
Turn-off (out of city)	-	5.00	$1.43 \cdot 10^{-2}$
Lane change	25	2.16	-
Avoidance maneuver	17	5.00	-
Emergency stopping bay[113]	17	1.57	-
Highway access/exit	-	3.14	$6.67 \cdot 10^{-3}$
Highway interchange	25	3.14	$2.50 \cdot 10^{-3}$
Road class 1 (RC1)	25	2.45	$2.13 \cdot 10^{-3}$
Road class 2 (RC2)	25	3.43	$3.57 \cdot 10^{-3}$
Road class 3 (RC3)	20	3.43	$7.14 \cdot 10^{-3}$

[113] Forschungsgesellschaft für Straßen- und Verkehrswesen: Guideline for rural roads (2012), p.91.

4.4.3 Fallback Requirements

The fallback requirements for a steering system for AD3+ of trucks are split into the required fallback steering torque at the pitman arm, the fallback steering energy and the fallback steering power. The results are exemplary for a 26-t truck and are developed similar to the redundancy requirements of the steering system for the 12-t truck in a previous publication of the author of this thesis[114].

Figure 4-9 shows the maximum fallback steering torque and angular velocity occurring at the pitman arm during the different maneuvers. The maneuver types can be classified into three groups according to their required maximum steering torque. Group A considers the maneuvers requiring less than 1,100 Nm of torque at the pitman arm. With those maneuvers, regular driving on RC1 and RC2 is possible including lane changes, interchanges as well as the driving into an emergency stopping bay. Group B with required steering torques between 1,100 Nm and 2,000 Nm contains the RC3 driving without turning maneuvers, the highway access/exit as well as an avoidance maneuver. The most advanced steering torque requirements between 2,000 Nm and 3,000 Nm arise for group C during turn-off maneuvers. The chamfers of the areas of the three different groups indicate the maximum occurring steering power of each group as also shown in Figure 4-10.

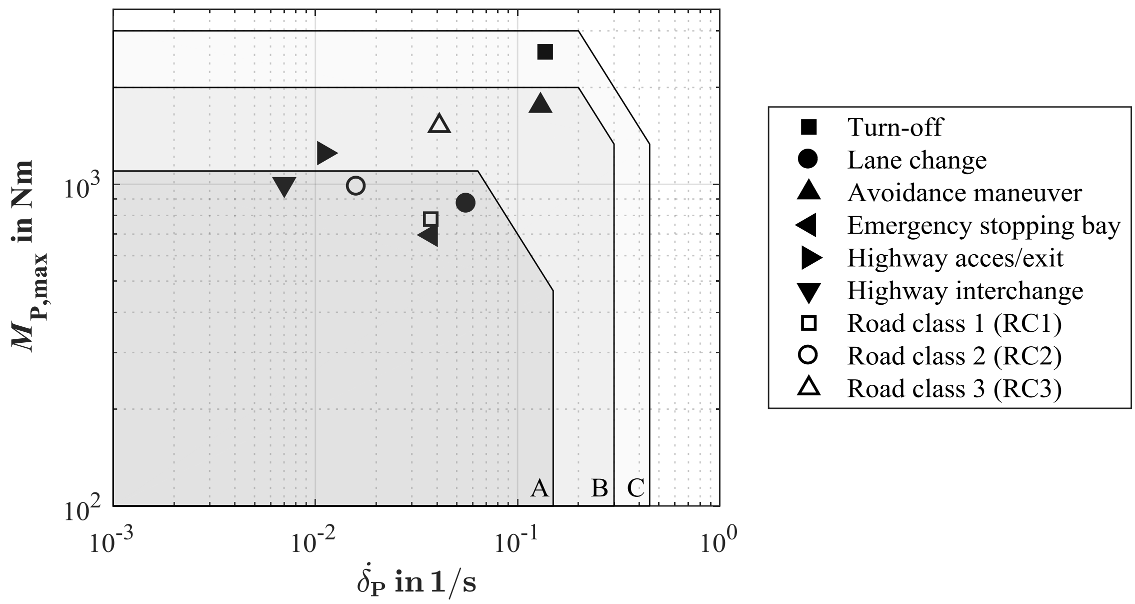


Figure 4-9: Maximum fallback steering torque

The energy required for steering during the different driving maneuvers (E_{P}) is calculated with equation (4-20). The calculated steering torque at the pitman arm (M_{P}) is integrated over the steering angle at the pitman arm (δ_{P}).

The maneuvers RC1, RC2 and RC3 are continuous maneuvers and the required energy depends on the driven distance. However, to get an indication for the required steering energy

[114] Herold, M. et al.: Power Requirements for RASS (2019), pp. 4–7.

during driving on these road classes, the maximum demanded steering energy on a driving distance of 1 km is used and illustrated with the required steering energy during the other maneuvers in Figure 4-10.

The classification into the three groups of driving maneuvers is also feasible for the steering energy. Group A again has the lowest requirements with a required steering energy of maximum 200 J for the drive of a single maneuver. Between 200 J and 500 J are required by the maneuvers of group B. Group C requires the most steering energy by far with values between 500 J and 1,300 J.

The steering power at the pitman arm of the truck occurring during the different driving maneuvers (P_P) is the derivation of the steering energy (E_P) as described in equation (4-21). Figure 4-10 shows the maximum steering power required during each maneuver type.

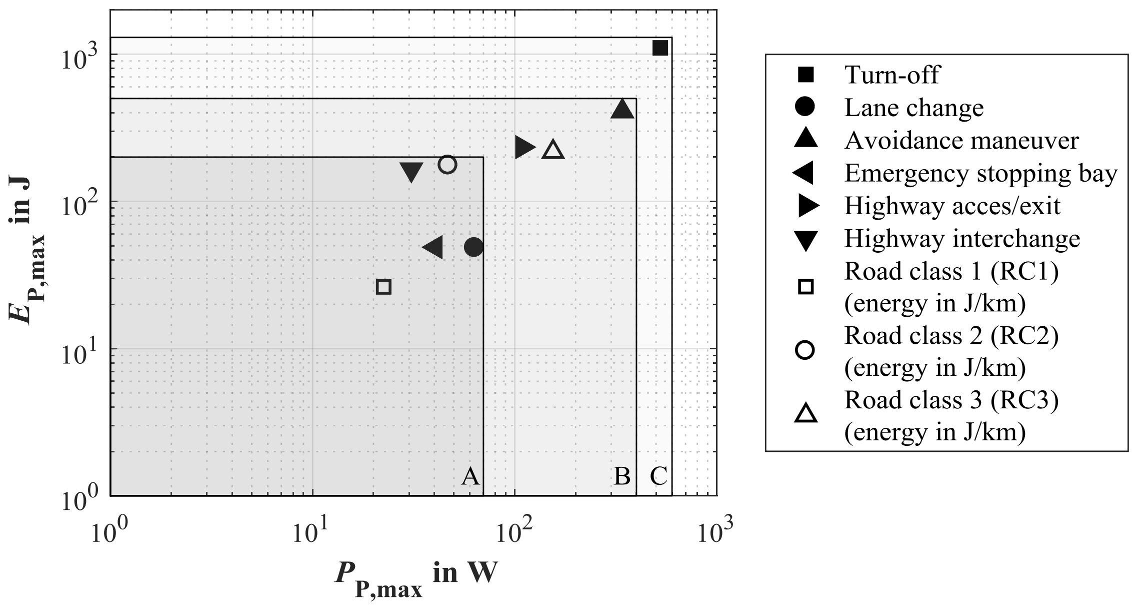


Figure 4-10: Maximum fallback steering energy and fallback steering power

The maneuver types are classified into three groups according to the maximum required steering power, similar to the previous classifications. Group A with the lowest power requirement of maximum 70 W contains the RC1 and RC2, highway interchanges, lane changes and the driving into emergency stopping bays. The highway exit, the avoidance maneuver and the RC3 driving form group B with a maximum required steering power between 70 W and 400 W. The maneuver turn-off requires between 400 W and 600 W of steering power and thus is classified to group C.

The simulation results for the described maneuvers considering the required steering torque, power and energy as well as the classification of these maneuvers into three groups according to their fallback requirements are listed in Table 4-7. The maneuvers are separated into the three groups without any overlapping.

Table 4-7: Groups of fallback requirements

Group	Maneuver type	$M_{\mathrm{P,max}}$	$P_{\mathrm{P,max}}$	$E_{\mathrm{P,max}}$
A	Lane change	< 1,100 Nm	< 70 W	< 200 J
	Emergency stopping bay			
	Highway interchange			
	Road class 1 (RC1)			
	Road class 2 (RC2)			
B	Avoidance maneuver	1,100 – 2,000 Nm	70 – 400 W	200 J – 500 J
	Highway access/exit			
	Road class 3 (RC3)			
C	Turn-off (rural roads)	2,000 – 3,000 Nm	400 - 600 W	500 – 1,300 J

4.4.4 Steering Redundancy Requirements

With the help of the determined fallback requirements in Table 4-7, steering redundancy requirements for the 26-t truck are developed for the three different road classes. This derivation is similar to that for a 12-t truck in a previous publication[115].

For each road class, an exemplary combination is set up, consisting of maneuvers out of the three different requirement groups in Table 4-7, which are necessary to reach the defined safe state. The exemplary cases are set up according to the worst-case situations for the occurrence of a steering system fault, which were found by analyzing the roads in the surrounding area of Darmstadt. The torque and power requirements are independent of the number of necessary maneuvers to reach the safe state. Of course, the required steering energy increases with an increasing number of necessary maneuvers and with an increasing necessary driving distance.

The steering redundancy requirements (RR) are split into the required steering redundancy torque (M_{RR}), the redundancy power (P_{RR}) and the redundancy energy (E_{RR}). These are listed in Table 4-8. For the RC1 with a permanent hard shoulder, a worst-case scenario of a fault is if the system fault occurs while the truck is on the third lane from the hard shoulder, and thus requires three lane changes to stop on the hard shoulder and reach the safe state. Because lane changes are not always possible immediately, 1,000 m of RC1 driving are considered as well for determining the redundancy requirements. In that case, a maximum pitman arm torque ($M_{\mathrm{P,max}}$) of 1,100 Nm, a maximum steering power ($P_{\mathrm{P,max}}$) of 70 W and a

[115] Herold, M. et al.: Power Requirements for RASS (2019), pp. 6–7.

maximum steering energy ($E_{\mathrm{P,max}}$) of 800 J are necessary. If an avoidance maneuver is necessary as well, the requirements for this road class increase to 2,000 Nm, 400 W and 1,300 J. The RC2 with emergency stopping bays is quite similar to RC1, but the difference is that a safe stop is not always immediately possible. A safe stop is only possible at an emergency stopping bay instead, which are only available at a distance of 1,000 m. Therefore, to reach a safe state of this road class in the worst case, 2,000 m of RC2 driving, two lane changes and the drive into an emergency stopping bay are necessary. Whereas the torque and power requirements are the same as for RC1, the required maximum steering energy ($E_{\mathrm{P,max}}$) increases to 1,000 J respectively to 1,500 J if an avoidance maneuver or another maneuver from group B is necessary. The RC3 differs from the other two, because there is no safe stop possible at the side of the road. In contrast, a turn-off maneuver is necessary to get to a road where a stop at the side of the road is possible or where the speed limit is below 70 km/h and thus a safe stop on the road is possible. Since a turn-off to such a road is not possible within a short distance in any case, a 5,000 m RC3 drive and a subsequent turn-off to a side road are considered as relevant to reach a safe state of this class. These maneuvers require the highest steering redundancy requirements of 3,000 Nm, 600 W and 3,800 J.

Table 4-8: Steering redundancy requirements (RR)

RC	**Exemplary maneuvers to reach safe state**	$\boldsymbol{M}_{\mathbf{RR}}$	$\boldsymbol{P}_{\mathbf{RR}}$	$\boldsymbol{E}_{\mathbf{RR}}$
1	4x group A (add. avoidance maneuver: 1x group B)	1,100 Nm (2,000 Nm)	70 W (400 W)	800 J (1.30 kJ)
2	5x group A (add. avoidance maneuver: 1x group B)	1,100 Nm (2,000 Nm)	70 W (400 W)	1.00 kJ (1.50 kJ)
3	5x group B 1x group C	3,000 Nm	600 W	3.80 kJ

4.4.5 Potential of Brake Steering as Fallback System

A fallback for the steering functionality for AD3+ is not only realizable by a fail-operational steering system, but also by the brake system, which is able to steer the vehicle by braking only the wheels on one side of the vehicle, so-called differential braking or brake steering. This function could be taken over by the electronic stability control (ESC), which is mandatory in modern trucks. By using this already available system, synergetic effects would be used and additional hardware superfluous.

However, the dynamic and the precision of brake steering are not comparable with those of a proper steering system.[116] The author of this thesis has also proven in a previous investigation that brake steering is not a sufficient fallback for a 12-t truck.[117]

[116] Michael Reichenbach: Interview with Alexander Gaedke (2017), p.24.

[117] Herold, M. et al.: Differential Braking for Steering Redundancy (2018).

The maximum steering torque that can be applied stationary by differential braking at the steering of the 26-t truck is calculated in this section, to examine if brake steering is able to fulfill the steering redundancy torque requirements determined in the preceding subchapter.

The steering torque that can be applied at the pitman arm by differential braking (M_{F_x}) is calculated by equation (4-22), whereby the differential brake force ($\Delta F_{x,\mathrm{fa}}$) is defined by equation (4-23). This differential force reaches its maximum if no braking force is applied at one front wheel and maximum braking force at the other front wheel. Hence, the maximum differential brake force ($\Delta F_{x,\mathrm{fa,max}}$) is defined in equation (4-24) by the maximum friction coefficient between the tire and the road surface (μ_{max}) and the front axle load ($F_{z,\mathrm{fa}}$). The stationary front axle load is used here for the calculation, because on the one hand additionally driving torque decreases the deceleration by differential braking and thus decreases the load shift to the front axle.[118] On the other hand, the wheel load on the inner front wheel, which is braked to generate steering torque due to the positive scrub radius, decreases because of the load shift to the outer wheels of the curve. It is assumed that the load shift to the front axle due to braking and the load shift to the outer wheels due to steering neutralize each other.

$$M_{\mathrm{F}_x} = \Delta F_{x,\mathrm{fa}} \cdot r_0 \cdot \cos\sigma \cdot \cos\tau \tag{4-22}$$

$$\Delta F_{x,\mathrm{fa}} = F_{x,\mathrm{fl}} - F_{x,\mathrm{fr}} \tag{4-23}$$

$$\Delta F_{x,\mathrm{fa,max}} = \mu_{\mathrm{max}} \frac{F_{z,\mathrm{fa}}}{2} \tag{4-24}$$

With the vehicle and suspension data of the 26-t truck[119] and assumed coefficients for the friction between a truck tire and a dry respectively a wet road surface, the maximum brake steering torque at the pitman arm for a dry road surface ($M_{\mathrm{F}_x,\mathrm{max,dry}}$) is 2,020 Nm and 1,443 Nm for wet road conditions ($M_{\mathrm{F}_x,\mathrm{max,wet}}$).

$$r_0 = 0.07\ \mathrm{m}, \sigma = 0.14, \tau = 0.06,$$
$$F_{z,\mathrm{fa}} = 83.4\ \mathrm{kN}, \mu_{\mathrm{max,dry}} = 0.7, \mu_{\mathrm{max,wet}} = 0.5,$$

$$M_{\mathrm{F}_x,\mathrm{max,dry}} = 2{,}020\ \mathrm{Nm}$$

$$M_{\mathrm{F}_x,\mathrm{max,wet}} = 1{,}443\ \mathrm{Nm}$$

Thereby, even in stationary state, the brake steering is not able to generate sufficient steering torque to fulfill the fallback requirements in Table 4-7. The brake steering is able to generate enough torque to meet the fallback steering torque requirements for group A and roughly for group B in dry conditions, but misses the requirements for group C significantly. In driving

[118] Herold, M. et al.: Differential Braking for Steering Redundancy (2018), pp. 10–11.

[119] Vehicle data see Table A-3 in the Appendix.

situations with lower friction coefficients such as a wet road, this gap increases even more. The dynamic drawbacks of brake steering are not even considered in this examination.

These results, together with the simulative study[120], permit the conclusion that brake steering is not a sufficient fallback system for the steering functionality for the AD3+ of trucks if, as intended in this thesis, the application is not limited to RC1 or RC2, but should be feasible for all road classes. In addition, the influences of wheel load oscillations and the highly limited maneuverability during permanent braking argue against the use of brake steering as a fallback level.

4.5 Conclusion of Requirement Analysis

Starting with the frame requirements for a truck steering system suitable for AD3+, the requirements for a correct operation of the steering system, but also the redundancy requirements for an incorrect operation due to a fault in the steering system are determined by the analysis in this subchapter. Whereas the frame requirements are already described in Table 4-1, the operational and the redundancy requirements are listed in Table 4-9. A complete requirement list is attached in the appendix in Table A-4.

Because the required fallback steering torque exceeds the maximum brake steering torque, a redundancy inside the steering system is necessary. The required steering system is called redundant active steering system (RASS). A fail-degraded redundancy is sufficient for the use case of AD3+ of trucks, since the redundancy requirements are much lower than the operational requirements for the correct operation of the steering system.

Table 4-9: Operational and redundancy requirements

	Requirement name	Values, data	Description
Operational requirements	Maximum output torque	8,500 Nm	Maximum torque at pitman arm during correct operation
	Maximum angular velocity	50 °/s	Maximum angular velocity at pitman arm during correct operation
	Maximum output power	3,500 W	Maximum power of overall steering system at pitman arm during correct operation
Redundancy requirements	Fallback torque	3,000 Nm	Maximum output torque at pitman arm in case of a partial failure of the steering system
	Fallback power	600 W	Maximum output power at pitman arm in case of a partial failure of the steering system
	Fallback energy	3,800 J	Available steering energy at pitman arm in case of a failure of the power supply

[120] Herold, M. et al.: Differential Braking for Steering Redundancy (2018).

There are three possible options for a RASS, which are a redundant electric power steering (EPS^2), a redundant hydraulic power steering (HPS^2) or a redundant hybrid power steering system (HPS + EPS). The EPS^2 is omitted, because the required maximum output torque of the steering system is not realizable solely by an electric motor with a power supply of 24 V and without a very high gear ratio, which would exceed the maximum dimensions of the steering system. Furthermore, the maximum output power would require a maximum current of 146 A, which would overload the vehicle's power network, which is designed for a permanent maximum current of 110 A.[121] The HPS^2 is excluded due to its low efficiency caused by the doubled hydraulic losses. Thus, a redundant hybrid steering system is the only suitable solution for a RASS for AD3+ of trucks. The used HPS subsystem delivers enough steering torque to fulfill the operational requirements, whereas the EPS only has to fulfill the degraded redundancy requirements. Furthermore, the EPS enables to increase the efficiency of the hybrid steering system compared to a standard HPS.

[121] Maximum current of generators for a 24 V on-board network, which was found during a search in the internet.

5 System Architecture

An appropriate system architecture is the basis for the development of a system that fulfills all previously determined requirements. Different possible functional structures of a hybrid RASS are analyzed in a first step. The derivation and implementation of a suitable functional safety concept is a crucial objective when developing systems for AD3+[122] and has a great impact on the system architecture as well. Different system architectures, which meet the described requirements including the developed functional safety requirements, are described and discussed. The superior system architecture is determined as the final system architecture at the end of this chapter.

5.1 Functional Analysis

According to the requirement analysis in the preceding chapter, a hybrid redundant active steering system (RASS) is required for AD3+ of trucks, since it is the only possible solution that fulfills the frame requirements as well as the operational and the redundancy requirements. The crucial subsystems of such a system are an electronic power steering (EPS) subsystem and an active hydraulic power steering (AHPS) subsystem. These two subsystems provide the necessary functions based on the functional structures of the current SoA active steering systems described in subchapter 2.4. The possible structural combinations of these functions are analyzed and discussed. The detailed functional structures and their specific features, advantages and disadvantages are described hereafter. There will be a comparison of all structures in section 5.1.5. To build up the functional structures, symbols according to VDI 2222 are used, as shown in Figure 5-1.

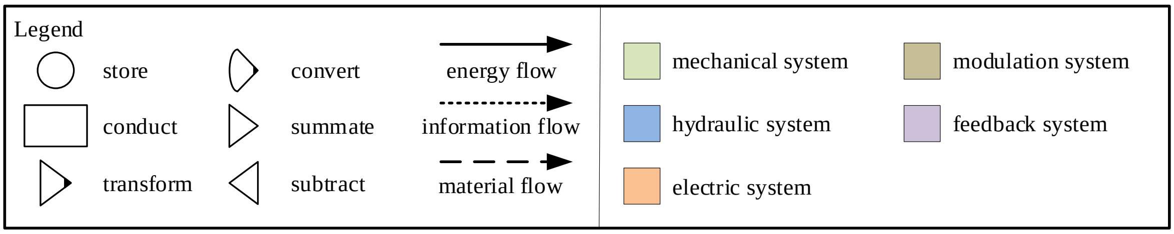


Figure 5-1: Legend for functional structures[123]

There is one symbol each for storing, conduction, transformation, conversion, summation and subtraction and different arrows for energy, information and material flow. The directions of the arrows represent the direction of the flows, i.e. the starting point is the steering wheel and the ending point is the steering linkage, which is connected to the wheels. The

[122] Stolte, T. et al.: Towards Automated Driving (2015), p.673.

[123] Verein Deutscher Ingenieure: VDI 2222: Method for Developing Technical Systems (1993), p.16.

physical values linked to the arrows can be an energy (E), a torque (M) with a steering angle (δ), a rotation angle (φ) or a volume flow (Q) with a pressure (p). The indexes of the physical values are listed in the index directory.

Furthermore, five different colors are used to illustrate, which function belongs to which subsystem. All functions required for a mechanical steering system are green, those that are added for a hydraulic power steering (HPS) are blue and those for an electric power steering (EPS) are orange. The additional functions for the modulation system are brown and the functions necessary for a feedback system are purple.

5.1.1 RASS A

The specific feature of the first function structure RASS A, shown in Figure 5-2, is the position at which the electric motor (EM) is connected to the steering shaft. It is connected to the input shaft of the steering gear (StG) by an additional gear before the torsion bar (TB).

Thereby, the electric motor is able to control the hydraulic steering assistance by twisting the torsion bar as well as the modulation actuator (MA) by twisting the valve sleeve via an additional gear. The fact that the hydraulics are controllable via two different ways independent of the driver, is the crucial edge of this specific functional structure. Thereby, the system is still able to generate high steering power by the hydraulic system independent from the driver if either the electric motor or the modulation actuator fails.

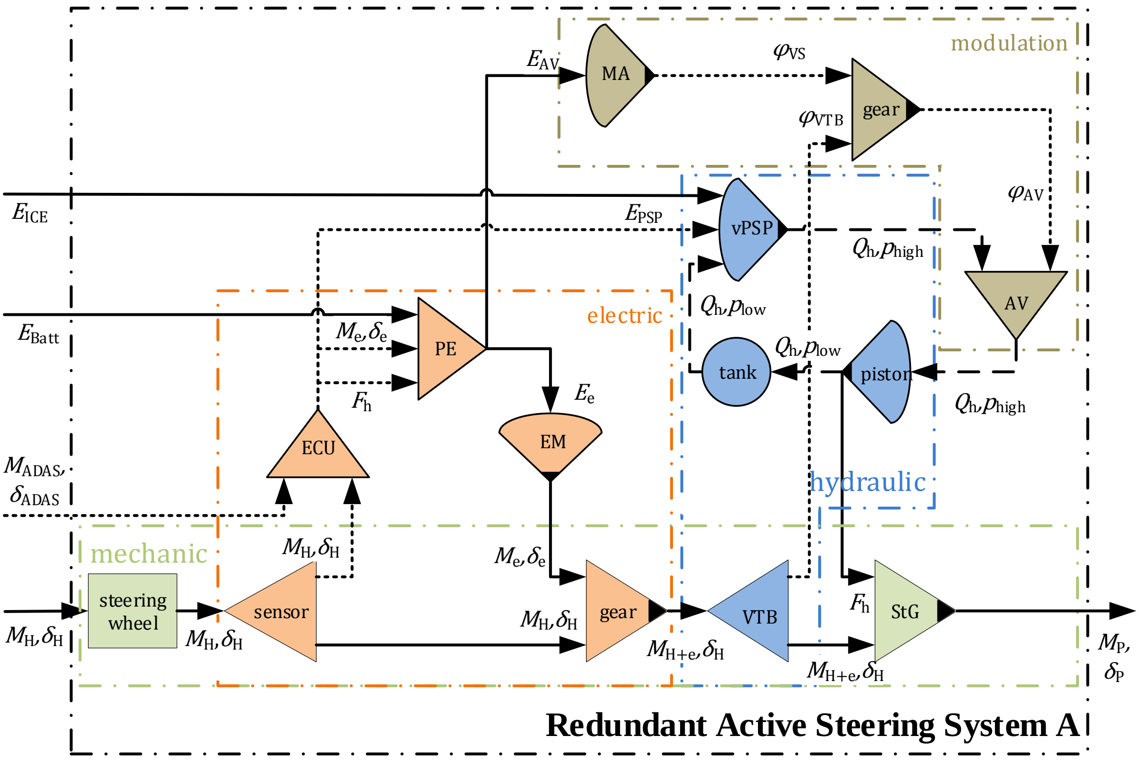


Figure 5-2: Functional structure RASS A with steering gear (StG), variable power steering pump (vPSP), electric control unit (ECU), power electronics (PE), electric motor (EM), modulation actuator (MA), valve torsion bar (VTB), and active valve (AV)

In addition to these two automated ways to control the hydraulic volume flow, the driver is able to control it by twisting the torsion bar, too. Thus, the driver gets the chance to oversteer the automated system. Since the torsion bar has to be designed for the torque of the electric motor, which is higher than the driver's maximum torque, it has to be much stiffer than a torsion bar for a standard HPS. If the stiffness of the torsion bar were designed according to the maximum input torque of the driver, the more powerful EPS would always twist the torsion bar fully to the mechanical stop when it applies torque. This would cause backlash in the steering system.

The stiffer torsion bar requires significantly more torque of the driver to steer compared to a standard HPS, which is why the driver functions only as a fallback in case of a fault of the electric system (EM and MA). But to comply the ECE R79 and in case of a fault, the steering force at the steering wheel is not allowed to exceed 450 N. This has to be considered during the design of the torsion bar.

The option to control the hydraulics by the electric motor and the modulation actuator is a big benefit, but the two ways have to be coordinated which is a relatively complex control task. The control algorithm is implemented in the electric control unit (ECU), which controls the distribution of the electric energy by the power electronics (PE) and the adjustment of the energy demand of the power steering pump (PSP) by an adjustment mechanism (AM). Another characteristic of RASS A is that there is no need for a special steering gear. The special control module with the torsion bar, the modulation actuator and gear and the valve are adapted to a standard recirculating ball (RCB) steering gear.

5.1.2 RASS B

The second functional structure RASS B distinguishes itself from RASS A by the position of the gear, which connects the electric motor to the steering shaft. As seen in Figure 5-3, the gear is fitted after the torsion bar (TB) at the steering gear (StG), which causes different characteristics compared to RASS A.

Because the application point of the electric motor is behind the torsion bar, the electric motor (EM) is not able to twist it and thus to control the hydraulic steering assistance. Therefore, the modulation is the only driver-independent way to control the hydraulics. If it fails, the system can still steer by the electric motor, but the maximum torque is limited and considerably lower compared to the maximum torque of the system with hydraulic assistance. However, compared to RASS A, a less complex control strategy is necessary, since the modulation is the only way to automatically control the hydraulics.

Because the electric motor applies torque only behind the torsion bar, the design of the torsion bar is independent of the maximum torque of the electric motor. Hence, a standard torsion bar from a standard HPS is used, which requires significantly less torque by the driver to control the hydraulics and in case of a fault to oversteer the system, compared to RASS A. The position of the electric motor at the steering gear is not fixed in this functional structure. A mounting at the top or at the bottom of the steering gear's input shaft is possible as well as at its output shaft. Like the first functional structure, RASS B neither requires a special steering gear.

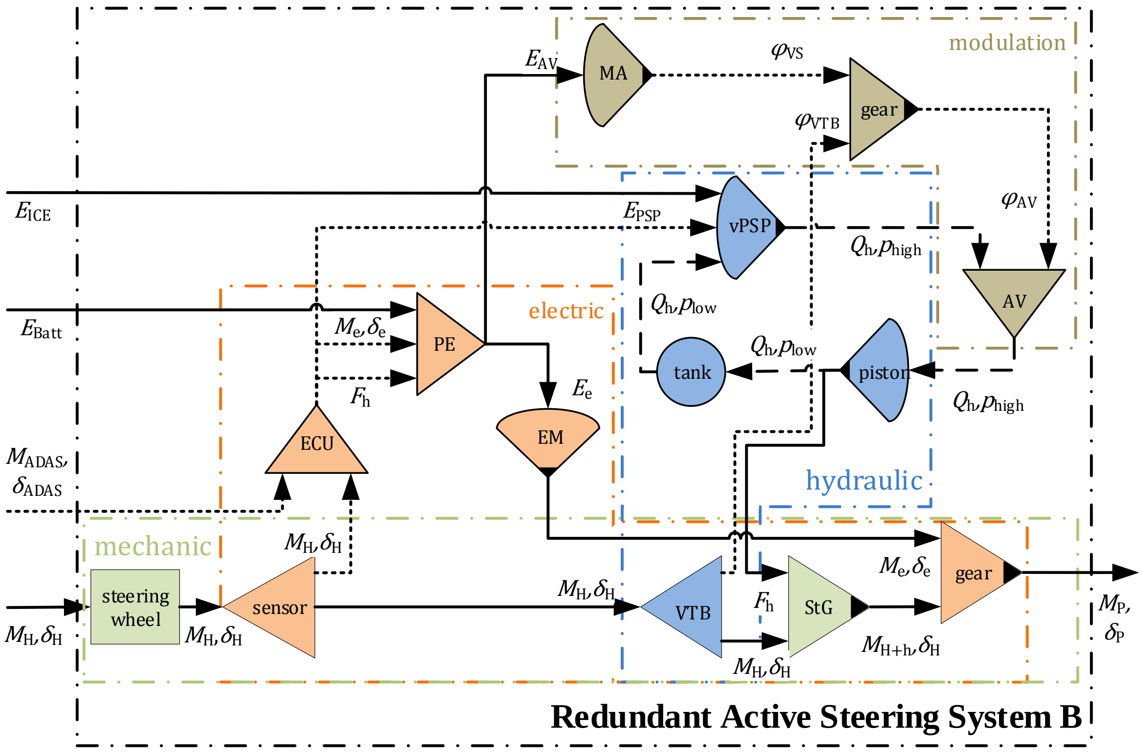


Figure 5-3: Functional structure RASS B with steering gear (StG), variable power steering pump (vPSP), electric control unit (ECU), power electronics (PE), electric motor (EM), modulation actuator (MA), valve torsion bar (VTB) and active valve (AV)

5.1.3 RASS C

In Figure 5-4, the third function structure RASS C is displayed. It differs from the first two functional structures because there is no torsion bar between the steering wheel and the steering gear.

Due to the missing torsion bar, the hydraulics are only controllable by the modulation actuator (MA). Even the driver is not able to control the hydraulics, which is a big disadvantage if the modulation fails. In this case, the electric motor (EM) is the fallback level with its limited maximum torque. If this fails as well, the driver is the last fallback level using the mechanical passage from the steering wheel to the wheels. Since he would not get any steering assistance, the steering performance would be restricted to the maximum manual torque of the driver.

Another big drawback of this specific functional structure is that the driver is hardly able to oversteer the system, because of his significantly lower power compared to the power of the steering assistant systems.

The biggest advantage of the RASS C is the freedom of design of the steering valve. As there is no torsion bar in the RASS C, the valve doesn't have to be compatible to the torsion bar (e.g. rotary slide valve), but can be any valve design (e.g. proportional valve). Therefore, the design of the entire steering system and the control of the hydraulics are less complex. Also, a standard steering gear is used for the RASS C.

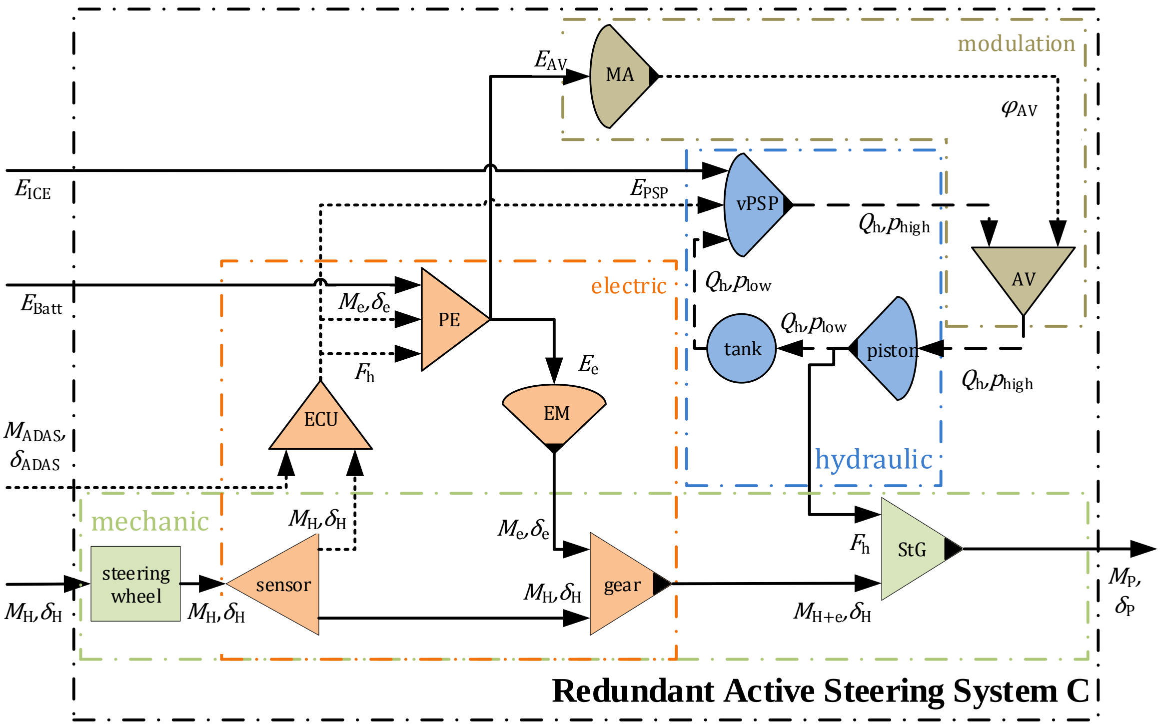


Figure 5-4: Functional structure RASS C with steering gear (StG), variable power steering pump (vPSP), electric control unit (ECU), power electronics (PE), electric motor (EM), modulation actuator (MA) and active valve (AV)

5.1.4 RASS D

The distinctive feature of last functional structure RASS D, shown in Figure 5-5, is the missing mechanical passage between the steering wheel and the steering gear. Thereby, RASS D is a steer-by-wire system (SbW).

The control of the hydraulics is similar as it is in RASS C and has the same pros and cons. The difference is the missing mechanical fallback level. In case of a fault of the modulation, the electric motor (EM) is the last fallback level, since the driver is not able to transfer torque to the steering gear due to the missing mechanical passage.

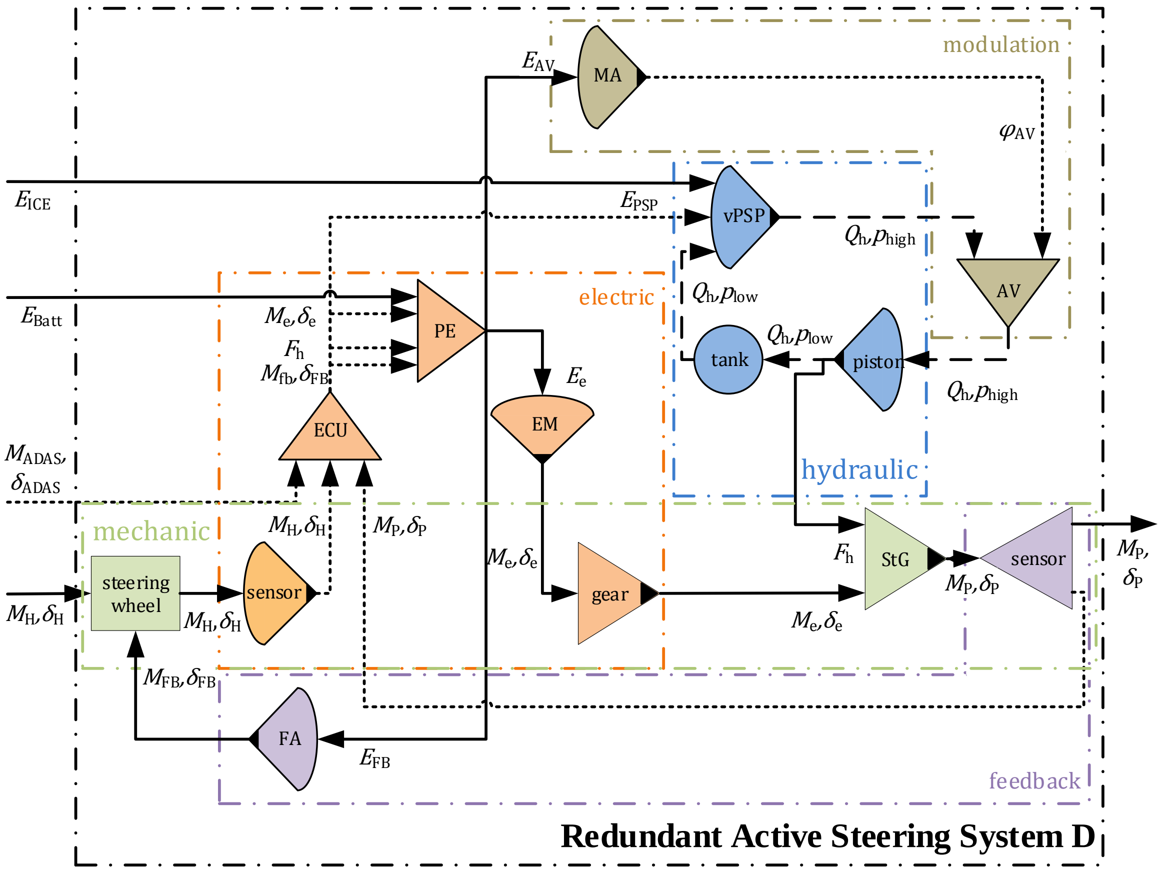


Figure 5-5: Functional structure RASS D with steering gear (StG), variable power steering pump (vPSP), electric control unit (ECU), power electronics (PE), electric motor (EM), modulation actuator (MA), active valve (AV) and feedback actuator (FA)

The lack of the mechanical linkage between the steering wheel and the wheels cancels the dependency between the steering angle and the steering torque. It causes a design freedom of the steering behavior and the steering feel. A feedback actuator (FA) generates the feedback to the driver artificially. It is controlled by the ECU using sensor information about the torque and the angle at the output shaft of the steering gear. However, these additional components for feedback generation make the whole system more complex. Another benefit of the missing steering shaft is a raise of safety in case of a frontal crash. In addition, the same standard steering gear can be used for vehicles with left- and right-hand drive.

5.1.5 Comparison of Different Functional Structures

The detailed functional structures of a RASS are described in the previous sections. Table 5-1 gives an overview of the special features as well as of the advantages and the disadvantages of the individual functional structures.

As the driver is not able to control the hydraulic power steering (HPS) system for RASS C and RASS D, these two functional structures are not further considered in this thesis. The HPS is the most powerful steering actuator of the hybrid RASS, thus in case of an erroneous steering torque generation by the HPS, the driver is only able to counteract if he is able to override the active HPS control. Because RASS C and RASS D do not provide this ability to the driver, they miss the override ability requirement of the Vienna's convention on road traffic[124] and of the ECE R79[125].

Table 5-1: Comparison of function structures (FS)

FS	RASS A	RASS B	RASS C	RASS D
Special features	EM connected to steering shaft before steering valve	EM connected to steering shaft after steering valve	No TB, but mechanical passage from SW to wheels	No TB & no mechanical passage from SW to wheels (SbW)
Advantages	- double active HPS control - HPS controllable by the driver - oversteerable by the driver - standard RCB with stiffer TB	- HPS controllable by the driver - standard TB: easy to oversteer by the driver - easy control: just one active HPS control - standard RCB	- no TB: free design of valve - easy control of HPS - driver last fallback-level - standard RCB	- no TB: free design of valve - easy control of HPS - free design of steering behavior (T & δ) - free design of steering feel - no steering shaft: more safety & design freedom - standard RCB
Disadvantages	- stiff TB requires high torques by driver to control HPS without EPS - complex coordination of double active HPS control	- just one active HPS control	- just one active HPS control - HPS not controllable by the driver - hardly overrideable by the driver	- just one active HPS control - no mechanical fallback-level - additional feedback generation

[124] United Nations: Convention on Road Traffic (1968), p.11.

[125] United Nations: ECE R79 r4 (2018), p.14.

5.2 Safety Analysis

The safety analysis is a central step within the development of safety-relevant systems. The methodology of the safety analysis according to the ISO 26262 is illustrated in Figure 5-6. The methods used in the single steps of the analysis are listed on the left side, whereas the several results are illustrated symbolically on the right.

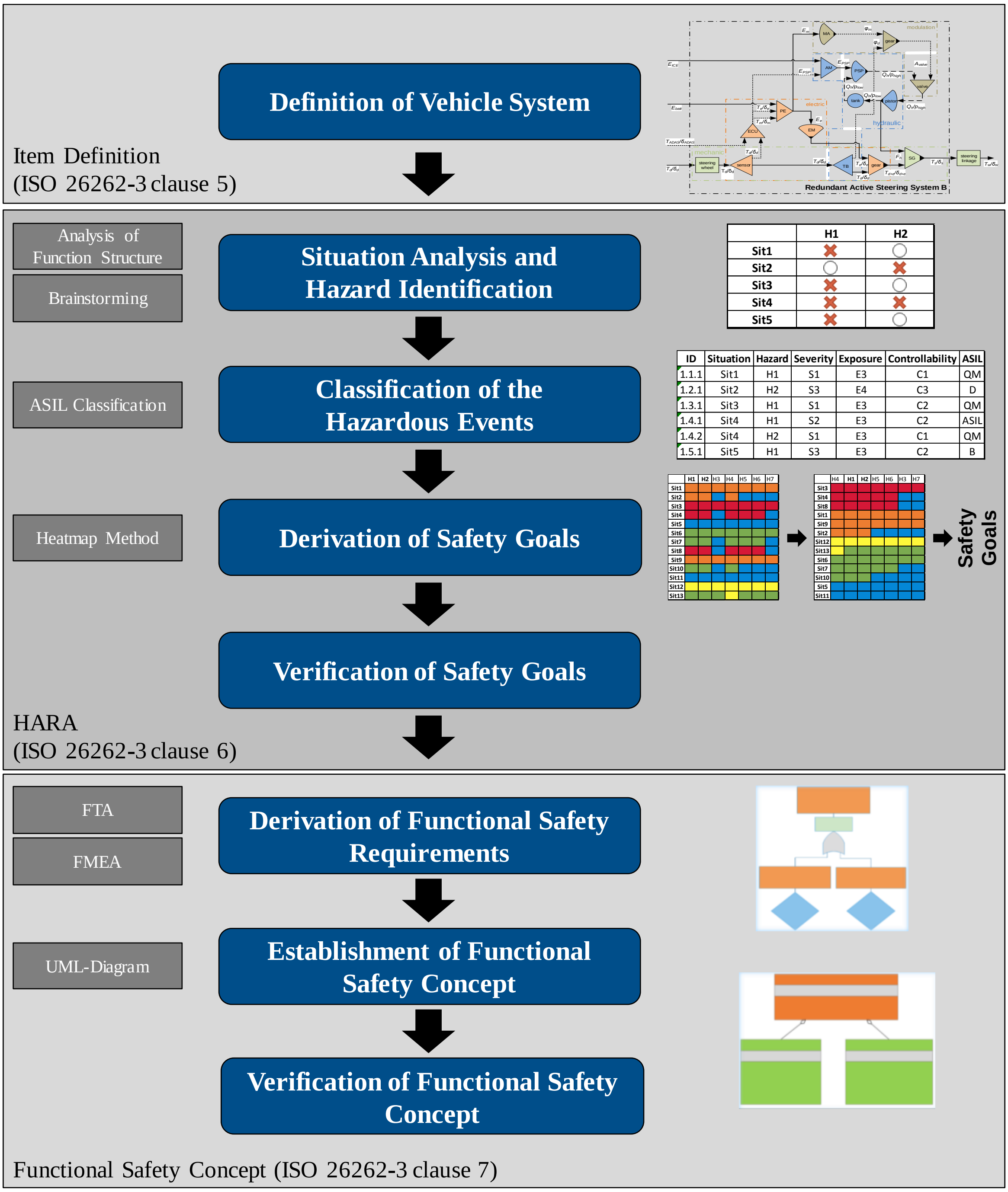


Figure 5-6: Methodology of safety analysis

The methodology is divided into three main steps. The first step is the item definition. Here, the steering system, which is in focus of the project, is defined in detail. The next main step, the hazard analysis and risk assessment (HARA), starts with the analysis of the possible driving situations, which are relevant for the steering system. Thereby, the possible hazards are identified. Those are classified in the next step by assessing the ASIL of the single hazardous situations. After the derivation of the safety goals, the HARA ends with their verification. The last main step of the safety analysis is the development of the functional safety concept. Therefore, the functional safety requirements (FSR) are derived using the methods of fault tree analysis (FTA) and failure mode and effects analysis (FMEA). Those functional safety requirements are allocated to a functional safety concept by using unified modeling language (UML) diagrams. With the result of a functional safety concept and its verification, the safety analysis ends and the development continues on system, hardware and software level.

5.2.1 Item Definition

For an appropriate functional safety analysis, it is important to define the item of the analysis properly. Here, the item is the RASS for AD3+ of trucks, with an increased energy efficiency compared to standard HPS. The two functional structures RASS A and RASS B defined above and selected for the further approach are both considered for the safety analysis.

The intended functionality of the RASS includes standard power steering and AD2- as well, but since the AD3+ is the innovative functionality of the RASS, the safety analysis focuses on that. However, for a complete valid safety analysis, the other functionalities have to be considered as well. For AD3+, the RASS generates steering torque independently of the driver and provides a system internal fallback level for the case of a fault, which provides fail-degraded active steering functionality until the driver takes over control or the vehicle is transferred into a safe state.

The driver and the automated lateral guidance system deliver the signal inputs to the steering system, which in turn gives feedback about the driving situation to the driver. The steering system's power inputs are the vehicle's power network for the electric power supply and power steering pump (PSP) driven by the internal combustion engine (ICE) for the hydraulic power supply.

Table 5-2 lists the intended system operation states. Besides the different levels of automation (LoA), the RASS has different power steering states (PS), in which the steering system generates the steering torque solely electric (EPS state), hydraulic (HPS state) or combined (hybrid state). Because the required steering torques decrease with increasing vehicle's velocity, at high velocities the EPS state is active to increase the efficiency. Only if the velocity is below a threshold (v_{Switch}), the hydraulic is active. Depending on the LoA and the fault state (FS), the system has different fallback states. During manual driving (LoA 0), the steering system has power steering and comfort functionality (e.g. active return to straight) and in case of a fault of one of the power steering systems, the other power steering system serves as a fallback. In AD3+ state, the steering system generates steering torque independently of the driver and steers automatically. The driver might take off his hands from the steering wheel and focus on other tasks. In case of a fault of one active steering system, the other active system serves as a fallback and a take-over request (ToR) is sent to the driver, who has a specific period to take over the control. Otherwise, the system transfers the vehicle into a defined safe state. The driver is able to override the active steering system at any time independently of the LoA. If the driver intervenes during automated driving, the automated steering function is deactivated and just the power steering function remains. An unmotivated steering action should be prevented safely at any time.

Table 5-2: Intended system operation states, including power steering state (PS), SAE level of automation (LoA) and fault state (FS)

PS	LoA	FS	Fallback/Safe state
EPS	0	No fault	-
EPS	0	Fault	Driver & HPS by TB
EPS	3+	No fault	-
EPS	3+	Fault	HPS by modulator & take-over request (ToR) → transition to driver/safe state
HPS	0	No fault	-
HPS	0	Fault	Driver & EPS
HPS	3+	No fault	-
HPS	3+	Fault	EPS & ToR → transition to driver/safe state
Hybrid	0	No fault	-
Hybrid	0	EPS Fault	Driver & HPS by TB
Hybrid	0	HPS Fault	Driver & EPS
Hybrid	3+	No fault	-
Hybrid	3+	EPS Fault	HPS by modulator & ToR → transition to driver/safe state
Hybrid	3+	HPS Fault	EPS & ToR → transition to driver/safe state
All Modes	All Levels	No fault or Fault	Override able by the driver

5.2.2 Derivation of Safety Goals

The second main part of the concept phase of the ISO 26262 is the hazard analysis and risk assessment (HARA), which results in different safety goals for the item of the analysis. The HARA is conducted according to the ISO 26262 and contains three main parts. First, the hazards and the situations in which those hazards can occur have to be determined. The following step is to classify the automotive safety integrity level (ASIL) of the determined hazardous situations. The final step of the HARA is the derivation and verification of the safety goals for the different hazards, which a fault of the steering system can cause.[126]

Table 5-3 lists the steering system states and the driving situations, which are considered in the HARA. Each driving situation is analyzed separately for each combination of velocity and LoA.

Table 5-3: Steering system states and driving situations considered during HARA

Steering system state		**Description**
Velocity	$v_{\mathrm{V}} \leq v_{\mathrm{Switch}}$	Vehicle's velocity is below the velocity threshold. The steering system is in hybrid power steering (hybrid) state.
	$v_{\mathrm{V}} > v_{\mathrm{Switch}}$	Vehicle's velocity is above the velocity threshold. The steering system is in electric power steering state.
Level of automation	0	Manual driving
	AD3+	Highly automated driving
Driving situation		**Description**
Straight	Immediate stop possible	In case of a fault, an immediate stop without any steering action is possible.
	Immediate pullout possible	In case of a fault, an immediate pullout to the shoulder without any additional steering action is possible.
	Delayed pullout possible	In case of a fault, a pullout to the shoulder after a short drive with additional steering action is possible.
	Turn-off necessary	In case of a fault, a turn off the current road is necessary to reach a safe state.
Corner	During	Fault occurs during cornering.
	Before	Fault occurs less than 1s before cornering.
Lane change	During	Fault occurs during lane changing.
	Before	Fault occurs less than 1s before lane changing.
Avoidance maneuver	During	Fault occurs during evasive maneuvering.
	Before	Fault occurs less than 1s before evasive maneuvering.
Maneuvering	-	Maneuvering, e.g. on a yard with $v_{\mathrm{V}} < 10\ \mathrm{km/h}$.
Transition to driver	Hand-over	Automated driving system intends a hand-over to the driver and gives a ToR. After a defined Take-over Time (ToT) the driver takes over the control.
	Driver intervention	Automated driving system does not intend a hand-over to the driver. The driver takes over the control by intervening the automated driving system.

[126] International Organization for Standardization: ISO 26262-3 (2018), pp. 5–12.

The driving situations are distinguished into four different kinds of straights, depending on which maneuver is necessary to get to a safe stop. The next three driving situations, which are corner, lane change and avoidance maneuver, are distinguished regarding whether the vehicle is already in this situation or close to it. A driving situation that is only executed with low velocities is the maneuvering, e.g. on yards. The hand-over of the steering task from the automated system to the driver as well as the intervention of the driver into the automated driving maneuver are two driving situations, which have to be considered, too.

Beside the driving situations, the possible hazards caused by the RASS have to be analyzed for the HARA. Possible hazards are faults of the components of the hybrid steering system. System faults can be errors or failures, which differs in the way that in case of an error, the system provides an erroneous function and in case of a failure it provides no function. The relevant hazards for the HARA are listed in Table 5-4.

Table 5-4: List of potential hazards

Function	Hazard	Abbreviation
Energy supply	Failure of battery	Batt F
	Failure of engine	ICE F
Energy adjusting	Error of power electronics of RASS	PE E
	Failure of power electronics of RASS	PE F
	Error of adjustment mechanism for volume flow of vPSP	vPSP E
	Failure of adjustment mechanism for volume flow of vPSP	vPSP F
Energy converter	Error of electric motor of EPS in RASS A	EM E (A)
	Error of electric motor of EPS in RASS B	EM E (B)
	Failure of electric motor of EPS in RASS A	EM F (A)
	Failure of electric motor of EPS in RASS B	EM F (B)
	Error of modulator actuator of modulator in RASS A	MA E (A)
	Error of modulator actuator of modulator in RASS B	MA E (B)
	Failure of modulator actuator of modulator in RASS A	MA F (A)
	Failure of modulator actuator of modulator in RASS B	MA F (B)
Sensor	Error of steering wheel sensor	Sensor E
	Failure of steering wheel sensor	Sensor F
Control	Error of electronic control unit of RASS	ECU E
	Failure of electronic control unit of RASS	ECU F

The derivation process of the safety goals (SG) from the relevant driving situations and the potential hazards is not a central issue of this thesis and thus not described in detail here. However, a suitable approach following the ISO 26262 is presented by Gillen.[127] The hazardous events are identified from the combination of the driving situations and the potential hazards and assessed with an ASIL according to ISO 26262. The safety goals of the RASS are derived from the identified hazardous events and their ASIL. To keep the amounts of safety goals manageable, the hazardous events are classified into groups with a technical relationship and similar ASIL according to Table 5-5. One safety goal is defined for each of

[127] Gillen, C.: Dissertation, Development of efficient safety concepts for steering systems (2015).

these groups. Thereby, the whole steering system is covered with only seven safety goals, which are also listed in Table 5-5.

An error of the steering wheel sensor or of the ECU has to be prevented, in order to prevent unintended steering action at any time. The prevention of an error of the EPS or an error of the active valve have the same target. An erroneous power supply by the power electronic (PE) or the variable power steering pump (vPSP) would cause the generation of an erroneous amount of torque. This has to be prevented as well.

Since the driver is not available as a fallback during AD3+, it is essential to guarantee that the active steering function of the RASS does not fail at any time during AD3+. To ensure this, a simultaneous failure of components within this group that would cause a failure of the EPS and the AHPS has to be avoided. To detect the driver's intention always safely, a failure of the steering wheel sensor has to be prevented. For the safe power steering function, the internal combustion engine (ICS) and the vPSP should not fail.

Table 5-5: List of safety goals for RASS

ID	Group	Hazard	Safety goal (SG)
SG 1	Sensor & ECU error	Sensor E	An erroneous sensing of the steering wheel input or an erroneous control signal of the electronic control unit should be prevented safely.
		ECU E	
SG 2	EPS error	EM E (A)	An erroneous torque output of the electric power steering should be prevented safely.
		EM E (B)	
SG 3	Active valve error	MA E (A)	An erroneous opening of the active hydraulic valve by the modulator should be prevented safely.
SG 4	Power supply error	PE E	An erroneous power supply should be prevented safely.
		vPSP E	
SG 5	Active steering failure	Batt F	A failure of the driver-independent steering torque generation should be prevented safely.
		PE F	
		EM F (A)	
		EM F (B)	
		MA F (A)	
		MA F (B)	
		ECU F	
SG 6	Sensor failure	Sensor F	A failure of the sensing of the steering wheel input should be prevented safely.
SG 7	Power steering failure	ICE F	A failure of the power steering system should be prevented safely.
		vPSP F	

The defined safety goals are connected with their automotive safety integrity level (ASIL) in Table 5-6. There is an ASIL for each safety goal and each system state. Thereby, each safety goal has one ASIL depending on the power steering state and the LoA of the steering system.

The derived safety goals are verified according to ISO 26262-8 to ensure consistency regarding the item definition and suitability with the identified hazardous events and the determined ASIL.[128]

The hazardous events are derived directly from the item definition and classified into groups with a technical relationship. Thereby and by assigning each of these groups directly to the highest ASIL occurring in it, the safety goals are verified.

Table 5-6: ASIL of different safety goals for different LoA and system states

Safety groups			**Group 1**	**Group 2**	**Group 3**	**Group 4**	**Group 5**	**Group 6**	**Group 7**
			Sensor & ECU error	EPS error	Active valve error	Power supply error	Active steering failure	Sensor failure	Power steering failure
Safety goal			SG 1	SG 2	SG 3	SG 4	SG 5	SG 6	SG 7
ASIL	**MD**	$v_V \leq v_{Switch}$	C	B	C	A	QM	-	A
		$v_V > v_{Switch}$	D	C	-	C	QM	A	A
	AD3+	$v_V \leq v_{Switch}$	D	D	D	C	D	B	-
		$v_V > v_{Switch}$	D	D	-	D	D	B	-

5.2.3 Safety Concept

According to the ISO 26262, the objective of the functional safety concept is to derive the functional safety requirements from the safety goals, and to allocate them to the preliminary architectural elements of the item or to external measures in order to ensure the required functional safety. The functional safety concept includes the following parts:[128]

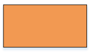
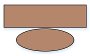
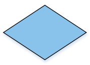
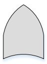
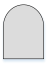
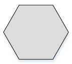
- fault detection and failure mitigation
- transition to safe sate
- fault tolerance mechanism, by which a fault does not lead directly to the violation of the safety goal(s) and which maintains the item in a safe state (with/without degradation)
- fault detection and driver warning in order to reduce the risk exposure time to an acceptable interval
- arbitration logic to select the most appropriate control request from multiple requests generated simultaneously by different functions

The functional safety requirements are derived based on the previously defined safety goals. Therefore, starting with the assumption of not reaching a specific safety goal, it is analyzed which error/failure or chain of errors/failures leads to that miss. The ISO 26262 recommends

[128] International Organization for Standardization: ISO 26262-3 (2018), pp. 12–15.

the deductive method of a fault tree analysis (FTA), which uses the symbols in Table 5-7 to analyze the causes of potential faults.

Table 5-7: Symbols of an FTA[129]

Symbol	Name	Description
	Event description	Description of the event, which indicates a failure/failure state, the state of system/component, a condition or an action
	Conditioning event with description	Event, which expresses a condition; Usually as input of an inhibit gate
	Undeveloped event	Event, whose cause is not further analyzed
	OR gate	Logic OR function of the input events
	AND gate	Logic AND function of the input events
	Inhibit gate	Gate with additional condition, which is connected with an AND function

The analysis is based on the top event at the top of the fault tree, which describes a missed safety goal. Therefore, there is one fault tree for each safety goal. Using logical linkages, the top event is connected with possible causes for the missed safety goal. Thus, especially for redundant systems, the FTA is a very useful method for the safety analysis.[130]

The fault path is explained exemplarily using the FTA of SG 5 shown in Figure 5-7. The superior event of each fault tree is on steering system level and represents the corresponding negated safety goal. The lower rectangles describe errors and failures on functional level. The diamonds at the bottom of each fault tree contain the errors or failures of the single components. Underneath the top event is the determined ASIL, depending on the steering system state.

The fifth safety goal is “A failure of the driver-independent steering torque generation should be prevented safely”. Hence, the undesired top event of the fault tree in Figure 5-7 is the “Failure of active steering”.

If there is no active steering during AD3+, the safety goal 5 with ASIL D is missed. On component level, the reasons for that fault are a failure of the electric power supply, a failure

[129] Edler, F. et al.: Fault tree analysis in theory and practice (2015), pp. 20–23.

[130] Deutsches Institut für Normung: DIN 25424 Fault Tree Analysis (1981).

of the ECU or a failure of the actuators. In the EPS state above v_{Switch}, a failure of the electric motor already causes a failure of the active steering, whereas in Hybrid state below v_{Switch} the active steering only fails if both the electric motor and the modulator fail simultaneously.

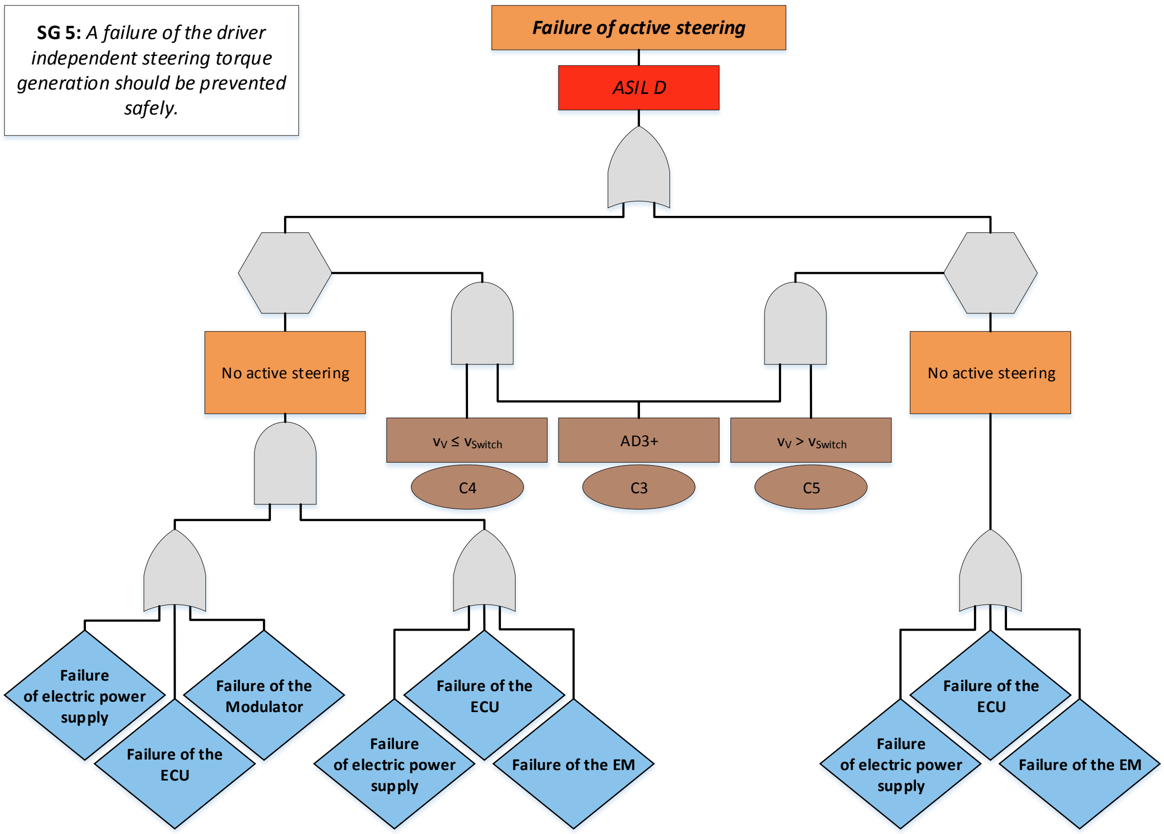


Figure 5-7: Fault tree analysis (FTA) for SG 5

The safety concept derives functional safety requirements from the safety goals. Theoretically, safety goals are abstract and superior safety requirements, but the safety goals refer to the overall system and not only to sub-systems, elements, components and their functions.[131]

Because of its high value, the functional safety concept has to be verified regarding its consistency with the safety goals and its ability to avoid hazardous situations. To avoid misunderstandings during the evaluation and to support an efficient method, an appropriate presentation of the functional safety concept is necessary. For this purpose, safety concept diagrams (SCD) are recommended by literature.[131]

Safety concept diagrams are related in the way of illustration to the system modeling language (SysML) respectively to requirement diagrams. SysML is based itself on the unified modeling language (UML). Figure 5-8 shows the exemplary SCD for SG 5.

The SCD is structured like an inverse tree, where the root of the tree is always a safety goal (SG). An identity (ID), an ASIL and a short description of the safety goals characterize the

[131] Gillen, C.: Dissertation, Development of efficient safety concepts for steering systems (2015), p.63.

safety goals. An ID, an ASIL and a short description of the requirements in the model also characterize the functional safety requirements (FSR). In this thesis, aggregations are used to illustrate that one or more requirements are necessary to fulfill superior functional safety requirements or safety goals. In the example of Figure 5-8, the FSR 5.1.1 and the FSR 5.1.2 are mandatory to fulfill the FSR 5.1. Furthermore, the aggregation transmits the ASIL, i.e. the lower FSR of a superior FSR with an ASIL D also have ASIL D.

Figure 5-8 describes the FSR of the safety concept, which have to be fulfilled to achieve SG 5, which is defined as “A failure of the driver-independent steering torque generation should be prevented safely”. This safety goal has ASIL D and is fulfilled if all first-level FSR beneath it, which are FSR 5.1, FSR 5.2, FSR 5.3 and FSR 5.4, are fulfilled. The SCD is explained exemplarily using the FSR 5.1, which consists of the sub-requirements “A failure of the electric power supply should be detected safely.” (FSR 5.1.1) and “In case of a failure of the electric power supply driver should be informed and the energy supply by a shunt circuit with a second redundant battery until the transition to a defined safe state is completed should be ensured.” (FSR 5.1.2).

A more extensive description of the FTA and SCD of the other safety goals is not content of this thesis, as the methodology of determining them has no new value. The crucial results of this thesis are the description of the final consequences of the safety concept and its implementation in the system architecture. Both are described below.

The central functional safety requirements and their implementation in the system architecture are named in Table 5-8. Furthermore, the safety goals allocated by the different functional safety requirements are listed here. The necessary redundant active steering functionality requires redundancies of the active steering actuators, the active steering control, the system observation and the electric power supply. To ensure that the driver can override the system, correct sensing of the steering wheel input by a redundant sensor setup is indispensable as well. The requirement for a redundant power steering functionality is fulfilled automatically if redundant active steering functionality is available. All safety goals are assigned at least to one FSR, whereby the FSR are verified. The implementation in the system architecture is described in the following subchapter in more detail.

Table 5-8: Functional safety requirements and their implementation

Functional safety requirements	**Implementation in system architecture**	**Allocated safety goals**
Redundant active steering actuators	Electric motor & modulator	SG 2, SG 3, SG 5
Redundant active steering control	Redundant ECU	SG 1, SG 2, SG 3, SG 4, SG 5, SG 7
Redundant system observation	Redundant ECU	SG 1, SG 2, SG 3, SG 4, SG 5, SG 6, SG 7
Redundant electric power supply	Generator & redundant batteries	SG 2, SG 3, SG 4, SG 5, SG 7
Redundant sensing of steering wheel input	Redundant steering wheel sensor	SG 1, SG 2, SG 3, SG 6, SG 7
Redundant power steering	EPS & HPS	SG 2, SG 3, SG 7

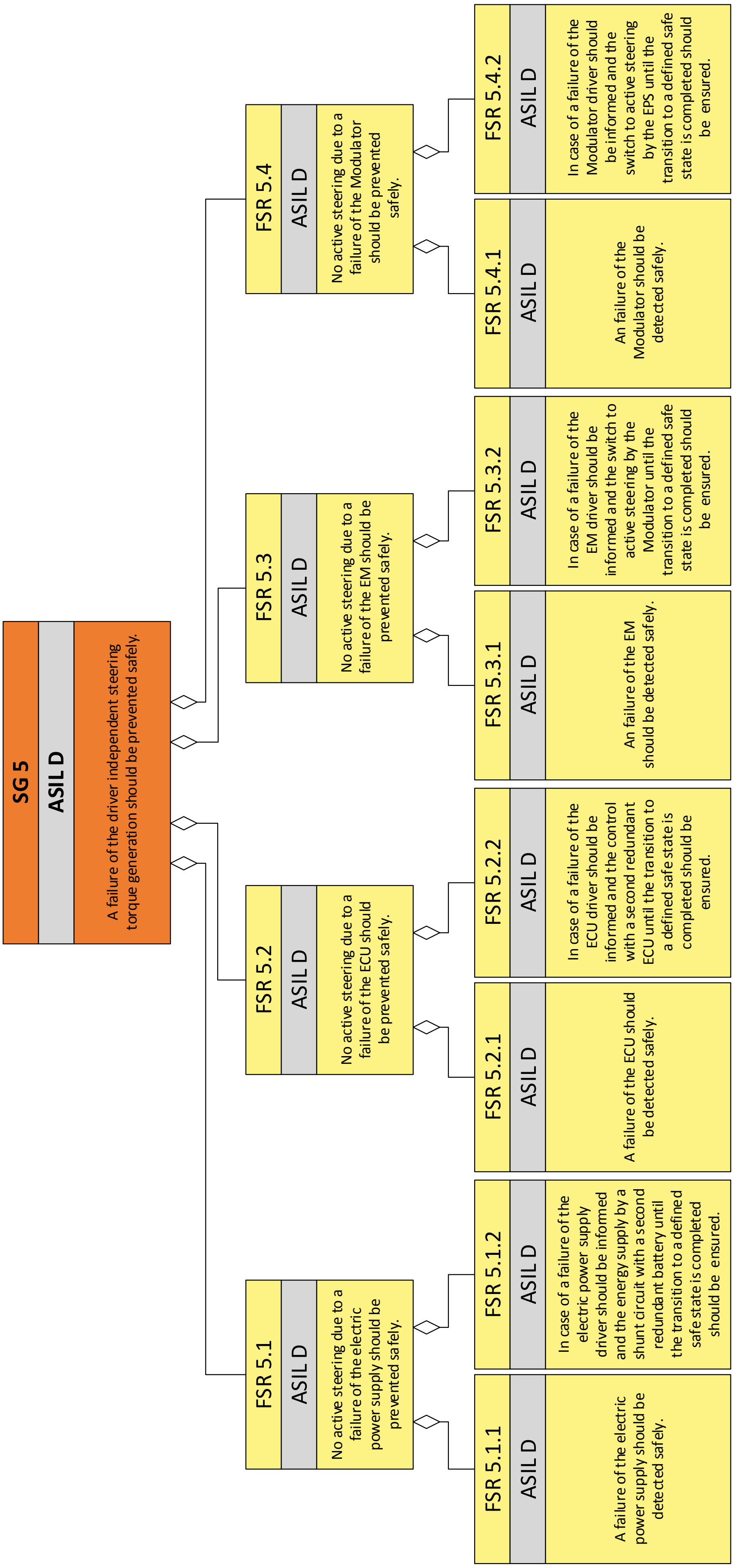


Figure 5-8: Safety concept diagram with functional safety requirements (FSR) for SG 5

5.3 Redundant Active Steering System

Different system designs are derived based on the determined requirements, the developed functional structures for RASS A and RASS B, and on the results of the safety analysis. All designs have the same components in common, but their structure differs. The components of the RASS are listed in Table 5-9.

The results of the safety analysis in Table 5-8 in combination with the operational and the redundancy requirements in Table 4-9 determine the composition and the specifications of the components of the RASS. The electric power steering (11) and the active hydraulic power steering (15) form the required redundant active steering system. The detailed purposes of each component of the RASS are described hereafter.

Table 5-9: Components of the RASS

ID	Component	ID	Component
1	Steering wheel	12	Electric motor
2	Redundant steering wheel torque & angle sensor	13	Overlay gear
3	Recirculating ball steering gear	14	Electronic control unit (ECU) 1
4	Internal combustion engine	15	Active hydraulic power steering (AHPS)
5	Redundant electric power supply	16	Active valve with modulator
6	Generator (24 V, 110 A)	17	Variable power steering pump (vPSP)
7	Battery isolator	18	Tank
8	Battery 1 (24 V, 160 Ah)	19	ECU 2
9	Battery 2 (24 V, 160 Ah)	20	Pitman arm
10	Diode	21	Push rod
11	Electric power steering (EPS)	22	Tie rod

5.3.1 Sensor and Controller

The redundant steering wheel sensors (2) ensure a safe detection of the driver's intention for a correct steering assistance in manual driving mode. During AD3+, the redundant sensors are responsible for a correct detection of a driver intervention or the observation of the transition process during a hand-over from manual to automated driving or reverse.

Two electric control units are implemented for the redundant active steering control, whereas the first ECU (14) is primarily responsible for the control of the electric motor and the second ECU (19) for the control of the active valve. This means that in normal operation, exactly one ECU is active per active steering subsystem. However, in the background the signals are continuously checked for plausibility by the other ECU in the system. The ECU thus take over the monitoring function among each other and determine which ECU is affected in the event of a fault by means of suitable structure and procedures.[132]

[132] Isermann, R.: Fault tolerance in mechatronic systems (2016), pp. 43–45.

5.3.2 Electric Power Steering System

The electric power steering (EPS) (11) serves as one of the two active steering systems, which are able to generate steering torque independently of the driver. In order to increase the efficiency of the RASS, the EPS also serves as the sole power steering as long as its maximum torque is sufficient. It is the fallback level for the AHPS in the event of a fault of the active valve.

To implement this functionality, the EPS consists of an electric motor (12) as actuator, a torque overlay gear (13) that reduces the rotational speed of the EM and overlays the electrically generated torque on the steering gear, and an already described ECU (14). The assembly position of the EPS and the connection to the other components of the RASS via the overlay gear is variable. A different assembly position causes different functionalities, as described for RASS A and RASS B, or just advantages or disadvantages regarding the assembly space.

5.3.3 Active Hydraulic Power Steering

The other active steering system of the RASS is the active hydraulic power steering (AHPS, 15). The AHPS supports the EPS if its torque is not sufficient for the power steering or the automated steering and serves as a fallback level in the event of a fault of the EPS.

Besides the standard HPS components such as the variable power steering pump (vPSP, 17) and tank (18), the AHPS has an additional active valve (16) and ECU (19) for the driver-independent generation of steering torque. The last two components are new compared to a standard HPS and are responsible for turning the passive HPS into an active steering system suitable for automated driving.

5.3.4 Electric Power Supply

For driverless steering, the RASS requires electrical power. The vehicle's electrical power network is responsible for this. In AD3+, the electric power supply must provide sufficient power at all times to enable the RASS to safely steer the vehicle. In the event of a fault, the electric power supply must provide at least as much power and energy as it is required to reach a defined safe state.[133]

Therefore, the redundant electric power supply (5) consists of a generator (6) and two parallel batteries (8, 9). An isolator (7) and a diode (10) isolate the two batteries to ensure power supply even in case of a fault of one battery.

[133] The redundancy requirement regarding the fallback energy is defined in Table 4-9.

5.3.5 Other Components

The remaining components are not connected to one of the determined requirements directly, but they are indispensable for the other components to fulfill them. The internal combustion engine (4) runs the power steering pump (17), which transports the hydraulic liquid from the tank (18) to the active valve (16). The pitman arm (20) is the output interface of the recirculating ball (RCB) steering gear (3) to the steering kinematics. The push rod (21) transmits the output steering power from the pitman arm to one steered wheel, which is connected to the other wheel by the tie rod (22).

5.3.6 System Design

The components described above do not represent an innovation in themselves. Here, the components are assembled into innovative system architectures for the RASS regarding the comprehensive requirements identified in this thesis. The first system design RASS A is shown in Figure 5-9.

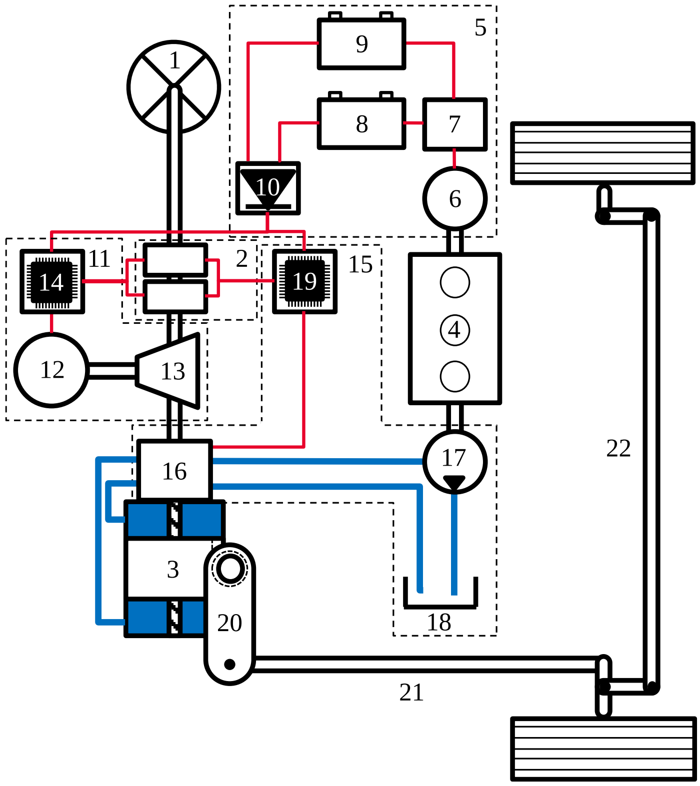


Figure 5-9: System architecture RASS A

Its particular feature is that the EPS (11) is connected to the steering system at the input shaft of the active valve (16) by a gear (13) to overlay torque to the driver's input torque. Because the EPS is mounted upstream of the valve, it is able to control the valve as well besides the driver and the AHPS (15). Thereby, an additional possibility to control the active valve for example in the event of a fault of the modulator is enabled on the one hand. On the other hand, this causes the problem that the torque of the EPS is transmitted through the torsion bar, which makes its design very difficult. Usually the stiffness of the torsion bar is designed according to the maximum input torque of the driver and has a significant influence on the steering feel. Because the maximum torque of the EPS is considerably higher than that of the driver, such a design would usually cause a complete twisting of the torsion bar and a steering with play in the center position when the EPS is actuated. If the stiffens of the torsion bar was designed according to the maximum torque of the EPS, it would significantly increase the driver's effort to control the valve and thus cause a sluggish steering feel. A torque split between the driver and the EPS solves this issue as long as the EPS does not fail. Another drawback of this design is that the output torque generated by the EPS and that by the AHPS are not controllable independently of each other, because the EPS has a direct influence on the active valve.

Figure 5-10 shows the second possible system design RASS B1. Compared to the first system design, the EPS (11) is mounted by the torque overlay gear (13) at the shaft between the active valve (16) and the RCB steering gear (3). Thereby, the opportunity to control the active valve by the EPS does not exist, but the difficulties with the stiffness of the torsion bar neither. The stiffness of the torsion bar is designed according to the driver's maximum input torque. Because the torque of the EPS is added downstream the active valve, the control of the EPS and the AHPS are independent.

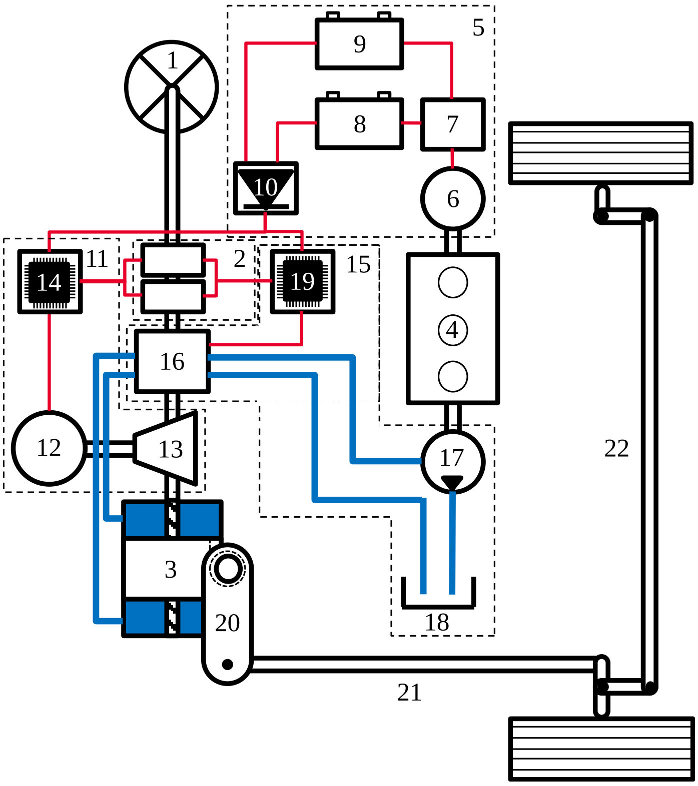


Figure 5-10: System architecture RASS B1

The third possible system design RASS B2 is illustrated in Figure 5-11. Compared to the first two system designs, the EPS (11) is connected at the end of the input shaft of the steering gear (3) by the overlay gear (13). This system design has the same functionality as RASS B1, but the difference is the mounting position of the EPS below the steering gear. Because the assembly position of the steering gear compared to the steered axle is relatively fixed to minimize the influence of the suspension on the steering system and thus also the position of the steering gear compared to the cab floor is more or less fixed, the assembly space above the steering gear is limited. The mounting of the EPS below the steering gear enables a significantly bigger assembly space for the EPS, but also for the active valve above the steering gear. The space below the steering gear is limited only by the ground clearance of the vehicle.

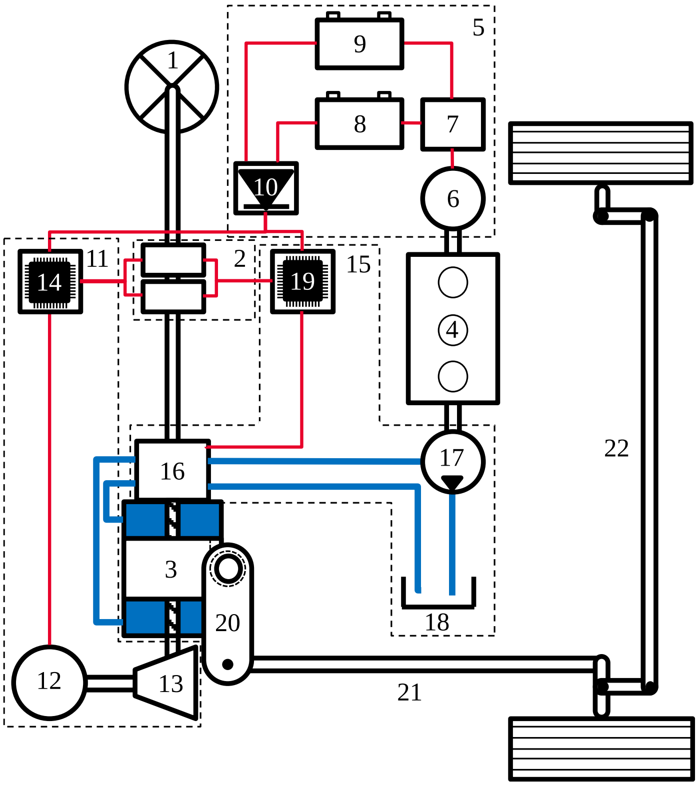


Figure 5-11: System architecture RASS B2

A fourth possible system design that is considered has the same functionality as RASS B1 and RASS B2, but the EPS (11) is mounted at the output shaft of the steering gear, by an overlay gear (13) close to the pitman arm (20). However, this design has the big drawback that EPS does not use the already available transmission from the input shaft to the output shaft of the steering gear to amplify its torque at the pitman arm. Hence, for this steering design the required ratio of the overlay gear (13) is significantly higher than for the other system designs. A higher required gear ratio not only increases the costs of the overlay gear, but also increases its dimensions and its weight. Therefore, this system design is not considered as a suitable solution in this thesis.

5.4 Conclusion on New System Architecture

Two suitable functional structures (RASS A & RASS B) are developed based on the requirement analysis in chapter 4. The item definition according to the developed functional structures is used as an input for the safety analysis, which results in several safety goals for the RASS. These safety goals are used to develop a suitable safety concept and to derive different functional safety requirements.

The final safety requirements as well as the frame requirements, the operational and the redundancy requirements serve as a basis for the determination of the specifications of the subsystems and components of the RASS. These components are combined into three different suitable system designs, whereas the functionality of RASS A differs from the functionality of the other two system designs (RASS B1 & RASS B2). Each of these system architectures represents an innovation, since there is not a single steering system in the SoA that meets all the requirements determined in this thesis for AD3+ of trucks.

RASS B2 is selected as the final system design for the further thesis. It is superior compared to the other system designs, since it allows an independent control of the EPS and the AHPS and has advantages over the other designs in terms of assembly space.

6 State Transitions

The increased functionality of the new Redundant Active Steering System (RASS) and its increased complexity compared to standard steering systems of trucks require a strategy for the operation of the RASS. Such a strategy is developed in this chapter. Therefore, all potential states, which are necessary for the intended functionality of the RASS, are first determined and described. Various indicators are used to determine the current state of the system.

The operation strategy distinguishes between three types of state transitions on the first level, which are transitions of the power steering state, transitions of automation state and transitions caused by system faults.

6.1 System States

The RASS and its components require different states to provide the high functionality. The operation strategy of the RASS uses several state indicators to identify the currently required state for the system and transmits the system to it. All state indicators used by the operation strategy are listed in Table 6-1.

The automation state indicator (a) describes if the vehicle drives manually or automatically on the one hand and the behavior of the driver on the other hand. The driver is inactive during automated driving and remains inactive until the system requests him to take over. In contrast, the driver gets active if he intends to intervene and to override the automated system. During manual driving, the driver is obviously active to steer the vehicle. In case the driver intends to transmit to automated driving, he sends a hand-over command to the system and becomes inactive.

The different power steering states of the RASS are signaled by the power steering indicator (p). It defines if the electric power steering (EPS) state, the stand-by hybrid state, the active hybrid state or the hydraulic power steering state is enabled. The first three power steering states are operational states, whereas the hydraulic power steering state is only activated, in case of a fault of the EPS.

This EPS fault as well as a fault of the active valve are displayed by the fault state indicator (f). The strategy differs between errors and failures. In the event of an error, the affected system delivers an incorrect functionality, whereas it has no functionality in case of a failure. Hence, a deactivation of the affected system is required only in case of an error.

The transition progress indicator (tp) observes the progress of the state transition process. It may be the case that an intended state transition is not possible, e.g. due to non-fulfilled transition conditions. Another possibility is that a transition is not completed successfully, because a system fault occurs during the transition process, whereby the transition is not possible anymore and fails. The transition progress indicator shows if the transition process is not possible, still in progress, completed or failed. The state transition process is shown in Figure 6-1 for a generic state transition.

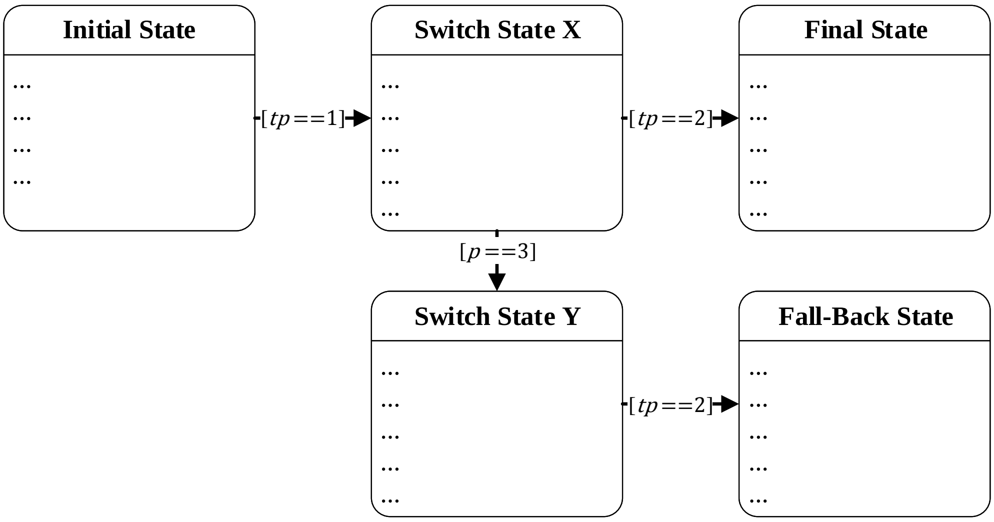


Figure 6-1: Process of a generic state switch

In the event of a system fault during AD3+, when no driver is available to take over the control, the RASS enables the vehicle to drive into a defined safe state. The safe state indicator (s) signals the RASS whether the vehicle is already in a safe state or not.

Irrespective of the level of automation, the RASS issues a take-over request to the driver when a fault occurs. The take-over request indicator (tor) displays whether a take-over is requested by the system and in case it is, for how long it is already requested. This period is important in the event that the driver does not comply with the ToR. If the driver does not comply with the ToR within a defined period, the vehicle transfers itself into a safe state as long as the RASS is still operational.

The last two indicators, the vehicle's velocity indicator (v) and the steering torque indicator (m), are decisive for the selection of the power steering state of the RASS. To increase the efficiency of the RASS and to decrease its hydraulic losses, it intends to perform as many driving situations in EPS state as possible. Because the steering torque requirements are attainable for all driving situations above a defined velocity of the vehicle, the HPS is not required here and is disabled. Below this velocity threshold, the RASS enables the HPS, but only uses it in the event the EPS maximum steering torque is not sufficient for the required torque. For this purpose, the vehicle's velocity indicator (v) shows whether the velocity is below or above the threshold (v_{Switch}) and the steering torque indicator (m) displays whether the EPS maximum torque is sufficient for the required steering torque or not.

Table 6-1: System state indicators

Indicator	Value	Description
Automation state indicator		
a	0	Driver active, manual driving
a	1	Driver inactive, hand-over command, manual driving to automated driving (intended by the driver)
a	2	Driver active, intervention, automated driving to manual driving (intended by the driver)
a	3	Driver inactive, take-over request, automated driving to manual driving (intended by the system)
a	4	Driver inactive, automated driving
Power steering state indicator		
p	0	Electric power steering state
p	1	stand-by hybrid state
p	2	active hybrid state
p	3	Hydraulic power steering state
Fault state indicator		
f	0	No fault
f	1	EPS failure, EPS deactivated
f	2	Active valve failure, Active valve deactivated
f	3	EPS error
f	4	Active valve error
Transition progress indicator		
tp	0	State transition not possible
tp	1	State transition in progress
tp	2	State transition completed
tp	3	State transition failed
Safe state indicator		
s	0	Vehicle not in safe state
s	1	Vehicle in safe state
Take-over-request indicator		
tor	0	No take-over request
tor	1	Take-over request for less than x seconds
tor	2	Take-over request for more than x seconds
Steering torque indicator		
m	1	$M_{\mathrm{PS}} \leq M_{\mathrm{EPS,max}}$
m	2	$M_{\mathrm{PS}} > M_{\mathrm{EPS,max}}$
Vehicle's velocity indicator		
v	1	$v_{\mathrm{V}} < v_{\mathrm{Switch}}$
v	2	$v_{\mathrm{V}} > v_{\mathrm{Switch}}$

The main states of the RASS and their indicators are illustrated in Figure 6-2. The composition of the torque at the pitman arm (M_{P}), the driver's torque at the steering wheel (M_{H}) as well as the torque (M_{AHPS}), volume flow (Q_{PSP}) and active valve angle (φ_{AV}) of the AHPS are defined for each state as well.

The four operational states of the RASS are the combinations of manual driving and automated driving with EPS state and hybrid state. As long as the system is operated correctly, it is in one of these four states. The other four states are only relevant in the event of a fault inside the RASS. The target state is determined based on the type of fault and the state in which the RASS is at the time the fault occurs.

The processes of the transition of the power steering states, of the automation states and the transitions caused by faults are described in the following subchapters in detail.

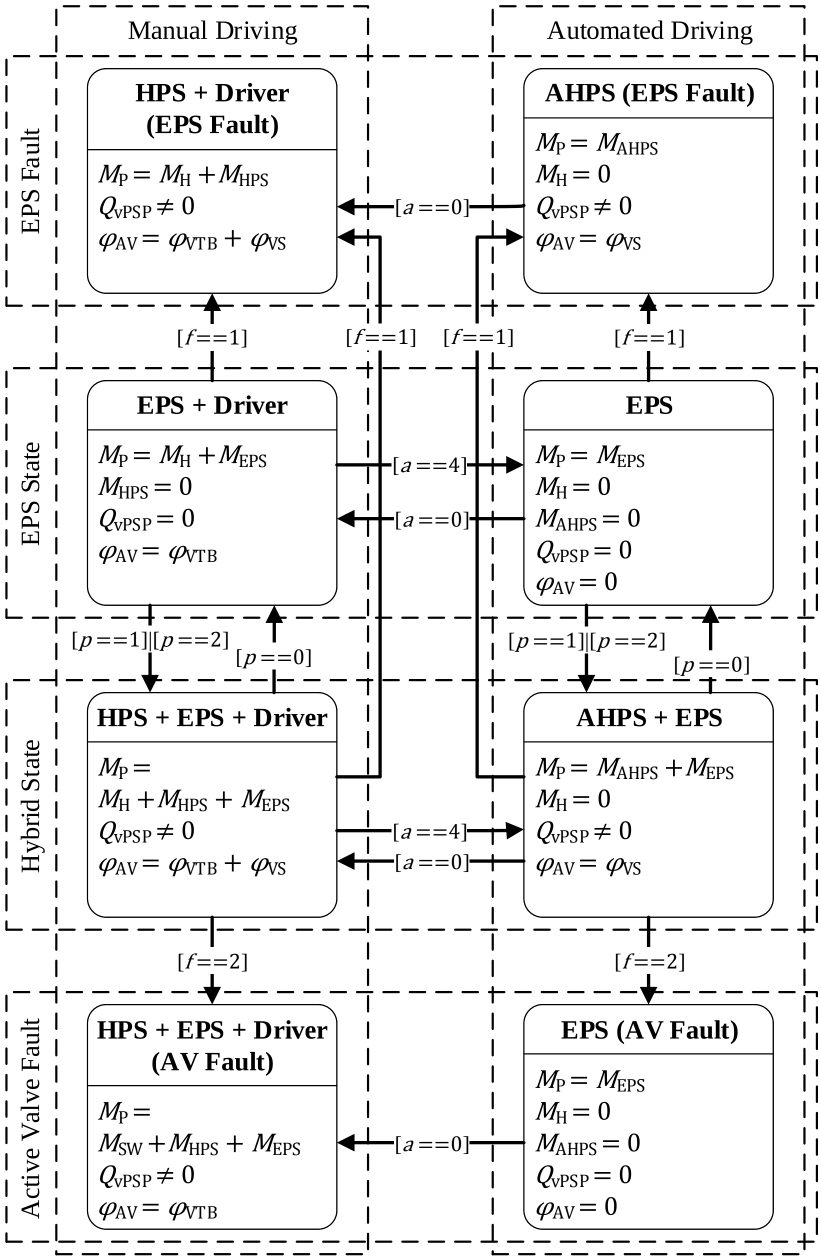


Figure 6-2: Main states of RASS

6.2 Power Steering States

The transitions of the power steering states are described first, since the power steering state is independent of the automation state, but not vice versa. Starting with the description of the intended functionality of each power steering state and the conditions for a transition between the states, this subchapter concludes with the implementation of the different power steering states in the system and component design.

6.2.1 Intended Functionality

The base for the determination of the power steering state is shown in Figure 6-3. The starting point for determining the required power steering torque at the pitman arm (M_{P}) is, on the one hand, the steering wheel torque of the driver (M_{H}) and, on the other hand, the current vehicle velocity (v_{V}). The required torque at the pitman arm (M_{P}) is the sum of the torque required by the automated driving system (M_{AD}) and the servo steering torque (M_{Servo}), which is defined by the servo characteristic based on the driver's input torque and the velocity.

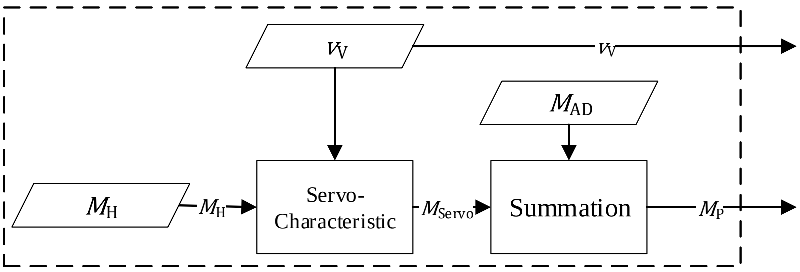


Figure 6-3: Base for the determination of the power steering state

The goal of the different power steering states and their transitions, which both are shown in Figure 6-4, is to increase the efficiency of the RASS by decreasing its hydraulic losses compared to a standard truck HPS. This is achieved by a reduced and demand-oriented use of the HPS. Since the required steering torques in the high velocity range above a defined velocity threshold (v_{Switch}) do not exceed the maximum torque of the EPS, it is permissible to deactivate the HPS by reducing the hydraulic volume flow (Q_{vPSP}) to zero and steer solely electrically. This prevents hydraulic losses in the steering system. Only potential losses in the power steering pump remain. This state is called EPS state ($p = 0$).

Below this velocity threshold (v_{Switch}), it is possible that steering torques are required which the EPS cannot deliver. In that case, the HPS provides the missing torque. However, during most of the driving maneuvers in the lower velocity range, the maximum EPS torque is sufficient. In order to use this efficiency potential, sole electric steering is intended in these situations as well. Because high required torque peaks can occur in this velocity range at very short notice at any time, it is necessary to have the HPS constantly ready (stand-by) for operation to support the EPS. A deactivation of the HPS is not possible, but it could be disabled by keeping the active valve in center position ($\varphi_{AV} = 0$). This state is called stand-by hybrid state ($p = 0$), whereas it is called active hybrid state ($p = 2$) if the HPS supports the EPS. A way to turn off the hydraulics in the stand-by hybrid state is developed and described in the following section.

In order to avoid constant changes between the EPS state and the hybrid state, a hysteresis is implemented in the system by making the velocity threshold variable. If the RASS is in EPS state, the threshold is set to a lower value ($v_{\text{Switch,low}}$) and to an upper value ($v_{\text{Switch,high}}$) as soon as it transfers to the hybrid state.

The RASS is also suitable for a solely HPS state ($p = 3$). Because this state does not lead to an increase in efficiency compared to a standard HPS, it is only used as a fallback state in the event of an EPS fault. Reversely, the EPS is used as a fallback state for an HPS fault.

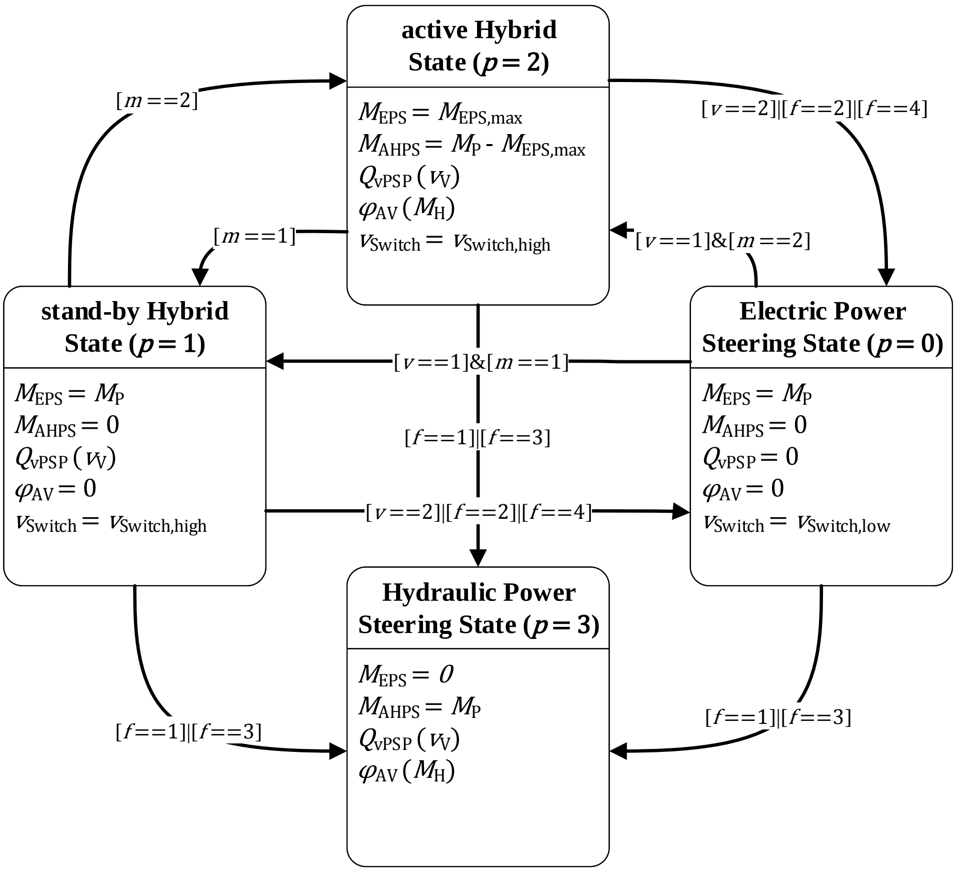


Figure 6-4: Intended power steering states

6.2.2 Implementation in System Design

The intended electric power steering state is easy to implement into the system design of the RASS. This state does not directly affect the system design, but it requires a variable power steering pump (vPSP) with an adjustable volume flow, as already available on the market today[134]. As explained before, the EPS State is selected in the velocity range above the threshold (v_{Switch}), where the maximum torque of the EPS is sufficient for all required steering torques. Hence, the HPS is deactivated by reducing the volume flow of the vPSP (Q_{vPSP}) to zero as shown in Figure 6-5, whereby the hydraulic losses are minimized. The EPS serves the required steering torque completely.

Below the defined velocity threshold, the RASS is in hybrid state and the PSP is enabled. Since the potentially required steering torque increases with decreasing driving velocity, the volume flow that directly influences the torque of the HPS is increased with decreasing velocity as well. Exemplary characteristics of the volume flow of the PSP over the vehicle's velocity are shown in Figure 6-5.

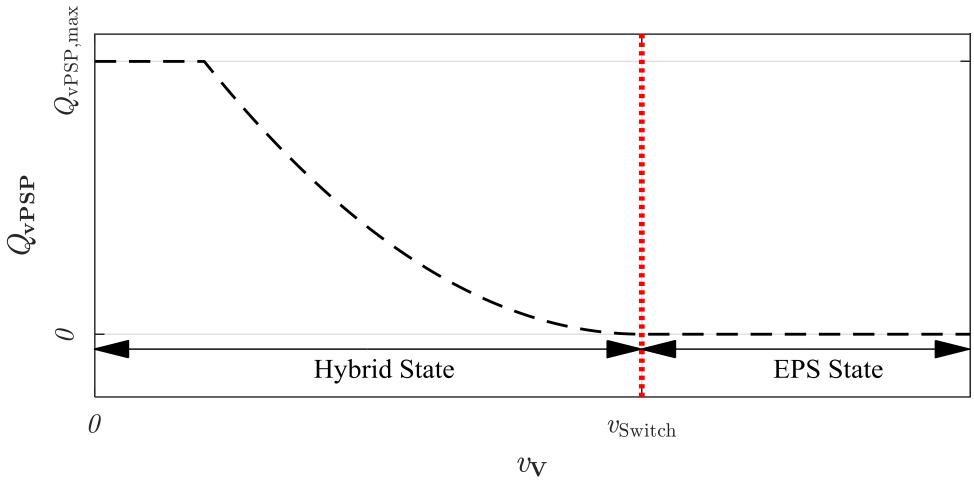


Figure 6-5: Variable volume flow of PSP

The implementation of the hybrid state in the system design, especially of the intended stand-by hybrid state requires an innovative approach. Figure 6-6 illustrates this approach for manual driving. Decisive for the desired functionality is a rotary slide valve, which can be actuated not only by the twist of the torsion bar by the driver, but also by a valve sleeve rotatable by a modulation actuator. By tracking the torsion of the torsion bar with the adjustable valve sleeve, it is possible to keep the valve in center position during the stand-by hybrid state as long as the maximum EPS torque is sufficient for the required steering torque (M_{req}). In this state, the torsion angle of the sleeve (φ_{VS}) is equal to that of the torsion bar (φ_{VTB}), whereby the resulting angle of the active valve (φ_{AV}) remains zero. The required steering torque at the pitman arm (M_{P}) is served solely by the EPS torque (M_{EPS}), besides by the torque of the

[134] Lauth, H. J. et al.: Needs based controllable pumps (2002), pp. 110–111.

driver at the steering wheel (M_{H}). The torque of the active HPS (M_{AHPS}) is zero. This reduces hydraulic losses even while the pump is running.[135]

The active hybrid state is enabled as soon as the torque required at the pitman arm exceeds the maximum torque of the EPS. The valve sleeve is turned back so that the active valve is opened and the AHPS generates the missing torque. An exact knowledge of the valve characteristics and an exact and fast control of the valve sleeve are necessary in this case so that the transition from stand-by hybrid state to active hybrid state and vice versa is controlled in such a way that the driver does not recognize any of this in the steering feel. However, the continuous hydraulic volume flow through the valve even in center position during the stand-by hybrid state does not cause any steps in the steering torque and thus supports a smooth steering feel. If the required steering torque (M_{req}) exceeds the available power steering torque, the difference must be applied by the driver through additional torque at the steering wheel (M_{H}). As a result, the active valve continues to rotate at very high required torques without generating additional servo torque.

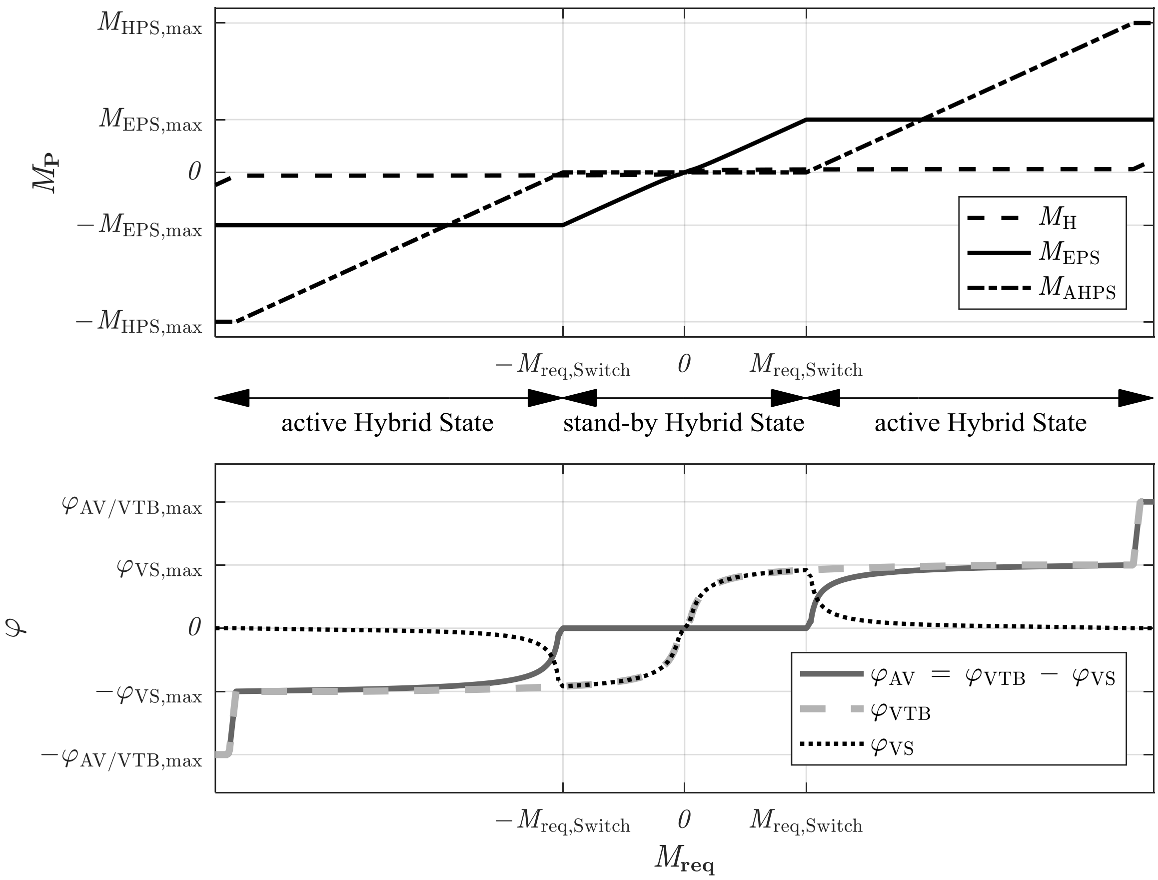


Figure 6-6: Hybrid state in manual driving with required steering torque (M_{req}), steering torque at pitman arm (M_{P}) and valve angles (φ)

135 This approach can save up to 712 W of hydraulic power losses at static torque. The proof is attached in the appendix A.5.

The hybrid state for automated driving is shown in Figure 6-7. In contrast to manual driving, the torsion bar of the active valve is not twisted here because the driver does not apply torque to the steering wheel ($M_{\mathrm{H}} = 0$) during automated driving. This facilitates the implementation of the stand-by hybrid state, as the torsion bar (φ_{VTB}), the sleeve (φ_{VS}) and thus the overall valve (φ_{AV}) all remain in center position without tracking. The required torque (M_{req}) is also served solely electrically by the EPS.

If this torque is not sufficient anymore, the active valve is opened by turning the valve sleeve to generate the additionally required steering torque. To achieve the same valve angle (φ_{AV}) as by the driver in manual mode, the sleeve (φ_{VS}) is rotated by the same amount as the torsion bar (φ_{VTB}) in manual mode in the opposite direction.

By the adjustable sleeve of the active valve and by the tracking of the sleeve during manual driving, it is possible to reduce the hydraulic losses not only in EPS state, but also in stand-by hybrid state.

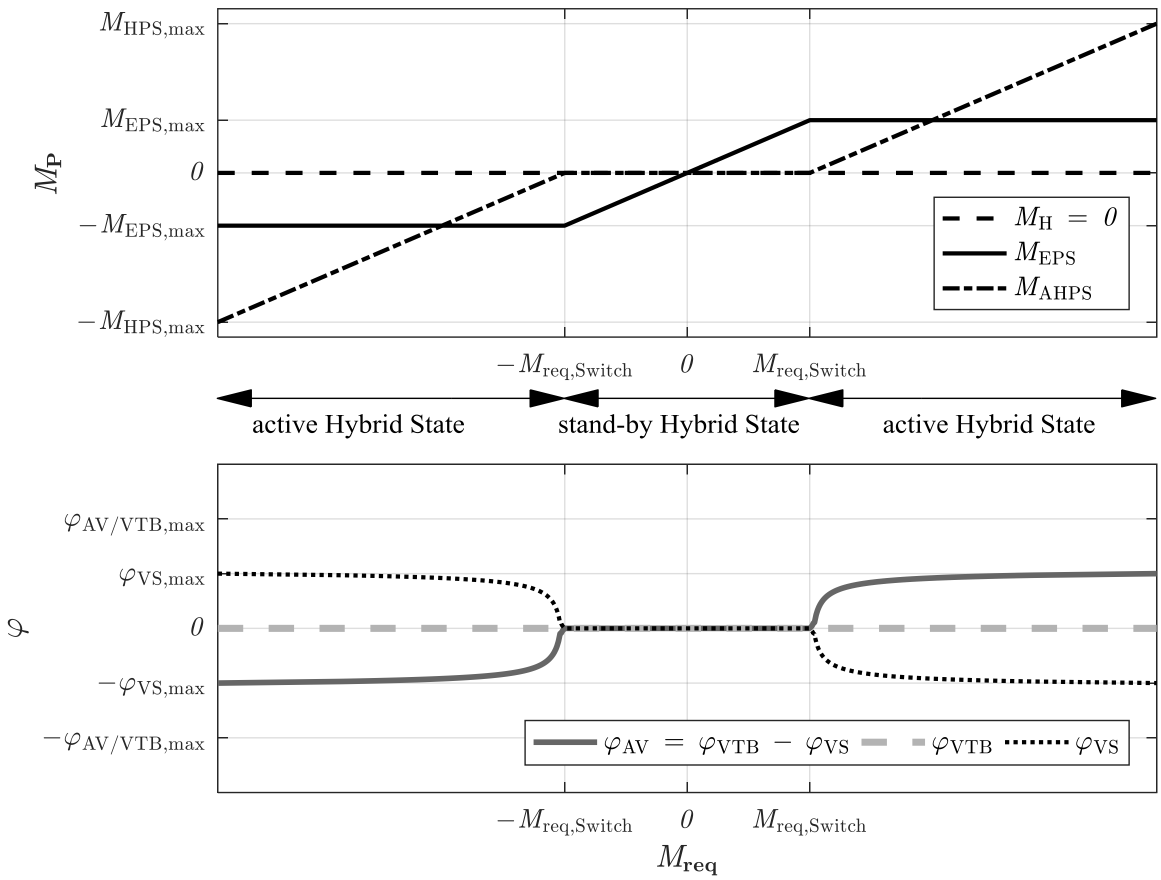


Figure 6-7: Hybrid state in automated driving with required steering torque (M_{req}), steering torque at pitman arm (M_{P}) and valve angles (φ)

6.3 Transitions of Automation State

Besides the two automation states manual driving and automated driving[136], there are three different types of transitions between these automation states. Table 6-1 already introduced the state indicators for those states and transitions. The three types of transitions are the hand-over from manual driving to automated driving, the take-over by the driver from the automated driving and the driver intervention in the automated driving. The following section describes the intended functionality of the RASS concerning these transitions and discusses the focus on the implementation of the driver intervention. Its implementation in the system design is developed afterwards.

6.3.1 Intended Functionality

The operation strategy of the RASS differs in two automation states and three types of transitions between them. Figure 6-8 shows the corresponding intended functionality of the RASS, which is independent of the previously described power steering state.

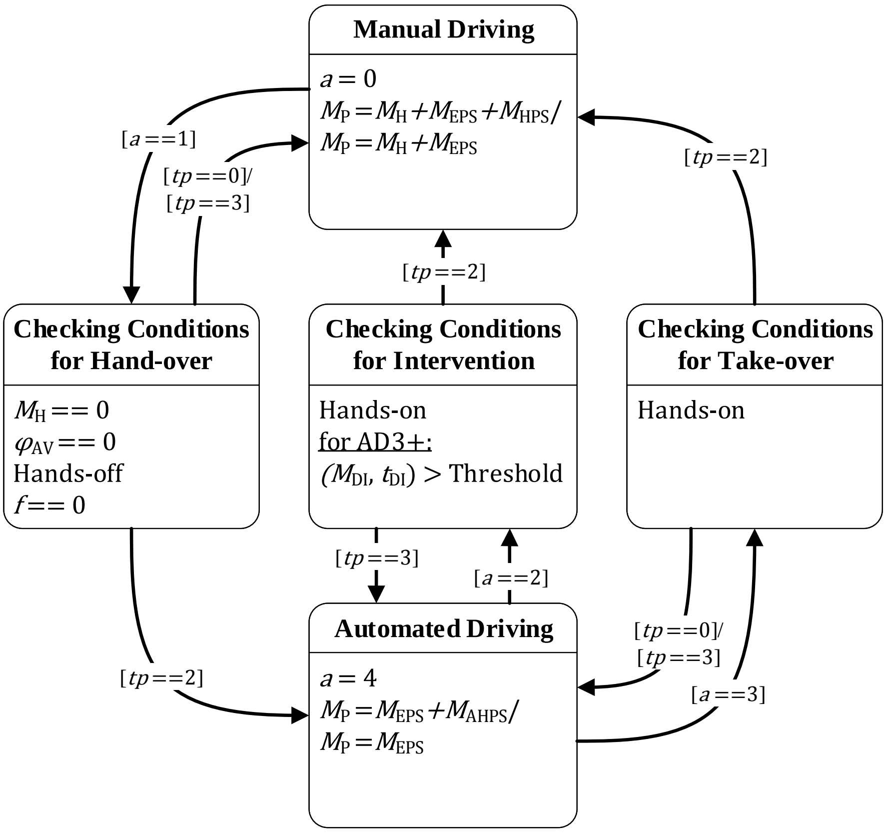


Figure 6-8: Intended functionality for transitions between MD and AD

[136] The focus in this thesis is on AD3+.

In manual driving state ($a = 0$), the driver controls the steering system and is assisted by the EPS or by both EPS and HPS, whereas the driver is inactive during automated driving state ($a = 4$) and the steering is automatically controlled, either solely by the EPS or hybrid by the EPS and the AHPS. The exact function for both the manual driving and the automated driving is described in the previous subchapter on power steering.

In this thesis, the term hand-over describes the transition from manual driving to automated driving. This is the simplest of the three transitions, as the driver has to initiate it consciously, which means that he is aware of the situation. When the driver starts the transition process in manual driving state ($a == 0$), the automation state indicator jumps to ($a == 1$) and the system checks whether the conditions for a hand-over are fulfilled. If the driver applies no torque at the steering wheel ($M_{\mathrm{H}} == 0$), the active valve is in center position ($\varphi_{\mathrm{AV}} == 0$), the driver has his/her hands off the steering wheel and no fault occurs ($f == 0$), the conditions for the hand-over are fulfilled. As soon as the transition process is completed ($tp == 2$), the RASS is in automated driving state ($a == 4$).

The first type of transition from automated driving state to manual driving state is the transition intended by the automated system, e.g. because the system reaches its limits. This transition is called take-over in this thesis. If the automated system is no longer able to safely control the vehicle, it requests the driver to take over the control ($a == 3$). During AD3+, this happens with a certain lead time, so that the driver has enough time to become aware of the situation again in order to safely take over control. The transition to manual driving state is carried out when the hands-on condition for a take-over is fulfilled. The amount of time that the driver has to take over the control has been investigated in various publications and varies from five seconds[137,138] to 15 seconds[139]. The steering torque generated independently of the driver is reduced to zero as soon as the driver has taken over the control.

The second type of transition from automated driving state to manual driving state is called driver intervention, because it is not intended by the automated system but by the driver. This transition occurs, for example, if the automated system makes an incorrect steering input and the driver wants to correct it. Thereby, the Vienna's convention on road traffic also requires the driver to have control over the vehicle at all times.[140] This is only guaranteed if the driver can switch off the automated system and switch to manual driving state by intervening. However, it is not practicable to make an immediate transition in case the driver's intervention is very subtle.

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[137] Ito, T. et al.: Time Required for Take-over from Automated to Manual Driving (2016), p.6.

[138] Gold, C. et al.: Take over (2013), p.1940.

[139] Becker, J.; Helmle, M.: Architecture and System Safety Requirements for AD (2015), p.42.

[140] United Nations: Convention on Road Traffic (1968), p.11.

On the one hand, it should be avoided that the driver unintentionally interferes the automated steering system, on the other hand it is essential for safety that the take-over time in an emergency situation is as short as possible. The driver interferences on the automated steering are avoidable by high torques on the steering wheel that prevent the driver from overriding, and by a SbW. Nevertheless, it has to be ensured that the driver is able to intervene directly in critical situations and override the system. Both presented options, the high torque solution and the SbW solution, create problems to implement this emergency behavior. A new approach is required.[141a]

Since the RASS is no SbW system, the driver interferences are avoided by high steering wheel torques. Since an additional device for the input of an intervention request by the driver[141a] is too complicated, not intuitive and probably too slow in a critical situation, the approach chosen in this thesis is that the system recognizes an intervention by the driver only via his steering wheel input, but avoids unintended interferences. The implementation of this intended functionality is described in the section hereafter.

If the driver initiates the intervention in order to switch to manual driving ($a == 2$), the system checks the following transition conditions. The steering wheel torque by the driver (M_{H}) has to extend the intervention torque threshold (M_{DI}). However, to avoid unintended interferences of the driver, the system conducts the transition not immediately as soon as the steering wheel torque exceeds the torque threshold. The driver has to apply this amount of torque over a defined period of time to intervene the system. The system registers the time that the driver's torque exceeds the torque threshold. If this time exceeds the defined intervention time threshold (t_{DI}), the system conducts the transition. If the driver's torque exceeds the torque threshold for less time than the defined time threshold, no transition happens.

The detailed abort action of the automated steering is not in focus of this thesis, but it can be designed, for example, as a steep decrease, as a slower fade out or a stepwise decrease. A complete shutdown of the automated system as soon as the driver intervention is detected would be the most effective intervention.[141b]

While other publications discuss whether a driver intervention is correct or not, this thesis does not address this issue. For the RASS, it is only important to recognize whether the driver intervention is intended or not and, if intended, to switch to manual driving state as fast as possible.

6.3.2 Example of Implementation

The implementation example of the intended functionality concerning the transition of the automation state focuses on driver intervention and the associated ability to override the

[141] Kalb, L.; Bengler, K.: Controlling Automated Steering (2018), (a) pp. 588–597 | (b) 592–595.

automated steering by the driver. The implementation of the previously described new approach for the driver intervention is illustrated in Figure 6-9. The torque the driver applies on the steering wheel (M_{H}) is shown here over the period of time (t) he applies the torque.

In order to exclude undesired interferences of the driver, but to transfer the control as fast as possible in the case of desired interventions by the driver, the torque applied by the driver is considered in combination with the application time in order to identify an intended intervention.

If the driver applies a smaller amount of torque at the steering wheel, he has to apply this for a longer period of time so that the operation strategy of the RASS recognizes the intervention request. This prevents, for example, the system from transferring to manual driving state due to an accidental touching of the steering wheel by the driver.

However, to guarantee a fast transition to the driver if he strengthens his transition request by applying a big amount of steering wheel torque, e.g. because of an emergency evasion maneuver, but also guarantee no unintended transitions, the torque threshold (M_{DI}) and the time threshold (t_{DI}) are dynamic values depending on each other. By using any mathematical relation, the target behavior is that the torque threshold (M_{DI}) decreases with an increasing application time of steering wheel torque. For example, this behavior could be achieved by an integral of the steering wheel torque over the application time or by a characteristic line as shown in Figure 6-9.

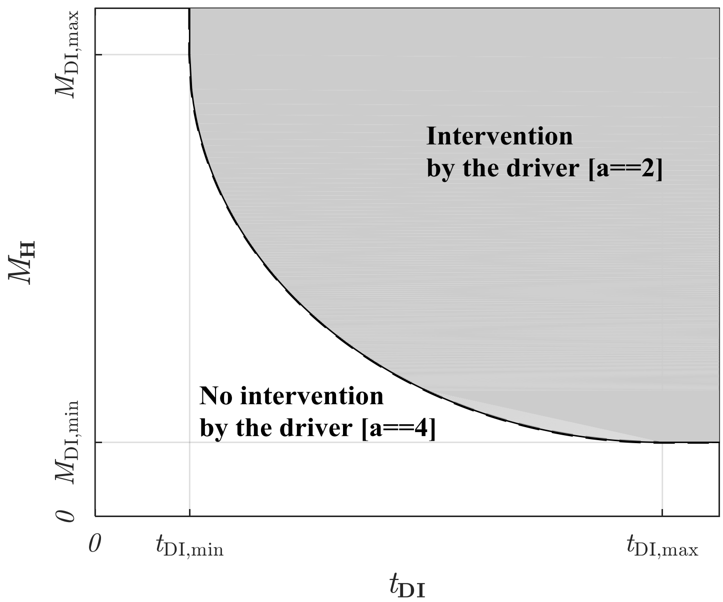

Figure 6-9: Driver intervention thresholds with the driver intervention time (t_{DI}) and the driver's torque at the steering wheel (M_{H})

In this example, the intervention time threshold is minimum ($t_{\mathrm{DI,min}}$) if the driver applies a high amount of steering torque to guarantee a fast transition to manual driving. If the driver applies exactly the minimum value of the intervention torque threshold ($M_{\mathrm{DI,min}}$), the intervention time threshold reaches its maximum ($t_{\mathrm{DI,max}}$) before a transition is initiated. The characteristic between the two extreme values ($t_{\mathrm{DI,min}}$ for $M_{\mathrm{DI,max}}$ & $t_{\mathrm{DI,max}}$ for $M_{\mathrm{DI,min}}$)

has to be monotonously decreasing and can be linear, degressive, or progressive curve. If the steering wheel torque of the driver is smaller than the intervention torque threshold or the time of application is shorter than the intervention time threshold or both, no transition is carried out.

The implementation of the steering torque generation during the manual driving state and during the automated driving state is already described in subchapter 6.2 about the different power steering states.

6.4 Transition to Fallback States

In addition to the already presented states in fault-free operation, the RASS also has additional fallback states that represent the system-internal fallback level in the event of a fault. This is necessary for AD3+, since per definition the driver is not available as an immediate fallback level if a fault occurs. Faults can occur both during manual driving and during automated driving[142]. Both cases are considered below. First, the intended functionality is discussed before the implementation in the system design is explained.

6.4.1 Intended Functionality

During manual driving state, the RASS primarily has power steering functionality in addition to possible comfort functions. Due to the two different power steering system, the RASS is able to assist the driver in EPS state or in hybrid state, as described in subchapter 6.2. In the event of a fault within one of those systems, the RASS offers an internal fallback level. The transition logic that the system follows is shown in Figure 6-10.

If an error occurs in the EPS ($f == 3$) during the EPS state or during the hybrid state, it is first deactivated to avoid unmotivated steering actions. The driver is then informed of the omitted EPS function and the HPS comes into effect as a power steering fallback level, in the same way as in the case of a failure of the EPS ($f == 1$). In this case, the HPS is not disabled in high velocity ranges for safety reasons, as it would have been in normal operation. If the active valve shows an error ($f == 4$), the erroneous system is also deactivated first. Just as in the event of an active valve failure ($f == 2$), the driver is informed and the RASS transfers to the fallback level. If the active valve is omitted, the EPS always represents a fallback level. If, apart from the active valve, the remaining HPS components are functional, the standard HPS also represents a fallback level during manual driving.

[142] The focus in this thesis is on AD3+.

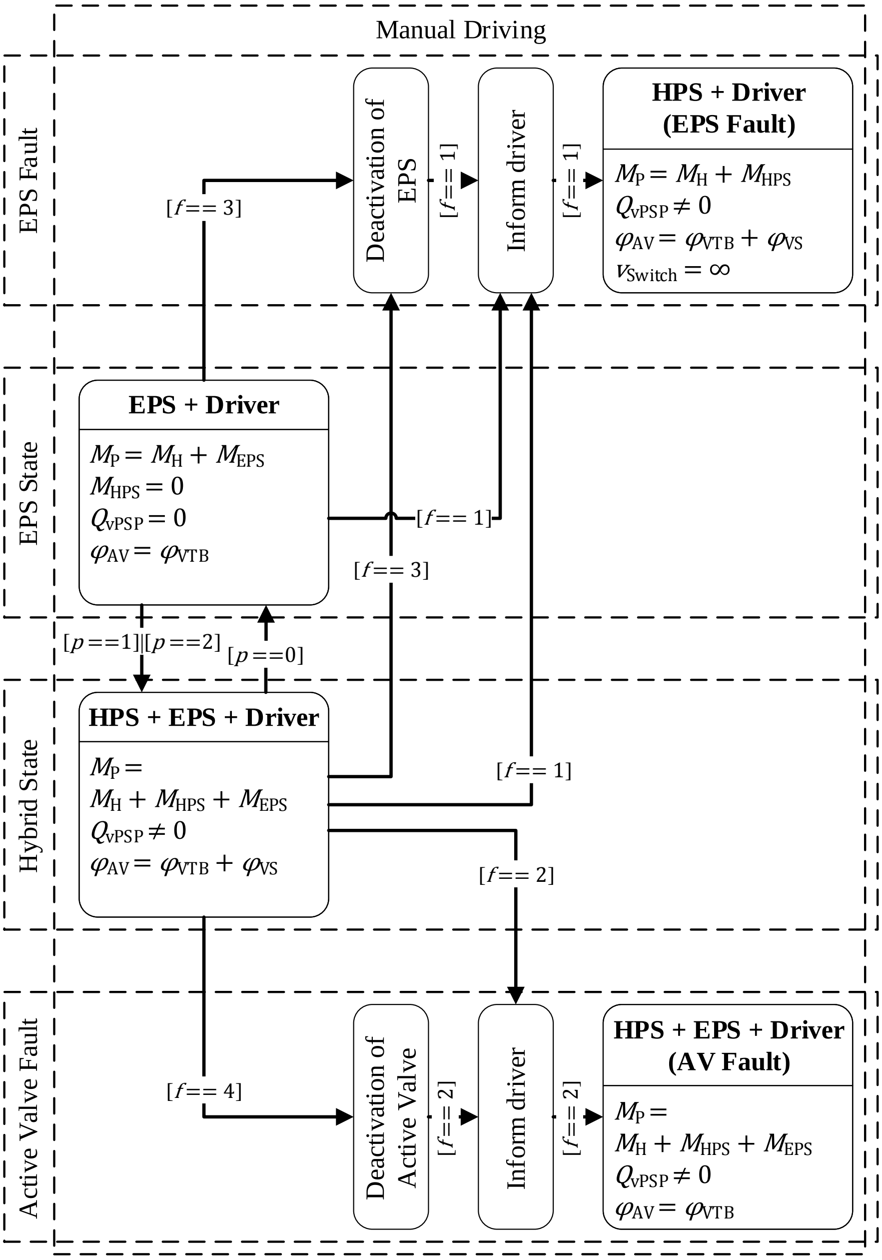

Figure 6-10: Transitions caused by faults during MD

During automated driving, the RASS represents the driver-independent steering function. The special feature of the new system is the presence of two driver-independent steering systems with the EPS and the AHPS, which enable AD3+. This redundancy is required for AD3+ because it cannot be assumed that the driver is available as an immediate fallback. In case of a fault of one of the active systems, the RASS has to offer an own fallback level. Figure 6-11 illustrate the transition logic in case of a fault during AD3+.

The procedure in the event of an error ($f == 3$, $f == 4$) or a failure ($f == 1$, $f == 2$) during automated driving is the same as for manual driving. The erroneous system is deactivated and the driver is informed in both cases. However, the further transition logic differs compared to manual driving.

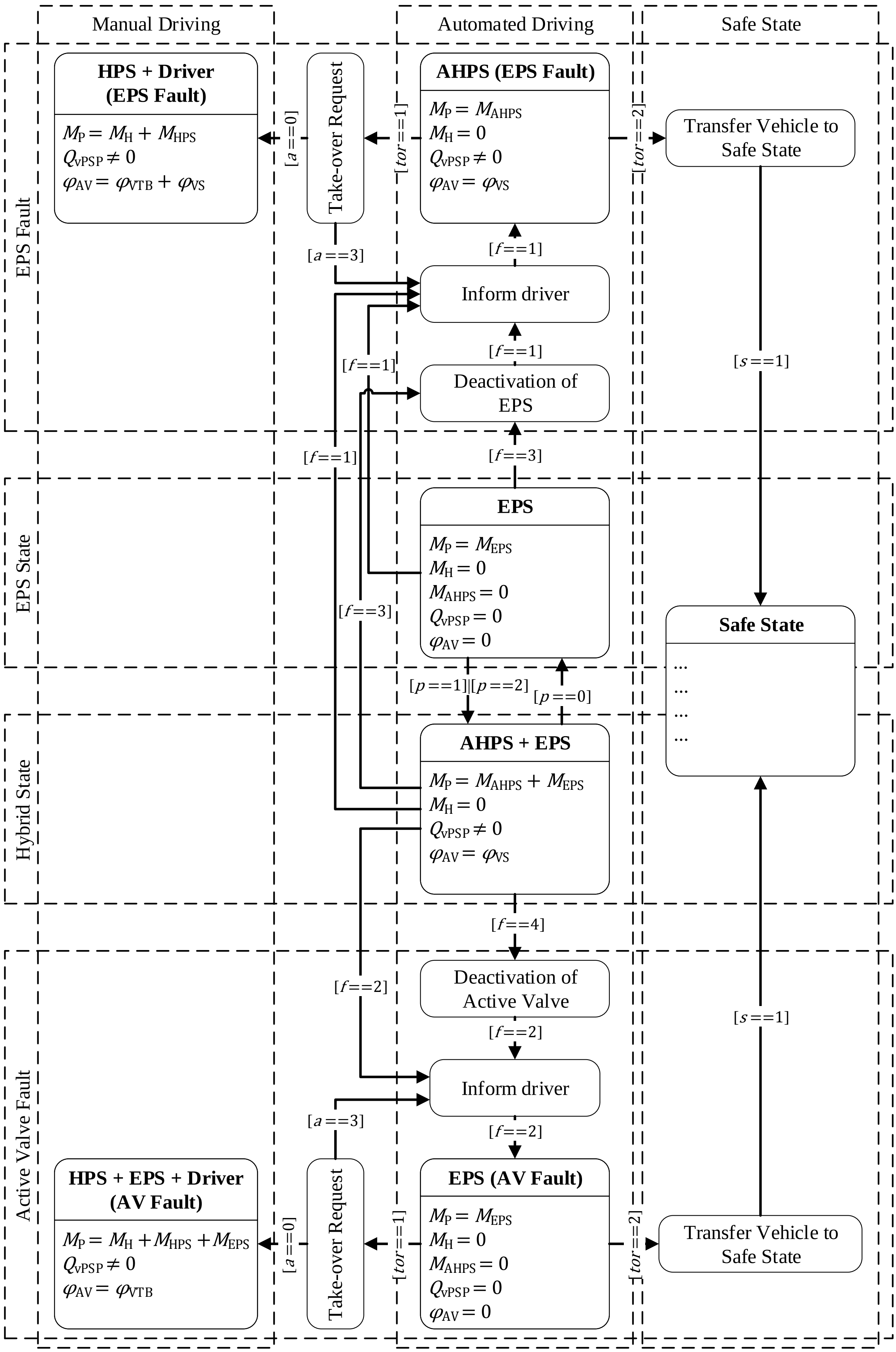

Figure 6-11: Transitions caused by faults during AD3+

Since the driver is not available as an immediate fallback during AD3+, the system transfers directly to the AHPS state as fallback if an EPS fault has been detected and to the EPS as fallback if the AHPS has a fault. In both cases, the RASS sends a take-over request to the driver ($tor == 1$), as the redundancy required for AD3+ is no longer given due to the omission of one active system.

The time required by the driver to take over the steering task safely varies between five seconds and more than ten seconds, as indicated in the literature.[143] This time should at least be given to the driver to take over the control. However, if the driver does not react after a longer period to be still defined ($tor == 2$), the RASS automatically transfers the vehicle into a defined safe state ($s == 1$). The safe states and the steering requirements the fallback systems have to fulfill for a safe transfer to them are defined in subchapter 4.4. If the driver reacts within this period, the RASS transfers to manual driving ($a == 0$) with the corresponding power steering state.

6.4.2 Implementation in System Design

The transitions caused by an error and resulting shutdown of the EPS are independent of the current automation state. Since the AHPS is already enabled during hybrid state, if the EPS is omitted only an adjustment of the sleeve of the active valve is necessary in order to compose the missing steering torque. During the EPS state, the omission of the EPS is not uncritical because here the AHPS is deactivated and the PSP does not stand-by with pumping hydraulic fluid. In order for the AHPS to serve as a fallback for the EPS, the PSP has to build up hydraulic volume flow and pressure first. Since modern PSP build up their maximum volume flow within 30 milliseconds, this start-up time affects the steering feel, but it is not critical for safety.[144]

An error or failure of the AHPS while the RASS is in EPS state does not cause a transition directly. However, the RASS informs the driver and transfers the vehicle into a safe state if the driver does not react to a take-over request. If the function of the AHPS is omitted due to an error or failure of the active valve during hybrid state, the transition to the fallback level depends on the valve position at the time of the failure. If the active valve fails in center position, the AHPS generates no torque and the EPS can easily jump in as a fallback. In contrast, if the active valve fails out of center position, the AHPS constantly produces steering torque. In this case, there are three different ways to safely transfer the RASS to its fallback level.

The first option is to implement a mechanism that returns the active valve into its center position in the event of a failure of the valve actuator without any external energy required. A torsion spring is a way of turning the valve back to the center position in the event of a

[143] Gold, C.; Huesmann, A.: Controllability of highly automated vehicle guidance (2017), p.50.

[144] Lauth, H. J. et al.: Needs based controllable pumps (2002), p.102.

fault, but it requires increased power during correct operation. This drawback excludes this option as a solution.

Reducing the volume flow of the PSP to zero is the second option to evade an off-centered failure of the active valve. If there is no hydraulic pressure and no volume flow, an open active valve causes no steering torque. Thus, only the EPS generates steering torque as fallback. Drawbacks of this option are that if the PSP is deactivated, the HPS can no longer be used by the driver in manual driving state and that it is only possible with an intact electrical power supply, as the PSP delivers full volume flow in the currentless state for safety reasons.

The third option is designed for a currentless state with an off-centered failed active valve. In this case, the active valve must be designed so that the driver is able to override the valve sleeve even if its torsion angle is maximum. This design is described in section 7.2.2.

6.5 Conclusion of State Transitions

The previous chapter describes the potential states of the RASS, which are mainly the power steering states, the automation states, and the fallback states. Several state indicators are defined, which are used by the operation strategy of the RASS to determine the current system state and to perform necessary state transitions.

The power steering can switch to three different states in order to increase the efficiency of the RASS. The tracking of the valve sleeve represents a novel solution to reduce the hydraulic losses even while the PSP is running. This solution requires a special valve design, which is described in the following chapter.

On the one hand, the transition from the manual driving state to the automated driving state is defined as hand-over and it is only conducted if the driver intends it and all conditions for a safe hand-over are fulfilled. On the other hand, two different transitions are distinguished for the switch from automated driving state to manual driving state. If the automated driving system exceeds its limits and is not able to control the vehicle safely without the driver, it sends a take-over request to the driver. In contrast, if the driver wants to override the automated system without its intention, it is called driver intervention. An approach with dynamic intervention torque and time thresholds distinguishes an intended driver intervention from an unintended driver interference.

In addition, the RASS is intended to be able to perform the power steering task in manual driving state as well as the automated steering task in the automated state in two different ways. Therefore, it provides fallback states for each potential fault in each combination of power steering state and automation state. The transition of the vehicle to a defined safe state and the driver's ability to override the automated steering system even in case of any potential fault are guaranteed by fallback logics of the RASS and the design of the crucial components. The latter are developed and described in the following.

7 Design Specifications of Components

In the last step of this thesis, the design specifications of the two active steering systems of the RASS and their components are developed based on all previous findings. Therefore, the requirements for the EPS including the electrical power supply are further specified first and, based on these, an initial constructive design is developed. In addition, the specifications for the AHPS are derived with a special focus on the active valve. This is specially designed for the tracking mechanism of the valve sleeve and the driver's ability to override the system.

7.1 Electric Power Steering

The electric power steering (EPS) consist of the electric motor (EM), which is connected at the lower end of the input shaft of the steering gear by an overlay gear (OG) and is controlled by the ECU. It also includes the redundant electric power supply, which supplies not only the EPS with electric power, but also all other components of the RASS.

The requirements for the electric power steering are derived directly from the redundancy requirements. The maximum steering torque that the EPS generates at the pitman arm ($M_{\mathrm{EPS,max}}$) is defined by the maximum required fallback torque (M_{RR}) in equation (7-1). The maximum power of the EPS ($P_{\mathrm{EPS,max}}$) is also determined by the maximum required fallback power (P_{RR}) in equation (7-2)

$$M_{\mathrm{EPS,max}} = M_{\mathrm{RR}} = 3{,}000 \text{ Nm} \tag{7-1}$$

$$P_{\mathrm{EPS,max}} = P_{\mathrm{RR}} = 600 \text{ W} \tag{7-2}$$

7.1.1 Redundant Electric Power Supply

The redundant electric power supply is responsible for a safe energy supply of all components of the RASS, specifically for providing the energy for the EPS.

The maximum required fallback steering power (P_{RR}) and the voltage of the vehicle power network (U) set the minimum required current from the generator in equation (7-3) to 25 A. This is required in addition to the current of a standard truck generator with 80 A. A generator with a maximum current of 110 A covers the additional demand.

$$I_{\mathrm{EPS}} = \frac{P_{\mathrm{RR}}}{U} = \frac{600 \text{ W}}{24 \text{ V}} = 25 \text{ A} \tag{7-3}$$

The batteries are designed redundantly, to ensure the power supply even in the event of a failure of the generator and one of the two batteries. The minimum capacity ($C_{\min}$) of 160 Ah of each battery is defined in equation (7-4) by the voltage of the power network and the maximum required fallback steering energy (E_{RR}) in Table 4-9 to guarantee enough electric power supply to reach a safe state in case of a fault of the generator and one battery.

$$C_{\min} = \frac{E_{\mathrm{RR}}}{U} = \frac{3{,}800\ \mathrm{J}}{24\ \mathrm{V}} = 158.\overline{3}\ \mathrm{Ah} \approx 160\ \mathrm{Ah} \tag{7-4}$$

The specifications of the components of the redundant electric power supply are listed in Table 7-1.

Table 7-1: Specifications of components of redundant electric power supply

ID	Component	Specifications
5	Redundant electric power supply	-
6	Generator	$U = 24$ V, $I_{max} = 110$ A
7	Battery isolator	-
8	Battery 1	$U = 24$ V, $Q_{\min} = 160$ Ah
9	Battery 2	$U = 24$ V, $Q_{\min} = 160$ Ah
10	Diode	-

7.1.2 Electric Motor

The voltage of the direct current vehicle power network (U) sets the voltage of the electric motor to 24 V. Due to the higher power density and better efficiency compared to asynchronous motors, a brushless direct current (BLDC) motor is used. This allows higher power and torque to be generated with the same dimensions. In addition, BLDC are very precise and dynamically controllable. [145]

A potential choice for an electric motor has a nominal power ($P_{\mathrm{EM,n}}$) of 701 W and a maximum torque ($M_{\mathrm{EM,max}}$) of 6.45 Nm.[146] Hence, the motor fulfills the fallback power requirement. The maximum torque of the EPS at the pitman arm ($M_{\mathrm{EPS,max}}$) depends on the ratio of the overlay gear (i_{EM2H}).

[145] Gaedke, A. et al.: Electric Power Steering Systems (2017), p.422.

[146] Data sheet of electric motor is attached in appendix A.4.

7.1.3 Torque Overlay Gear

The gear ratio of the overlay gear from the electric motor to the input shaft of the steering gear (i_{EM2H}) is set so that the maximum output torque of the electric motor ($M_{\mathrm{EM,max}}$) is sufficient to generate the required torque at the pitman arm ($M_{\mathrm{EPS,max}}$). Since the torque of the EM is also transferred by the RCB from the input shaft to the pitman arm, this ratio (i_{H2P}) has to be considered as well. Equation (7-5) calculates the required ratio of the overlay gear of the EPS:

$$i_{\mathrm{EM2H}} = \frac{M_{\mathrm{EPS,max}}}{M_{\mathrm{EM,max}} \cdot i_{\mathrm{H2P}}} = \frac{3{,}000\ \mathrm{Nm}}{6.45\ \mathrm{Nm} \cdot 24} \approx 19.4 \tag{7-5}$$

This high required transmission ratio is fulfilled, for example, by a two-stage planetary gear, as shown in Figure 7-1. The gear is flanged coaxially under the steering gear to the input shaft. The electric motor is also mounted coaxially under the steering gear to the housing of the steering gear via the housing of the gear unit. The right side of Figure 7-1 shows the installation position. Between the overlay gear and the steering gear is a flange in which the oil return of the HPS to the tank is integrated.

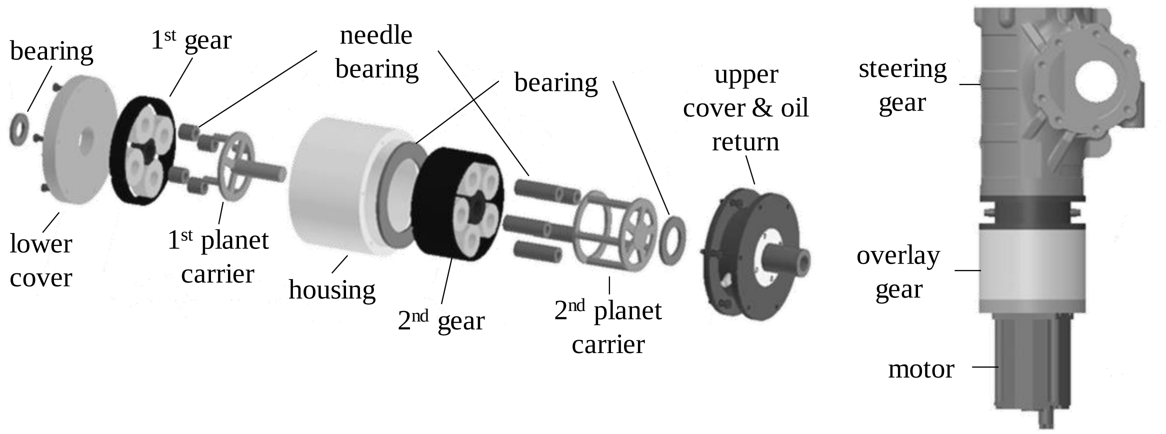

Figure 7-1: Exploded view of overlay gear (left), installation position of EPS (right)

Table 7-2 lists the defined specifications of the components of the electric power steering (EPS). An additional requirement of the overlay gear not specified here is the lowest possible backlash in order to guarantee backlash-free steering.

Table 7-2; Specifications of components of EPS

ID	Component	Specifications
11	Electric power steering (EPS)	$M_{\mathrm{EPS,max}} = 3{,}000$ Nm[147] $P_{\mathrm{EPS,max}} = 600$ W
12	Electric motor	$U = 24$ V, DC $M_{\mathrm{EM,max}} = 6.45$ Nm
13	Overlay gear	$i_{\mathrm{EM2H}} = 20$
14	Electronic control unit (ECU) 1	-

[147] Torque generated at pitman arm

7.2 Active Hydraulic Power Steering

Like standard HPS, the active hydraulic power steering (AHPS) also consists of a hydraulic cylinder integrated in the RCB steering gear, a hydraulic valve, a variable power steering pump (vPSP) and a tank for the hydraulic fluid. In contrast to the standard system, however, the AHPS is controllable actively and independently of the driver by the active valve. The ECU controls the active valve and the adjustable PSP.

The maximum steering torque the AHPS generates at the pitman arm ($M_{\mathrm{AHPS,max}}$) results from the difference between the maximum required operational torque (M_{OS}) and the maximum torque of the EPS at the pitman arm ($M_{\mathrm{EPS,max}}$) in equation (7-6). The same approach is used to determine the maximum power of the AHPS ($P_{\mathrm{AHPS,max}}$) in equation (7-7). The torque is referred to the pitman arm.

$$M_{\mathrm{AHPS,max}} = M_{\mathrm{OS}} - M_{\mathrm{EPS,max}} = 8{,}500\ \mathrm{Nm} - 3{,}000\ \mathrm{Nm} = 5{,}500\ \mathrm{Nm} \qquad (7\text{-}6)$$

$$P_{\mathrm{AHPS,max}} = P_{\mathrm{OS}} - P_{\mathrm{EPS,max}} = 3{,}500\ \mathrm{W} - 600\ \mathrm{W} = 2{,}900\ \mathrm{W} \qquad (7\text{-}7)$$

7.2.1 Hydraulic Piston & Adjustable Power Steering Pump

The hydraulic piston and the vPSP are standard HPS parts. However, by the summation of the output torques and power of the EPS and the AHPS, they can be designed smaller and less powerful compared to a standard HPS with the same operational requirements as the RASS. The specifications of both components depend strongly on each other. The size of the piston area has a direct influence on the vPSP requirements.

Based on the required maximum torque of the AHPS ($M_{\mathrm{AHPS,max}}$) and angular velocity at the pitman arm ($\dot{\delta}_{\mathrm{P,max}}$), the required hydraulic force at the piston ($F_{\mathrm{Piston,max}}$) and the piston's maximum velocity ($v_{\mathrm{Piston,max}}$) are calculated using the piston area (A_{Piston}) in equation (7-8) and (7-9). Based on these, the required maximum pressure ($p_{\mathrm{vPSP,max}}$) and maximum volume flow ($Q_{\mathrm{vPSP,max}}$) of the PSP are derived in equation (7-10) and (7-11).

$$F_{\mathrm{Piston,max}} = \frac{M_{\mathrm{AHPS,max}}}{i_{\mathrm{Piston2P}}} = \frac{5{,}500\ \mathrm{Nm}}{0.05\,\frac{\mathrm{m}}{\mathrm{rad}}} = 1.1 \cdot 10^{5}\ \mathrm{N} \qquad (7\text{-}8)$$

$$v_{\mathrm{Piston,max}} = \dot{\delta}_{\mathrm{P,max}} \cdot i_{\mathrm{Piston2P}} = 0.87\,\frac{\mathrm{rad}}{\mathrm{s}} \cdot 0.05\,\frac{\mathrm{m}}{\mathrm{rad}} = 4.36 \cdot 10^{-2}\,\frac{\mathrm{m}}{\mathrm{s}} \qquad (7\text{-}9)$$

$$p_{\mathrm{vPSP,max}} = \frac{F_{\mathrm{Piston,max}}}{A_{\mathrm{Piston}}} = \frac{1.1 \cdot 10^{5}\ \mathrm{Nm}}{7.3 \cdot 10^{-3}\mathrm{m}^2} = 150 \cdot 10^{5}\,\frac{\mathrm{N}}{\mathrm{m}^2} \qquad (7\text{-}10)$$

$$Q_{\mathrm{vPSP,max}} = v_{\mathrm{Piston,max}} \cdot A_{\mathrm{Piston}} = 4.36 \cdot 10^{-2}\,\frac{\mathrm{m}}{\mathrm{s}} \cdot 7.3 \cdot 10^{-3}\mathrm{m}^2 = 25\,\frac{\mathrm{l}}{\mathrm{min}} \qquad (7\text{-}11)$$

7.2.2 Active Valve

The central and new element of the AHPS is the active valve (AV). It enables the AHPS to control the steering torque independently of the driver input. Figure 7-2 illustrates a potential valve design, which is suitable for the described intended functionality. The components of this active valve design are listed in Table 7-3. The active valve is designed as a rotary slide valve with a valve sleeve adjustable by an integrated actuator.

Table 7-3: Components of the active valve

ID	Name
22	Input shaft
23	Output shaft
24	Torsion bar
25	Valve shaft
26	Valve sleeve
27	Angle overlay gear
28	Electric motor
29	Limiter rotation angle between valve shaft & valve sleeve
30	Limiter rotation angle between output shaft & valve sleeve
31	Limiter rotation angle between valve shaft & output shaft
32	Valve housing
33	Input throttle chamber A
34	Input throttle chamber B
35	Output throttle chamber A
36	Output throttle chamber B
37	Hydraulic Cylinder
38	Hydraulic Piston
39	Absolute Pressure Senor 1
40	Absolute Pressure Senor 2
41	Relative Pressure Senor
P	Hydraulic channel from power steering pump
A	Hydraulic channel to chamber A of hydraulic steering gear
B	Hydraulic channel to chamber B of hydraulic steering gear
T	Hydraulic channel to hydraulic tank

The input shaft (22), the output shaft (23), the torsion bar (24) and the valve shaft (25) are almost equivalent to those of a standard rotary slide valve. The valve shaft (25) is fixed to the input shaft (22), which in turn is mechanically connected to the output shaft (23) via the torsion bar (24) but can be rotated relative to it. This torsion angle of the torsion bar (φ_{VTB}) describes the torsion angle of the valve shaft relative to the output shaft and is induced by the driver's torque at the steering wheel, whereby the driver controls the valve. The new active adjustment mechanism consists of the adjustable valve sleeve (26), an angle overlay gear (27) and an electric motor (28) as actuator. The rotation of the valve sleeve (φ_{VS}) relative to the output shaft can be generated by the motor via the angle overlay gear. Thereby, the active valve is controllable independently of the driver by the electric motor, which itself is controlled by the ECU. In order to guarantee the ability of the driver to override the active valve, several mechanical stops are integrated in the valve design to limit the rotation angle between the valve shaft and the valve sleeve (29), the rotation angle between the output shaft and the valve sleeve (30) and the rotation angle between the valve shaft and the output shaft (31). The valve housing (32) is also specially designed considering the integration of the angle overlay gear and the electric motor on the one hand, and the sealing of the high hydraulic pressure at the adjustable valve on the other hand. The design of the housing cover of a standard HPS was used as a basis, in which the housing of a standard valve is integrated.

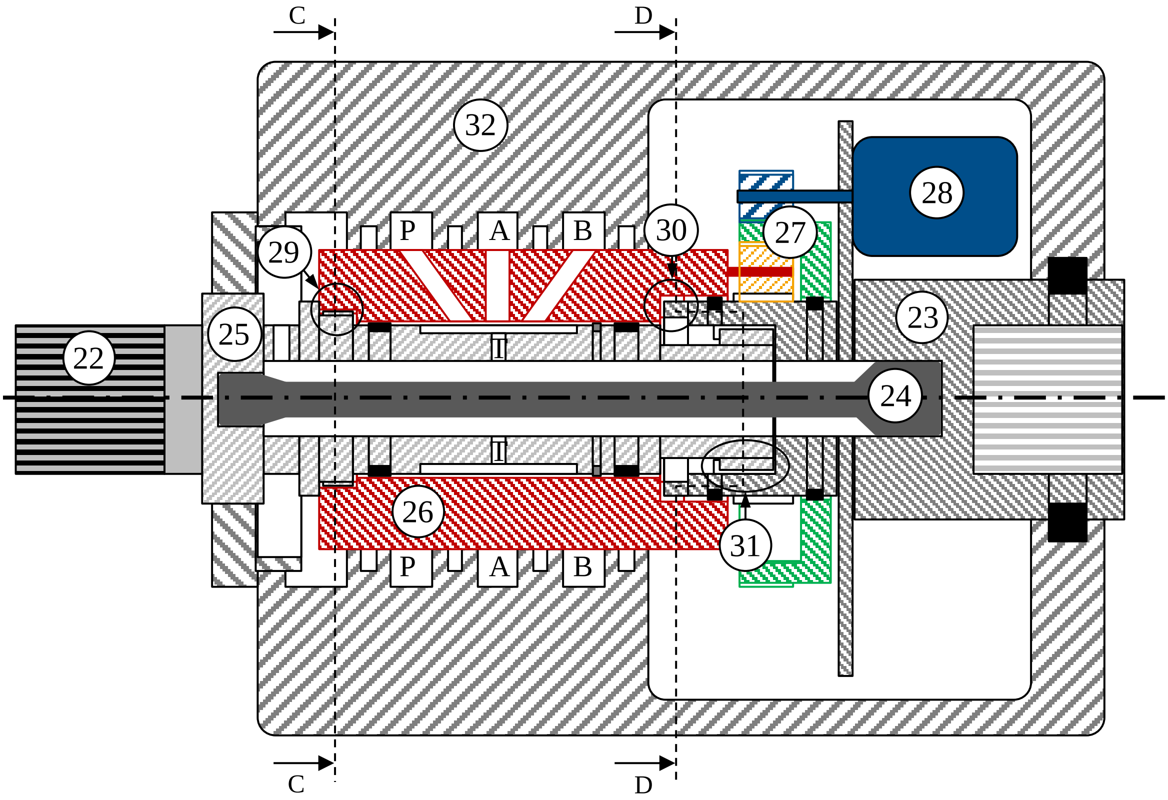

Figure 7-2: Conceptual design of active valve

In addition to the driver-independent control of the hydraulic steering torque, the active valve also functions in hybrid state to keep the valve in center position by tracking the valve sleeve as long as the EPS is sufficient for the required steering torque to reduce hydraulic losses. Furthermore, a mechanism is integrated into the active valve, which ensures that the driver can override the active valve and thus the hydraulic torque generated independently of the driver at any time. Both functions and mechanisms are described hereafter.

7.2.2.1 Tracking of Valve Sleeve

The tracking function of the RASS and its potential for energy saving[148] are already described in subchapter 6.2. The implementation and consequences on the valve design are explained here. Therefore, three different valve states are defined in Figure 7-3. The neutral state (a) in which no steering torque is generated, the stand-by hybrid state (b) in which the driver is only supported the EPS and the active hybrid state in which both EPS and AHPS support the driver are shown. The torque distribution and the valve angles are defined here for the hybrid state. The opening angle of the active valve (φ_{AV}) is defined in equation (7-12) by the torsion angle of the torsion bar (φ_{VTB}) and the rotation angle of the valve sleeve (φ_{VS}).

$$\varphi_{\mathrm{AV}} = \varphi_{\mathrm{VTB}} - \varphi_{\mathrm{VS}} \tag{7-12}$$

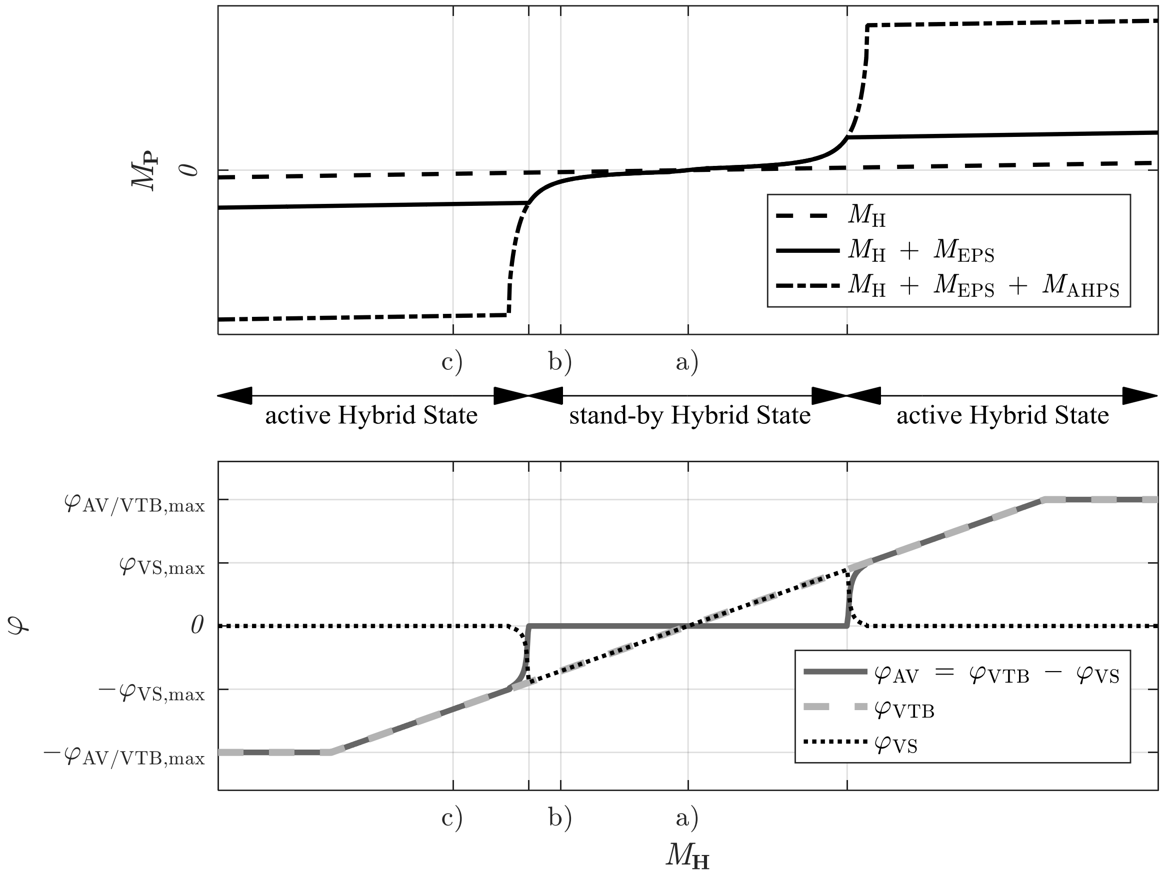

Figure 7-3: Hybrid state, a) neutral state, b) stand-by. c) active with steering wheel torque (M_{H}), steering torque at pitman arm (M_{P}) and valve angles (φ)

The valve states corresponding to the three states marked in Figure 7-3 are shown in Figure 7-4. All elements of the active valve are not rotated and in center position in the neutral state (a) and there is also no driver torque at the steering wheel, thus no steering torque is applied at the pitman arm. If the driver applies torque at the steering wheel, the torsion bar is twisted.

[148] This approach can save up to 712 W of hydraulic power losses at static torque. The proof is attached in the appendix A.5.

During stand-by hybrid state (b), the maximum EPS torque is sufficient to generate the required pitman arm torque. Hence, the valve sleeve is rotated by the same angle as the torsion bar to keep the valve effectively in center position and thus to reduce the hydraulic losses. The resulting torque at the pitman arm (M_P) is the sum of the driver's torque (M_H) and the torque of the EPS (M_{EPS}). The AHPS applies no torque (M_{AHPS}). If the RASS switches to active hybrid state (c), the valve sleeve is rotated back to neutral position, whereby the active valve is effectively opened. The AHPS supports the driver and the EPS with torque (M_{AHPS}).

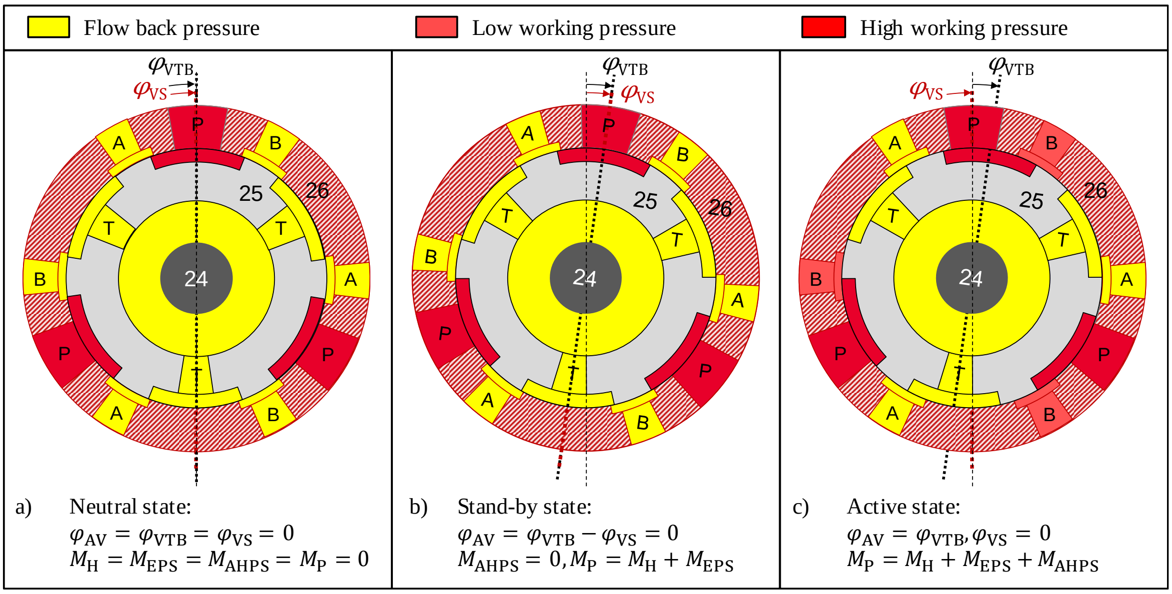

Figure 7-4: Valve states, a) neutral state, b) stand-by. c) active

By this mechanism implemented in the active valve, the hydraulic losses are reduced not only in EPS state, when the PSP is deactivated, but also in hybrid state with an active PSP as long as the EPS torque is still sufficient. However, as soon as the EPS cannot produce the required pitman arm torque, the AHPS generates the missing extended steering torque. The dynamics of the AHPS activation depends on the dynamics of the adjustment of the sleeve of the active valve.

7.2.2.2 Driver's Ability to Override the Active Valve

The second new function implemented in the new active valve design is the ability of the driver to override it. Therefore, different mechanical stops are integrated inside the active valve to limit the rotation angles between different parts of the active valve. Figure 7-5 illustrates the mechanical stops inside the valve design. The mechanical stop (29) limits the relative rotation angle between the valve shaft (25) and the valve sleeve (26) to a maximum, which corresponds to the maximum resulting overall active valve opening angle (φ_{AV}). This prevents the valve from over-twisting. The second mechanical stop (30) limits the relative rotation angle between the output shaft of the valve (25) and the valve sleeve (26) and thus the maximum rotation angle of the valve (φ_{AV}). In addition, the maximum rotation angle of the torsion bar relatively to the valve's output shaft (23) is limited by the third mechanical stop (31), which is also responsible for the transfer of the driver's steering wheel torque in case of a fully twisted torsion bar. This mechanical stop is also integrated into standard steering valves in order to limit the maximum torsion angle of the torsion bar and to transmit the additional torque applied by the driver directly to the valve outlet shaft.

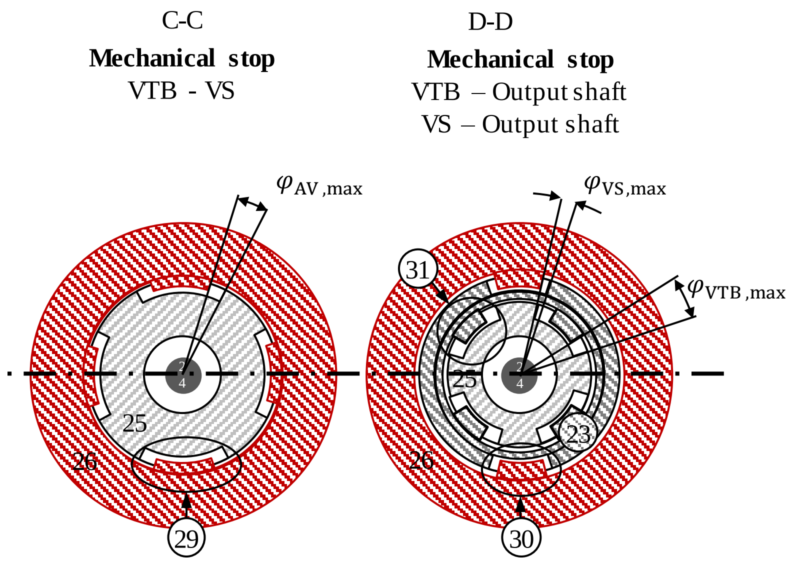

Figure 7-5: Mechanical stops of active valve[149]

These three mechanical stops have the function of ensuring that the driver can override the active valve, as explained in Figure 7-6. Five different points of the valve characteristic are used here to explain how the driver is able to counteract or to amplify the active valve. The corresponding valve states are shown in Figure 7-7 and Figure 7-8. The valve state (a) represents the initial state with a fully rotated valve sleeve ($\varphi_{VS} = \varphi_{VS,max}$) and a non rotated valve shaft ($\varphi_{VTB} = 0$). Equation (7-12) defines the resulting valve angle ($\varphi_{AV} = -\varphi_{VS,max}$).

If the driver applies torque on the steering wheel against the driver-independent torque, the valve shaft (25) is rotated in the same direction as the sleeve (26), reducing the valve opening until the valve results in the center position ($\varphi_{AV} = 0$) in state (b). The resulting torque of the AHPS decreases with a decreasing valve opening between state (a) and state (b) as shown in the upper part of Figure 7-6. If the driver continues to increase his torque at the steering wheel, the rotation angle of the valve shaft exceeds the angle of the valve sleeve ($\varphi_{VTB} > \varphi_{VS}$), which changes the sign of the generated torque at the pitman arm (M_P). The counteracting torque at the pitman arm increases until the valve shaft reaches its maximum rotation angle ($\varphi_{VTB,max}$) in state (c), limited by the mechanical stop (31). Due to the mechanical stop, a further increase in the driver's torque does not result in any further opening of the valve and thus does not increase the torque of the AHPS. The maximum rotation angle of the valve sleeve is limited by the mechanical stop to half of the maximum rotation angle

[149] Sectional view of Figure 7-2

of the valve shaft, as described in equation (7-13), to ensure the driver's ability to override the system.

$$\varphi_{\mathrm{AV,max}} = \varphi_{\mathrm{VTB,max}} = 2\varphi_{\mathrm{VS,max}} \tag{7-13}$$

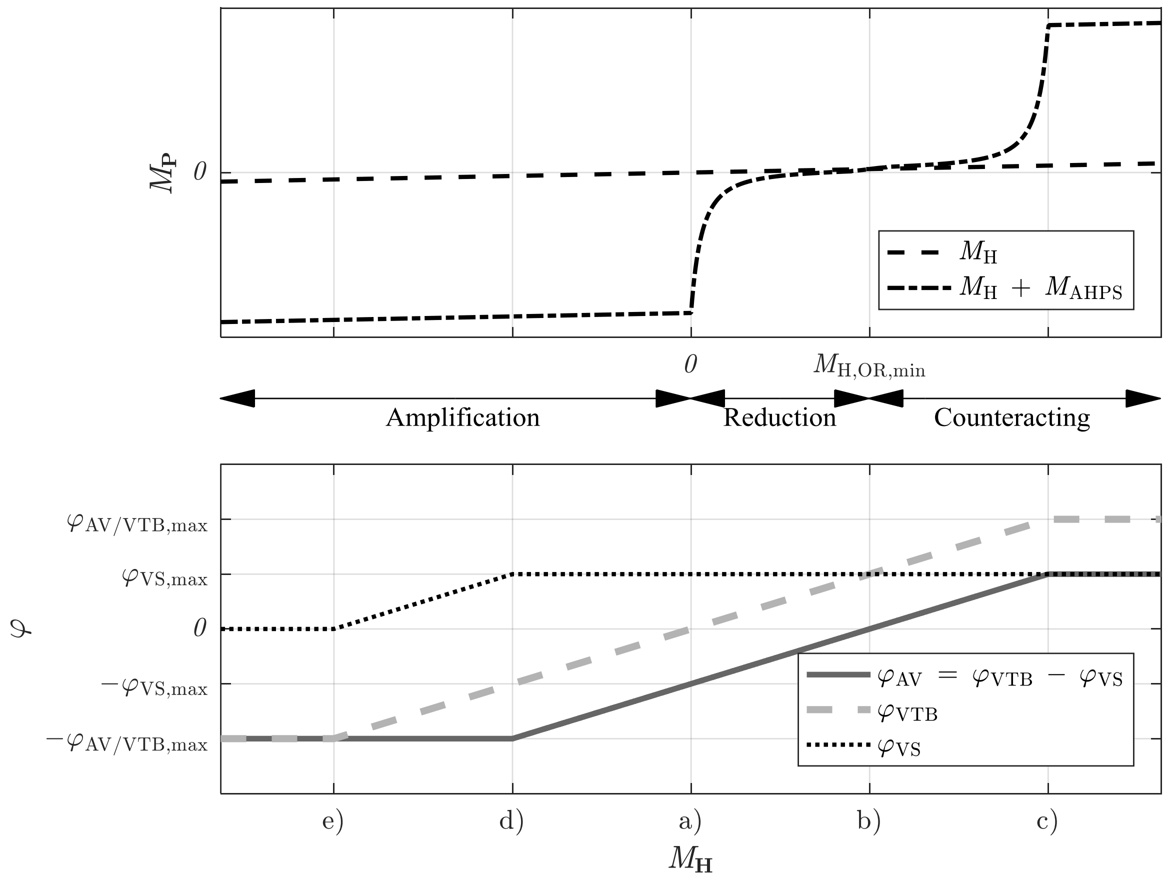

Figure 7-6: Overriding the active valve with steering wheel torque (M_{H}), steering torque at pitman arm (M_{P}) and valve angles (φ)

Thereby, the driver is able to open the active valve twice as much as the valve sleeve to override the automated system at any time. The stiffness of the torsion bar determines the torque ($M_{\mathrm{H,OR,min}}$) that the driver needs to apply at least to override the automated system. According to ECE R79, this torque must not exceed 12.5 Nm.

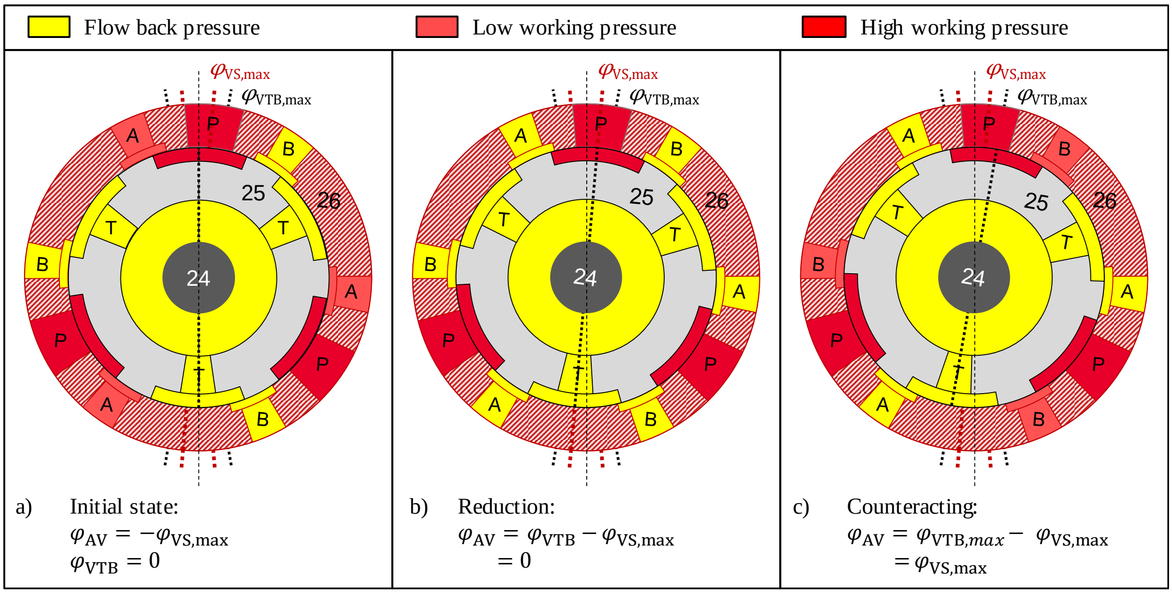

Figure 7-7: Valve states counteracting by the driver

The driver is not only able to counteract the AHPS, but also to amplify it. Figure 7-8 shows the same initial state (a) as before. By applying additional rectified torque at the steering wheel, the driver rotates the valve shaft in the opposite direction to the sleeve and thus opens the active valve further. This increases the torque of the AHPS until the valve hits the mechanical stop (29) between the valve shaft and the valve sleeve in state (d). The maximum opening angle of the active valve corresponds to the negative maximum rotation angle of the valve shaft ($\varphi_{\mathrm{AV}} = -\varphi_{\mathrm{AV,max}} = -\varphi_{\mathrm{VTB,max}}$). The torque generated by the AHPS is maximum, too. As the angle between the valve shaft and the valve sleeve is at its limit, a further increase of the driver's torque causes a further rotation of the valve shaft, but a turn-back of the valve sleeve to the center position ($\varphi_{\mathrm{VS}} = 0$) in state (e).

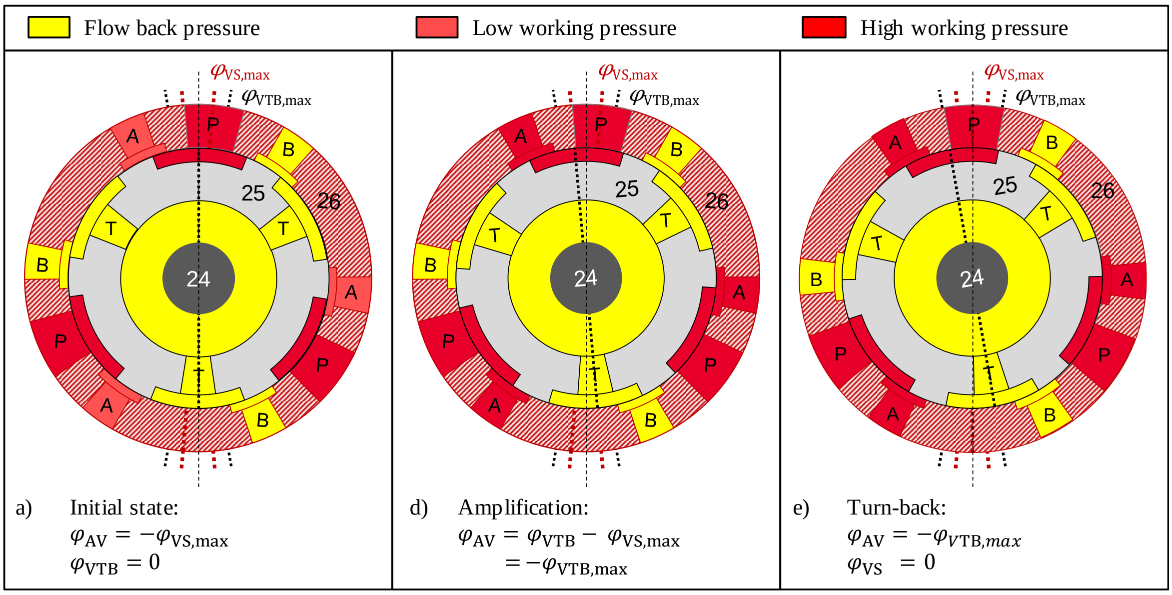

Figure 7-8: Valve states amplified by the driver

7.2.2.3 Sensing of the Valve State

For the implementation of the described functionality, it is necessary to sense the differential pressure (Δp_{HPS}) inside the hydraulic steering gear, the torsion angle of the valve torsion bar (φ_{VTB}), the steering wheel torque (M_{SW}), and/or the torsion angle of the active valve (φ_{AV}).

The differential pressure (Δp_{HPS}) is the most important parameter to know, because it is the output value of the active valve, which generates the hydraulic steering assistance. Two possible sensor setups are described in Figure 7-9.

Sensor setup 1 on the left side of Figure 7-9 uses two absolute pressure sensors (39, 40) and one differential pressure sensor (41). The pressure inside chamber A (p_{A}) is measured by the absolute pressure sensor (39) and the pressure inside chamber B (p_{B}) by sensor (40). The difference (Δp_{HPS}) described by (7-14) is measured with the differential pressure sensor (41). This sensor setup with three pressure sensors ensures the sensing of the crucial parameter (Δp_{HPS}) redundantly, whereby even in case of a fault of one sensor a correct sensing is guaranteed.

$$\Delta p_{\text{HPS}} = p_{\text{A}} - p_{\text{B}} \tag{7-14}$$

The other sensor setup 2 on the right side of Figure 7-9 requires one pressure sensor less. The differential pressure sensor (41) is the same as in the first setup. In contrast to setup 1, only one absolute pressure sensor (39) is used and measures the output pressure of the power steering pump (p_{PSP}) and thereby the pressure in the high-pressure chamber of the hydraulic cylinder. The direction of the differential pressure inside the cylinder is determined by the differential pressure sensor (41). This setup also guarantees the detection of a fault of one of the two sensors, but does not guarantee a safe operation after a sensor fault has occurred.

The torsion angle of the valve's torsion bar (φ_{VTB}) and the steering wheel torque (M_{SW}) are connected by the stiffness of the torsion bar (c_{VTB}) as described by (7-15). Therefore, it is sufficient to measure one of these two parameters and to calculate the other.

$$M_{\text{SW}} = c_{\text{VTB}} \cdot \varphi_{\text{VTB}} \tag{7-15}$$

Beside the measurement of the steering wheel torque (M_{SW}) with a standard torque sensor, an innovative possibility is to measure the angle of the torsion bar (φ_{VTB}) inside the valve and to calculate the steering wheel torque. Thereby, an additional torque sensor is not required for operation of the steering system, but it is mandatory to use it for redundancy.

Other sensor setups than these two are possible as well, but not described here.

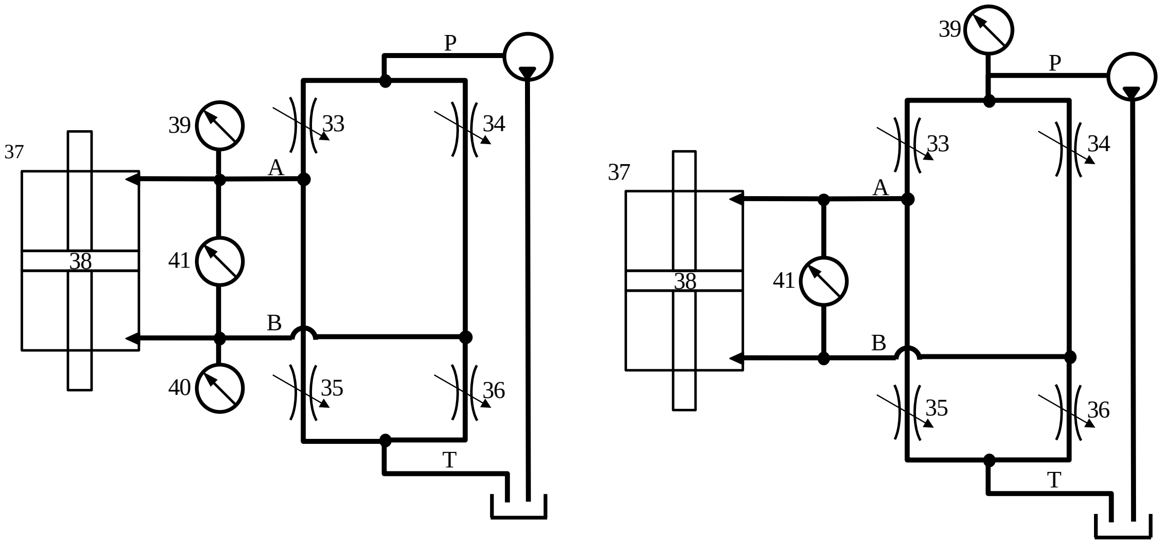

Figure 7-9: Valve scheme with sensor setup 1 (left) and sensor setup 2 (right)

The specifications of the components of the active valve, which are determined in this section, are summarized in Table 7-4.

Table 7-4: Specifications of components of AHPS

ID	Component	Specifications
3	Recirculating ball steering gear	$A_{\text{Piston}} = 7.3 \cdot 10^{-3}\text{m}^2$
15	Active hydraulic power steering (AHPS)	$M_{\text{HPS,max}} = 5{,}500$ Nm $P_{\text{HPS,max}} = 2{,}900$ W
16	Active steering valve with modulator	$M_{\text{OR}} \leq 12.5$ Nm
17	Variable power steering pump (vPSP)	$p_{\text{max}} = 150$ bar $Q_{\text{max}} = 25\,\frac{\text{l}}{\text{min}}$
18	Tank	-
19	ECU 2	-

7.3 Conclusion on Design Specifications

The designs of the two subsystems of the RASS, the electric power steering (EPS) and the active hydraulic power steering (AHPS), are specified in more detail in the preceding chapter. The redundancy requirements, determined in subchapter 4.4, are used to define the specifications of the EPS. The battery capacity of the redundant electric power supply fulfills the required fallback energy to be able to reach a safe state in any case. The maximum power and torque of the actuator of the EPS, which consist of the electric motor and a torque overlay gear, are defined based on the required fallback power and fallback torque. Thereby, in the event of a fault of the AHPS, the EPS is still able to steer the vehicle safely and to transfer it to a safe state.

The power and torque specifications of the AHPS are determined to fulfill the lack of power and torque between the operational requirements and the fallback requirements. Hence, the combination of the EPS and the AHPS fulfills the operational requirements, but both systems fulfill the fallback requirements on their own. This approach makes it possible to dimension the components of the AHPS smaller than the components of a comparable standard HPS and thus to save energy, weight, assembly space and costs. The active valve of the AHPS is specially designed for the described tracking mechanism and the ability of the driver to override the system. The tracking mechanism enables an arbitrary distribution of the required steering torque to the EPS and the AHPS, whereby the functionality of the RASS is significantly increased compared to the currently available truck steering systems. The transitions between the different states of the RASS are implemented using the new innovative functionality and the described operation strategies. In addition, this functionality enables to save energy by implementing the newly developed tracking mechanism. The most important feature of the active valve is the driver’s ability to override it at any time, which is guaranteed by the integration of different mechanical stops.

The design specifications determined in this chapter are the final step for the development of the steering system concept for AD3+. These specifications take into account all previously defined requirements of the RASS and implement the required fail-degraded functionality of the steering system for AD3+ of trucks for the first time. In addition, the innovative system architecture by combining EPS and AHPS enables the hydraulic components to be designed with significantly lower performance specifications than a comparable standard HPS, whereby the functionality is significantly increased.

8 Final Steering System Concept

The result of the approach of this thesis is a new innovative redundant active steering system for trucks with an increased functionality and efficiency compared to the currently available truck steering systems. The developed new system is called RASS and is described in an invention disclosure[150] which is being patented. Table 8-1 compares the functionality of the new RASS with those of the SoA of active steering systems for trucks that are described in subchapter 2.4.

The RASS, like all the SoA systems presented, has a variable steering torque assistance for the driver and is able to actively generate steering torque, i.e. independently of the driver. Hence, the RASS is suitable for AD2- of trucks just like the other systems. Unlike the other steering systems with hydraulic steering assistance, the EPS does not meet the high performance requirements of a truck steering system with the existing 24 V power network.

The new RASS enables an arbitrary distribution of the required steering torque between the two existing subsystems EPS and HPS through its innovative system architecture[150]. In addition, pure electric steering assistance is possible. This increased functionality is exploited by innovative operation strategies for driverless steering and increased system efficiency, which are documented in an invention disclosure[151] which is applied for patent.

HPS systems with an active valve only generate the steering power hydraulically and not by an electric drive. Their electric drive only controls the active valve to modulate the hydraulic steering torque, but does not generate steering torque by its own. Therefore, sole electric power steering is not possible with these systems.

In contrast, the existing hybrid power steering systems are able to generate steering torque solely electrically. However, they are not able to distribute the torque requirements to their HPS and EPS arbitrarily, because of their system architecture. In these systems, the EPS torque is transmitted through the steering valve, which means that the steering valve is not controllable independently of the EPS.

With the help of a variable PSP (vPSP), the new RASS as well as the other hydraulically assisted systems are able to reduce the hydraulic losses in driving situations with high velocities, e.g. highway driving. At high velocities, no high steering power is required, thus the hydraulic volume flow of the vPSP is reduced, which reduces the hydraulic losses. With the new RASS and the current hybrid steering systems, it is even possible to deactivate the pump in the high velocity range, in which the power steering is solely electric.

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[150] Herold, M. et al.: Redundant Active Steering System (2018).

[151] Herold, M.; Winner, H.: Method to control RASS (2019).

Table 8-1: Comparison of RASS with SoA

Functionality	EPS[152]	HPS with active valve	Hybrid power steering	RASS
Suitable for trucks with 24 V power network	No	Yes	Yes	Yes
Variable steering torque assistance	Yes	Yes	Yes	Yes
Active generation of steering torque	Yes	Yes	Yes	Yes
Arbitrary distribution of torque to EPS and HPS	-	No	No	Yes
Pure electric power steering	Yes	No	Yes	Yes
Reduction of hydraulic losses during high velocities	-	Yes	Yes	Yes
Reduction of hydraulic losses during low velocities	-	No	No	Yes
Redundancies	No	No	No	Diverse
Suitable for AD2-	Yes	Yes	Yes	Yes
Suitable for AD3+	**No**	**No**	**No**	**Yes**

What is new about the RASS, however, is that the hydraulic losses can also be reduced in the low velocity range. Although the EPS alone can cover most of the steering power even at low velocities, it is possible that higher steering power is required at any time. Therefore, it is not possible to reduce the volume flow of the PSP in the low velocity range, because otherwise the required steering power cannot be applied immediately. In order to still be able to steer purely electrically in this velocity range and thus reduce the hydraulic losses, the new functionality of the RASS to arbitrarily distribute the steering torque between the EPS and the HPS is used to develop an innovative operation strategy. By tracking the valve sleeve to the torsion of the torsion bar, the valve is kept in center position and thus no hydraulic torque is applied as long as the EPS torque is sufficient. While the valve is in center position, the differential pressure inside the hydraulic system is low, whereby the hydraulic losses are low as well. This tracking mechanism is recorded in an invention disclosure[153] which is currently being filed as a patent. This saves, for example, up to 712 W of hydraulic power loss at maximum static moment of the EPS. [154] This functionality is only enabled by the new RASS and not by the currently available steering systems.

The biggest advantage of the RASS over the existing systems is the presence of two active systems within the RASS. In combination with the RASS's redundant power supply and the redundant ECU, it fulfills the redundancy required for AD3+ and thus enabled the RASS for

[152] A steer by wire system is also a pure electric steering system and thus is not suitable for trucks with 24 V power network.

[153] Herold, M.; Winner, H.: Double Controlled Steering Valve (2018).

[154] The proof is attached in the appendix A.5.

AD3+.[155] These redundancies are all missing in the steering systems of the current SoA, whereby none of them is suitable for AD3+.

In addition to the functional advantages of the new system architecture of the RASS over the existing truck steering systems, the strategies developed for the various state changes also represent an innovation of this thesis.[156] The RASS has two different power steering states and two different automation states during fault-free operation. The strategy for the transfer of the power steering state selects the states based on the current vehicle speed and the required power steering torque. During the EPS state, the vPSP is deactivated to reduce the hydraulic losses. In stand-by hybrid state, the steering torque is still generated solely by the EPS, but the PSP is already activated to support the EPS immediately if the required steering torque exceeds its limits. In this case, the RASS switches to the active hybrid state and generates the steering torque combined by the EPS and AHPS.

Regardless of the power steering state, the RASS differs between three types of automation state transfer. The hand-over described the transition from the manual driving state to the automated driving state. When switching from automated driving state to manual driving state, a distinction is made between whether the change is intended by the system or by the driver. The former is called take-over, since the system requires the driver to take over the control as soon as the system exceeds its limits, for example. If the transfer from automated driving to manual driving is not desired by the system, but is caused by the driver's input at the steering wheel, this is referred to as driver intervention. In this case, a new approach with dynamic thresholds is developed in this thesis for the intervention torque and the minimum intervention time by the driver so that, on the one hand, unintended interferences of the driver on the automated steering system are prevented and, on the other hand, intentional interventions by the driver immediately lead to the transfer of control.[156]

Since the RASS must not only assist the driver or even take over the steering task completely in fault-free operation, but also in the event of a fault, strategies are also developed in this thesis on how the RASS switches to fallback states in the event of a fault of a subsystem. These strategies differ depending on whether the RASS is in the manual driving state or in automated driving state when a fault occurs. In manual driving state, the second system of the RASS compensates the fault of the other system and supports the driver as power steering fallback. The driver is informed about the switch to the fallback state. In the event of a fault of one active system during automated driving state, the other active system serves as fallback and continues generating steering torque independently of the driver. The system informs the driver of the switch into the fallback state and requests him to take over the control. If he does not comply with this request within a defined period of time, the RASS being in its fallback state transfers the vehicle into a defined safe state.[156]

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[155] Herold, M. et al.: Redundant Active Steering System (2018).

[156] Herold, M.; Winner, H.: Method to control RASS (2019).

The EPS and the AHPS subsystems of the RASS concept are elaborated in this thesis to such an extent that the specifications of the components, which are crucial for the function and the functional safety of the RASS, are defined. The energy capacity of the redundant electric power supply and the overall power of the EPS are determined based on the results of the analysis of the redundancy requirements. The power of the AHPS is determined by the lack of power of the EPS compared to the operational requirements. The active valve is designed in order to implement the described tracking functionality on the one hand, and, on the other hand, to ensure the ability of the driver to override the active valve at any time.[157]

[157] Herold, M.; Winner, H.: Double Controlled Steering Valve (2018).

9 Conclusion and Outlook

This thesis analyses the state of the art and research in the field of steering systems for trucks and identifies the significant gaps. Currently, there is no steering system that is suitable for highly automated driving (AD3+) of trucks. This is mainly due to the lack of redundancy in all known steering systems, which is required in the application of AD3+ to ensure the functional safety.

At the beginning of this thesis, driving tests and simulations based on these tests are used to determine operational and redundancy requirements for the steering system for AD3+ of trucks. The latter are the fallback requirements the steering system has to fulfill even in case of a fault inside the steering system to ensure a safe steering functionality. The investigations have shown that there is no need for a fully fail-operational steering system design for the AD3+ use case, but a fail-degraded design is sufficient. This means that in the event of a fault, the steering system does not have to meet all the operational requirements, but only the lower redundancy requirements.

This result has a decisive influence on the suitable system architectures. A redundant electric power steering system (EPS) is not able to fulfill those determined requirements using today's standard 24 V onboard power network of trucks. Since a redundant active hydraulic power steering system (HPS) does not fulfill current efficiency requirements, the only suitable steering system design for AD3+ considering the frame requirements of this thesis is a redundant hybrid steering system, which is a combination of an electric power steering system and an active hydraulic power steering system.

The performed functional safety analysis results in a functional safety concept with redundancies for the actuators, the sensors, the control and the power supply of steering system. The subsystems of the steering system are specified to fulfill all functional safety requirements and combined to realize a Redundant Active Steering System (RASS) which provides a diverse redundancy.

This thesis shows that a multiple of system states of current steering systems is required for the operation of such a RASS for AD3+ of trucks. As the RASS is a hybrid system, different power steering states are provided for its operation. Different automation states are required to enable manual driving as well as automated driving[158]. In the event of a system fault, the RASS suitable for AD3+ also has to provide fallback states to guarantee a safe operation without a take-over by the driver. In order to ensure that the RASS is always in the right state, this thesis develops an operation strategy that performs the state transitions based on logical rules.

[158] The RASS enables partially automated driving as well as highly automated driving, whereby only highly automated driving is considered within this thesis.

The active valve proves to be the crucial component for the implementation of the intended functionality as well as the functional safety and the guarantee that the driver is able to override the system.

The concept of a redundant active steering system (RASS) developed in this thesis serves as a basis for the further development of a steering system suitable for AD3+ of trucks. The next step of the development is the empirical proof of feasibility of the new innovative concept. This includes primarily the functional verification of the fallback strategies and of the arbitrary distribution of the required steering torque between the EPS and the AHPS. It has to be investigated to what extent the tracking mechanism increases the efficiency of the RASS in reality, whether the driver is always able to override the system by means of the integrated mechanical stops and how well the transitions of the power steering state or of the automation state work.

In addition to proving the functionality, there will be a great research focus on the steering feel. On the one hand, it is essential to prove the dynamics of the variable PSP to be sufficient for the switch from EPS state to hybrid state and the dynamics of the active valve for the transition between the stand-by hybrid state and the active hybrid state. On the other hand, there is research need on the transition process between manual driving and automated driving and reverse. Special focus should be on the driver intervention here. The exact values for the dynamic thresholds have to be evaluated in trial studies for example.

The necessity of a RASS for AD3+ trucks with a 24 V on-board power network, as developed in this thesis, is evident. As long as the 24 V vehicle power network remains the only on-board network for trucks, the RASS will be mandatory in all AD3+ trucks. As soon as, for example, the 48 V on-board power network replaces the 24 V network or supplements it as a second on-board network, the much less complex redundant EPS seems to be able to fulfill the operational requirements of trucks and thus will probably take precedence over the RASS.

A Appendix

A.1 Legal Requirements

A.1.1 Definitions from ECE R79

Table A-1: Definitions according to the ECE R79

Term	Description
Autonomous steering systems	System is controlled from off-board the vehicle and the driver will not necessarily be in primary control of the vehicle.
Automatically commanded steering function (ACSF)	Function within an electronic control system where actuation of the steering system results from automatic signals on-board the vehicle in order to assist the driver.
ACSF A	Function that operates at a speed no greater than 10 km/h to assist the driver, on demand, in low speed or parking maneuvering.
ACSF B1	Function that assists the driver in keeping the vehicle within the chosen lane, by influencing the lateral movement of the vehicle.
ACSF B2	Function which is initiated/activated by the driver and which keeps the vehicle within its lane by influencing the lateral movement of the vehicle for extended periods without further driver command/confirmation
ACSF C	Function which is initiated/activated by the driver and which can perform a single lateral maneuver (e.g. lane change) when commanded by the driver.
ACSF D	Function which is initiated/activated by the driver and which can indicate the possibility of a single lateral maneuver (e.g. lane change) but performs that function only following a confirmation by the driver.
ACSF E	Function which is initiated/activated by the driver and which can continuously determine the possibility of a maneuver (e.g. lane change) and complete these man oeuvres for extended periods without further driver command/confirmation.
Corrective steering function (CSF)	Control function within an electronic control system whereby, for a limited duration, changes to the steering angle may result from the automatic signals on-board the vehicle, in order: (a) To compensate a sudden, unexpected change in the side force of the vehicle; (b) To improve the vehicle stability (e.g. side wind, differing adhesion road conditions "µ-split") (c) To correct lane departure. (e.g. to avoid crossing lane markings, leaving the road).
Emergency steering function (ESF)	Control function which can automatically detect a potential collision and automatically activate the vehicle steering system for a limited duration, to steer the vehicle with the purpose of avoiding or mitigating a collision with another vehicle or obstacle.

A.2 Requirement Analysis

A.2.1 Vehicle Parameters

Table A-2: Vehicle Parameters for fully loaded MAN L2000 12.224

Parameter	Symbol	Value	Unit
Vehicle			
Mass	m_{V}	11,900	kg
Mass at front axle	$m_{\mathrm{V,fa}}$	4,340	kg
Wheelbase	ℓ_{W}	3.25	m
Distance from front axle to CoG	ℓ_{f}	2.06	m
Track width	w_{W}	1.94	M
Height of CoG	h_{CoG}	1.20	m
Yaw inertia	J_z	29,000	kgm^2
Wheel/Tire			
Tire dimensions	-	245/75 R17.5 134 L	-
Dynamic wheel radius	r_{dyn}	0.40	m
Tire pressure	p_{Tire}	$6 \cdot 10^5$	$\mathrm{N/m}^2$
Tire tread length	ℓ_{tread}	0.05	m
Tire bore stiffness	c_{Bore}	$14.65 \cdot 10^3$	Nm/rad
Maximum bore angle	$\delta_{\mathrm{Bore,max}}$	0.11	rad
Bore constant	b	3,307	Nm^2
Suspension			
Castor angle	τ	$6.1 \cdot 10^{-2}$	rad
Kingpin angle	σ	0.14	rad
Scrub radius	r_0	0.07	m
Castor offset	r_{τ}	$2.4 \cdot 10^{-2}$	m
Steering system			
Ratio between SW and W	i_{H2W}	$13.23 - 15.72$	-
Ratio between SW and P	i_{H2P}	$15.56 - 18.94$	-
Ratio between P and W	i_{p2W}	$0.82 - 0.93$	-
Inertia of steering system at pitman arm	$J_{\mathrm{eff,P}}$	15.00	kgm^2
Friction coefficient of steering	$\hat{T}_{\mathrm{F}}$	106	Nm
Damping coefficient of steering	d_{D}	1.38	Nm/rad

Table A-3: Vehicle parameters of fully loaded 26-t truck

Parameter	Symbol	Value	Unit
Vehicle			
Mass	m_{V}	$26{,}000$	kg
Mass at front axle	$m_{\mathrm{V,fa}}$	$8{,}500$	kg
Wheelbase	ℓ_{W}	$4.80 + 1.35$	m
Distance from front axle to CoG	ℓ_{f}	3.10	m
Track width	w_{W}	2.06	M
Height of CoG	h_{CoG}	1.40	m
Yaw inertia	J_z	$135{,}000$	kgm^2
Wheel/Tire			
Tire dimensions	-	385/55 R22.5 160 K	-
Dynamic wheel radius	r_{dyn}	0.48	m
Tire pressure	p_{Tire}	$8 \cdot 10^5$	$\mathrm{N/m}^2$
Tire tread length	ℓ_{tread}	0.07	m
Tire bore stiffness	c_{Bore}	$28.63 \cdot 10^3$	$\mathrm{Nm/rad}$
Maximum bore angle	$\delta_{\mathrm{Bore,max}}$	0.11	rad
Bore constant	b	$8{,}466$	Nm^2
Suspension			
Castor angle	τ	$6.1 \cdot 10^{-2}$	rad
Kingpin angle	σ	0.14	rad
Scrub radius	r_0	0.07	m
Castor offset	r_{τ}	$2.9 \cdot 10^{-2}$	m
Steering system			
Ratio between SW and W	i_{H2W}	$13.23 - 15.72$	-
Ratio between SW and P	i_{H2P}	$15.56 - 18.94$	-
Ratio between P and W	i_{p2W}	$0.82 - 0.93$	-
Inertia of steering system at pitman arm	$J_{\mathrm{eff,P}}$	15.00	kgm^2
Friction coefficient of steering	$\hat{T}_{\mathrm{F}}$	106	Nm
Damping coefficient of steering	d_{D}	1.38	$\mathrm{Nm/rad}$

A.2.2 Requirement List

Table A-4: Complete requirement list

	Requirement name	Values, data	Description
Frame requirements	Level of driving automation	AD3+	Steering system suitable for SAE level of automation 3 and higher
	Maximum steering wheel force/torque[159]	200 N/50 Nm (correct operation), 450 N/112.5 Nm (incorrect operation)	Maximum force required by the driver at the steering wheel;
	Override ability[160]/ Maxim force/torque for overriding[161]	50 N/12.5 Nm	Automated control of the steering is over-rideable by the driver at any time with limited effort
	Functional safety	ISO 26262 conforming	Development of mechatronic steering system according to latest version of ISO 26262
	Output interface	Pitman arm with push rod	Interface between output of steering sys-tem and truck's steering kinematic
	Electric power supply	24 V	Electric power supply by vehicle's power network
	Dimensions[162]	550 x 400 x 550 mm (length x height x width)	Dimensions of steering gearbox
	Hydraulic power supply[89]	max. 25 l/min max. 185 bar	Hydraulic power supply by power steer-ing pump
Operational re-quirements	Maximum output torque	8 500 Nm	Maximum torque at pitman arm during correct operation
	Maximum angular ve-locity	50 °/s	Maximum angular velocity at pitman arm during correct operation
	Maximum output power	3 500 W	Maximum power of overall steering sys-tem at pitman arm during correct opera-tion
Redundancy require-ments	Fallback torque	3 000 Nm	Maximum output torque at pitman arm in case of a partial failure of the steering system
	Fallback power	600 W	Maximum output power at pitman arm in case of a partial failure of the steering system
	Fallback energy	3 800 J	Available steering energy at pitman arm in case of a failure of the power supply

[159] Cp. United Nations: ECE R79 r4 (2018), p.30.

[160] Cp. United Nations: Convention on Road Traffic (1968), p.11.

[161] Cp. United Nations: ECE R79 r4 (2018), p.14.

[162] Data for MAN TGS 26:440

A.3 Functional Safety Analysis

A.3.1 Definition of Important terms of ISO 26262[163]

Table A-5: Definitions of ISO 26262

Term	**Description**
Automotive safety integrity level (ASIL)	“one of four levels to specify the item's or element's necessary ISO 26262 requirements and safety measures to apply for avoiding an unreasonable risk, with D representing the most stringent and A the least stringent level”[164]
Controllability	“ability to avoid a specified harm or damage through the timely reactions of the persons involved, possibly with support from external measures NOTE 1: Persons involved can include the driver, passengers or persons in the vicinity of the vehicle's exterior. NOTE 2: The parameter C in hazard analysis and risk assessment represents the potential for controllability.”
Degradation	“state or transition to a state of the item or element with reduced functionality, performance, or both”
Diversity	“different solutions satisfying the same requirement, with the goal of achieving independence NOTE 1: Diversity does not guarantee independency, but can deal with certain types of common cause failures. NOTE 2: Diversity can be a technical solution or a technical means to apply. NOTE 3: Diversity is one way to realize redundancy.”

[163] International Organization for Standardization: ISO 26262-12 (2018).

[164] In addition to the four ASIL, QM (quality management) denotes no requirement to comply with ISO 26262

Emergency operation	"operation mode of an item, for providing safety after the reaction to a fault until the transition to a safe state is achieved <u>NOTE 2</u>: When a safe state cannot be directly reached, or cannot be timely reached, or cannot be maintained after the detection of a fault, a safety mechanism can transition the item to emergency operation for providing safety until the transition to a safe state is achieved and maintained. <u>NOTE 4</u>: Degradation can be part of the concept for emergency operation."
Error	"discrepancy between a computed, observed or measured value or condition, and the true, specified or theoretically correct value or condition <u>NOTE 1:</u> An error can arise as a result of a fault within the system, or component being considered."
Exposure	"state of being in an operational situation that can be hazardous if coincident with the failure mode under analysis <u>NOTE 1:</u> The parameter "E" in hazard analysis and risk assessment represents the potential exposure to the operational situation"
External measure	"measure that is separate and distinct from the item which reduces or mitigates the risks resulting from the item"
Failure	"termination of an intended behavior of an element or an item due to a fault manifestation <u>NOTE:</u> Termination can be permanent or transient."
Fault	"abnormal condition that can cause an element or an item to fail"
Fault tolerance	"ability to deliver a specified functionality in the presence of one or more specified faults <u>NOTE 1:</u> Specified functionality can be intended functionality."
Functional safety	"absence of unreasonable risk due to hazards caused by malfunctioning behaviour of E/E systems"
Functional safety concept	"specification of the functional safety requirements, with associated information, their allocation to elements within the architecture, and their interaction necessary to achieve the safety goals"
Functional safety requirement	"specification of implementation-independent safety behavior, or implementation-independent safety measure, including its safety-related attributes <u>NOTE 1:</u> A functional safety requirement can be a safety requirement implemented by a safety-related E/E system, or by a safety-related system of other technologies, in order to achieve or maintain a safe state for the item taking into account a determined hazardous event. <u>NOTE 2:</u> The functional safety requirements might be specified independently of the technology used in the concept phase, of product development. <u>NOTE 3:</u> Safety-related attributes include information about ASIL."

Hazard	“potential source of harm”
Hazard analysis and risk assessment (HARA)	“method to identify and categorize hazardous events of items and to specify safety goals and ASILs related to the prevention or mitigation of the associated hazards in order to avoid unreasonable risk”
Item	“system or combination of systems, to which ISO 26262 is applied, that implements a function or part of a function at the vehicle level”
Redundancy	“existence of means in addition to the means that would be sufficient to perform a required function or to represent information <u>NOTE 1:</u> Redundancy is used in ISO 26262 with respect to achieving a safety goal or a specified safety requirement, or to representing safety-related information. <u>NOTE 2:</u> The redundancy could be implemented homogenously or within diversity. <u>EXAMPLE 1:</u> Duplicated functional components can be an instance of redundancy for the purpose of increasing availability or allowing fault detection. <u>EXAMPLE 2:</u> The addition of parity bits to data representing safety-related information provides redundancy for the purpose of allowing fault detection.”
Residual risk	“risk remaining after the deployment of safety measures”
Risk	“combination of the probability of occurrence of harm and the severity of that harm”
Safe state	“operating mode, in case of a failure, of an item without an unreasonable level of risk <u>NOTE 2:</u> While normal operation can be considered safe, the definition of safe state is only in the case of failure in the context of the ISO 26262 series of standards.”
Safety	“absence of unreasonable risk”
Safety architecture	“set of elements and their interaction to fulfill the safety requirements”
Safety goal	“top-level safety requirement as a result of the hazard analysis and risk assessment at the vehicle level <u>NOTE 1:</u> One safety goal can be related to several hazards, and several safety goals can be related to a single hazard.”
Safety measure	“activity or technical solution to avoid or control systematic failures and to detect or control random hardware failures, or mitigate their harmful effects <u>NOTE 1:</u> Safety measures include safety mechanisms.”

Safety mechanism	"technical solution implemented by E/E functions or elements, or by other technologies, to detect and mitigate or tolerate faults or control or avoid failures in order to maintain intended functionality or achieve or maintain a safe state <u>NOTE 1:</u> Safety mechanisms are implemented within the item to prevent faults from leading to single-point failures and to prevent faults from being latent faults. <u>NOTE 2:</u> The safety mechanism is either a) able to transition to, or maintain, the item in a safe state, or b) able to alert the driver such that the driver is expected to control the effect of the failure, as defined in the functional safety concept."
Severity	"estimate of the extent of harm to one or more individuals that can occur in a potentially hazardous event <u>NOTE:</u> The parameter "S" in hazard analysis and risk assessment represents the potential severity of harm."
T&B	"Trucks, Buses, trailers and semi-trailers"
Truck	"motor vehicle designed to transport goods, or equipment on-board the chassis <u>NOTE 1:</u> It may also tow a trailer."
T&B vehicle configuration	"technical characteristics of a T&B base vehicle and body builder equipment that do not change during operation <u>NOTE 1:</u> Changes may occur during rebuilding."
Unreasonable risk	"risk judged to be unacceptable in a certain context according to valid societal moral concepts"
Variance in T&B vehicle operation	"use of a T&B vehicle with different dynamic characteristics influenced by cargo or towing during the service life of the vehicle <u>EXAMPLE:</u> T&B with or without load, T&B with variations in load distribution, truck with or without trailer, tractor with or without semi-trailer (tractor solo)."

A.3.2 Classification of Hazardous events

Table A-6: Classes of severity[165a]

	Class			
	S0	**S1**	**S2**	**S3**
Description	No injuries	Light and moderate injuries	Severe and life-threatening injuries (survival probable)	Life-threatening injuries (survival uncertain), fatal injuries

Table A-7: Classes probability of exposure regarding operational situations[165a]

	Class				
	E0	**E1**	**E2**	**E3**	**E4**
Description	Incredible	Very low probability	Low probability	Medium probability	High probability

Table A-8: Classes of controllability[165b]

	Class			
	C0	**C1**	**C2**	**C3**
Description	Controllable in general	Simply controllable	Normally controllable	Difficult to control or uncontrollable

Table A-9: ASIL determination[165c]

Severity class	**Exposure class**	**Controllability class**		
		C1	**C2**	**C3**
S1	**E1**	QM	QM	QM
	E2	QM	QM	QM
	E3	QM	QM	A
	E4	QM	A	B
S2	**E1**	QM	QM	QM
	E2	QM	QM	A
	E3	QM	A	B
	E4	A	B	C
S3	**E1**	QM	QM	A
	E2	QM	A	B
	E3	A	B	C
	E4	B	C	D

[165] International Organization for Standardization: ISO 26262-3 (2018), p.8. (a) p.8 | (b) p.9 | (c) p.10

A.3.3 Fault Tree Analyses

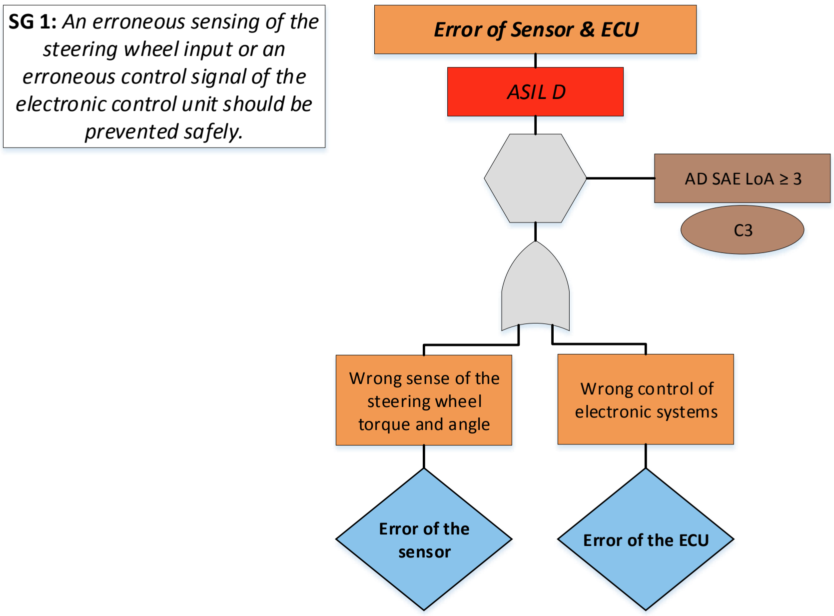

Figure A-1: Fault tree safety goal 1

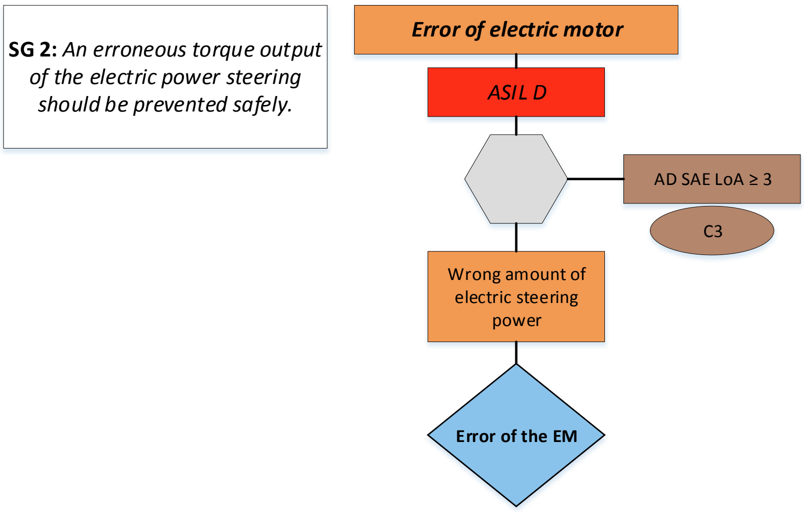

Figure A-2: Fault tree safety goal 2

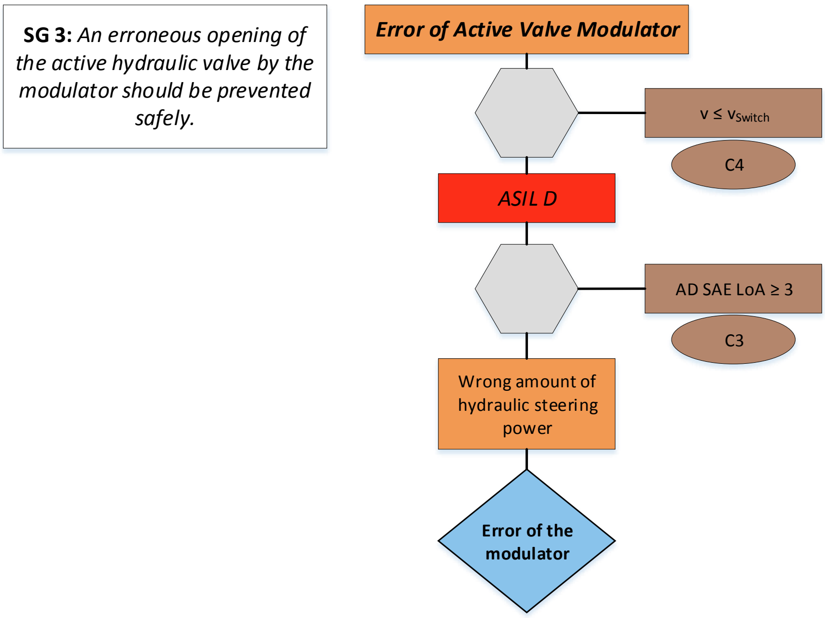

Figure A-3: Fault tree safety goal 3

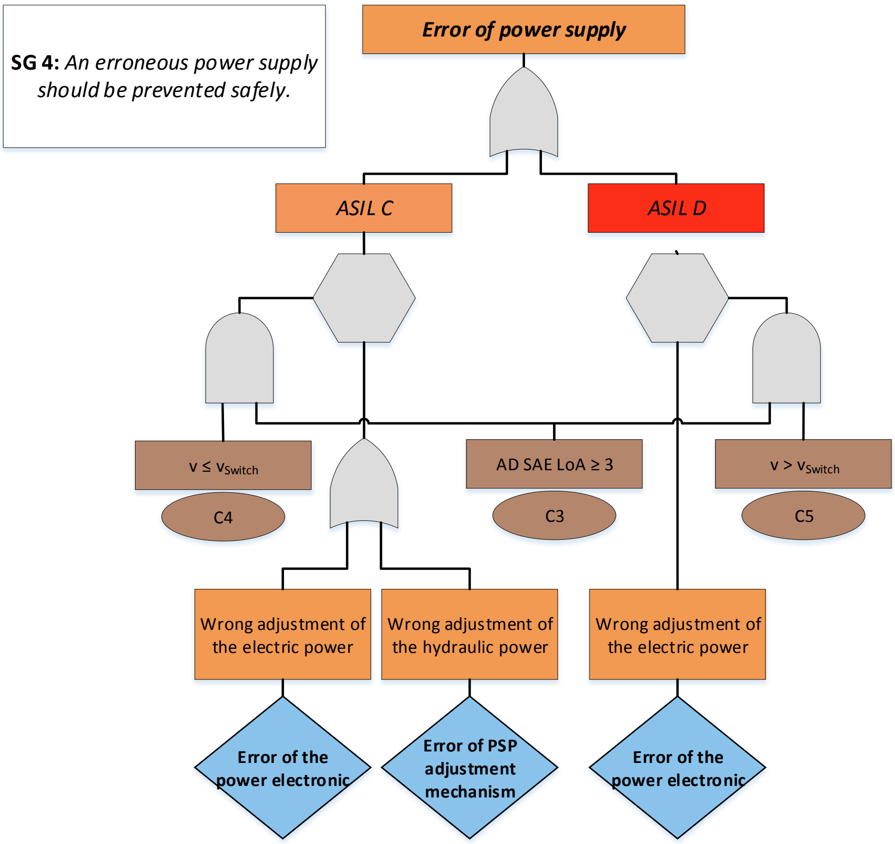

Figure A-4: Fault tree safety goal 4

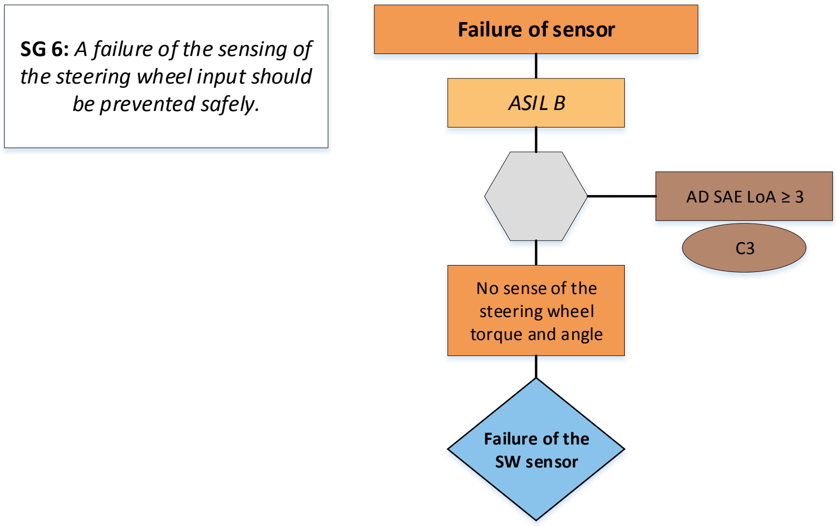

Figure A-5: Fault tree safety goal 6

A.3.4 Safety Concept Diagrams

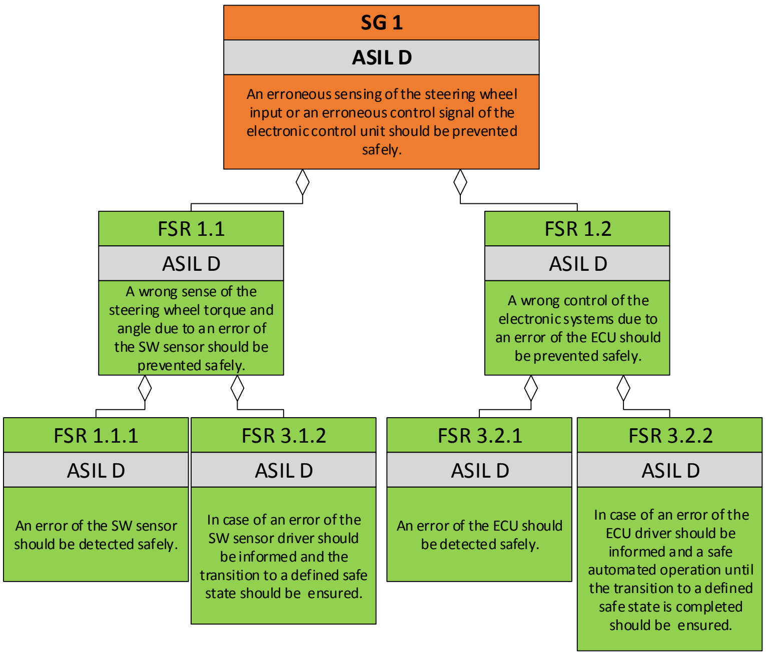

Figure A-6: Safety concept diagram safety goal 1

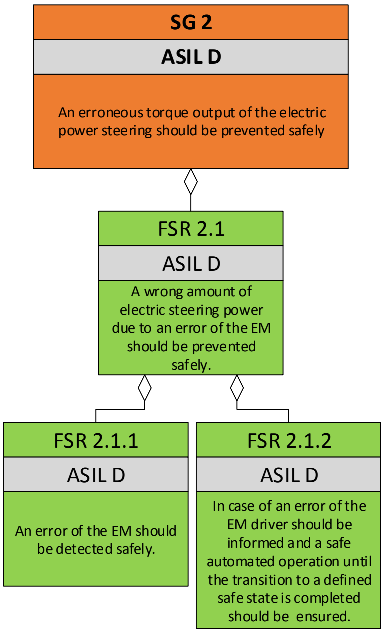

Figure A-7: Safety concept diagram safety goal 2

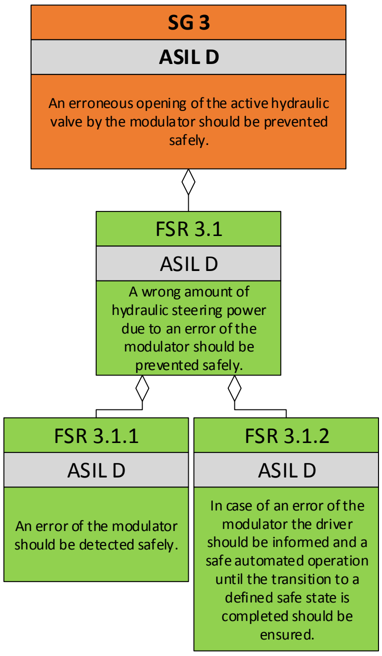

Figure A-8: Safety concept diagram safety goal 3

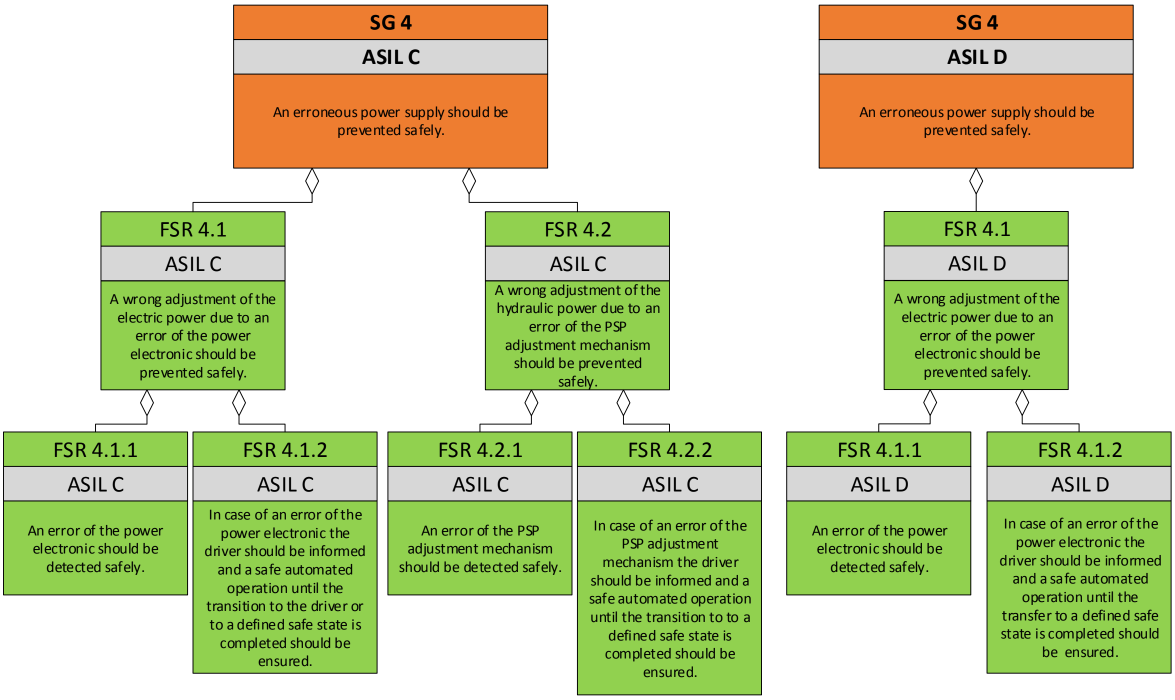

Figure A-9: Safety concept diagram safety goal 4

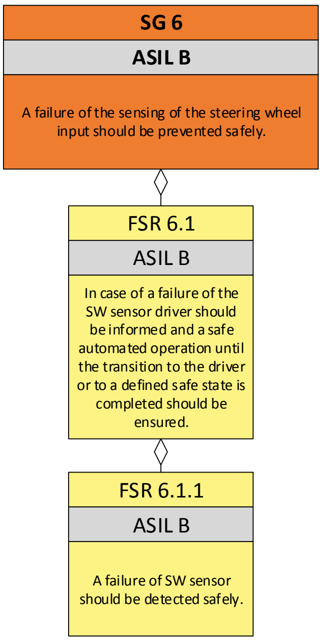

Figure A-10: Safety concept diagram safety goal 6

A.4 Data Sheet Electric Motor

Table A-10: Data sheet electric motor[166]

Specification	Symbol	Value	Unit
Nominal power	$P_{\mathrm{EM,n}}$	701	W
Nominal voltage	$U_{\mathrm{EM,n}}$	24	V
Nominal current	$I_{\mathrm{EM,n}}$	≤ 33.3	A
Nominal speed	$\dot{\delta}_{\mathrm{EM,n}}$	413	rad/s
Nominal torque	$M_{\mathrm{EM,n}}$	1.7	Nm
Starting current	$I_{\mathrm{EM,n}}$	485	A
Torque constant	K_{M}	0.057	Nm/A
Maximum power	$P_{\mathrm{EM,max}}$	1,819	W
Maximum speed	$\dot{\delta}_{\mathrm{EM,max}}$	995	rad/s
Maximum torque	$M_{\mathrm{EM,max}}$	6.45	Nm

[166] Dunkermotoren: BG95 (2018).

A.5 Proof of Reduction of Hydraulic Losses by Tracking

This chapter demonstrates by means of a simplified model calculation that the new tracking strategy developed in this thesis actually reduces hydraulic losses even when the PSP is running. For this purpose, the hydraulic power loss inside the valve in the center position during stand-by mode is calculated first. Then the HPS power is calculated that would be necessary to generate the same torque as the maximum EPS torque. The difference between these two powers minus the maximum power of the EPS results in the power loss that can be saved. It should of course be noted that the model is simplified and idealized. However, this shows that tracking is a promising strategy to increase the efficiency of the RASS.

The hydraulic steering valve is modeled as a Wheatstone bridge with four bridges (B1 – B4), as shown in Figure A-11. The bridges B1 and B4 or B2 and B3 are pairs and change their flow resistance in pairs. Due to the mechanical design of the steering valve, the cross section area of B1 (A_1) is always equal the cross section area of B4 (A_4). The same is valid for B2 and B3 ($A_2 = A_3$). The equations in this chapter describe the hydraulic behavior of the valve.[167]

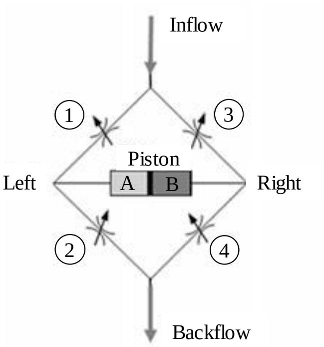

Figure A-11: Hydraulic diagram steering valve[167]

In the center position, the pressure inside the chamber A (p_{A}) is described with the pressure of the PSP (p_{PSP}) by (A-1)

$$p_{\mathrm{A}} = p_{\mathrm{PSP}} - \frac{Q_1^2}{B_1^2} \tag{A-1}$$

and with the pressure of the tank (p_0) by (A-2),

$$p_{\mathrm{A}} = p_0 + \frac{Q_2^2}{B_2^2} \tag{A-2}$$

[167] Hullmann, J. et al.: Mechanical and Hydraulic Gears (2017), pp. 310–313.

Whereas the bridge is defined by (A-3) with the valve flow constant (k_{Valve}), the associated cross section area (A_{ii}) and the desnity of the hydraulic oil (ρ_{Oil}).

$$B_i = k_{\text{Valve}} \cdot A_{ii} \cdot \sqrt{\frac{2}{\rho_{\text{Oil}}}} \tag{A-3}$$

All bridges are equally open in the center position, resulting in (A-4), (A-5) and (A-6).

$$B_1 = B_2 = B_3 = B_4 \tag{A-4}$$

$$A_1 = A_2 = A_3 = A_4 \tag{A-5}$$

$$Q_1 = Q_2 = Q_3 = Q_4 = \frac{Q_{\text{PSP}}}{2} \tag{A-6}$$

The pressure difference (Δp) between the PSP and the tank is defined by (A-7) based on the previous equations.

$$\Delta p = p_{\text{PSP}} - p_0 = \frac{Q_1^2}{B_1^2} + \frac{Q_2^2}{B_2^2} = \frac{Q_{\text{PSP}}^2}{4B_1^2} \tag{A-7}$$

Using (A-7) and the volume flow of the PSP (Q_{PSP}), the power that is lost inside the valve in center position during stand-by hybrid mode ($P_{\text{loss,sb}}$) is calculated by (A-8).

$$\mathrm{P}_{\text{loss,sb}} = \Delta p \cdot Q_{\text{PSP}} = \frac{\rho_{\text{Oil}}}{4k_{\text{Valve}}^2 \cdot A_1^2} \cdot Q_{\text{PSP}}^3 \tag{A-8}$$

With the following parameters[168] of the PSP and the valve, $P_{\text{loss,sb}}$ is 0.65 W.

$$\rho_{\text{Oil}} = 0.839\,\frac{\text{kg}}{\text{m}^3}$$

$$k_{\text{Valve}} = 0.8$$

$$A_1 = 5.39 \cdot 10^{-6}\text{m}^2$$

$$Q_{\text{PSP}} = 4.17 \cdot 10^{-4}\,\frac{\text{m}^3}{\text{s}}$$

$$P_{\text{loss,sb}} = 0.65\ \text{W}$$

The required hydraulic power ($P_{\text{HPS,active}}$) to reach the same static torque as the maximum torque of the EPS ($M_{\text{EPS,max}}$) is set by pressure difference (Δp) inside the two chambers of the hydraulic piston and the volume flow of the PSP (Q_{PSP}) in (A-9). The pressure difference is defined by the piston area (A_{Piston}) and the ratio from the piston to the pitman arm (i_{Piston2P}).

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[168] Data from steering supplier.

$$P_{\mathrm{HPS,active}} = \Delta p \cdot Q_{\mathrm{PSP}} = \frac{F_{\mathrm{HPS}} \cdot Q_{\mathrm{PSP}}}{A_{\mathrm{Piston}}} = \frac{M_{\mathrm{EPS,max}} \cdot Q_{\mathrm{PSP}}}{i_{\mathrm{Piston2P}} \cdot A_{\mathrm{Piston}}} \tag{A-9}$$

With the following parameters, the $P_{\mathrm{HPS,active}}$ is calculated to 3,425 W.

$$M_{\mathrm{EPS,max}} = 3000\ \mathrm{Nm}$$

$$i_{\mathrm{Piston2P}} = 0.05\,\frac{\mathrm{m}}{\mathrm{rad}}$$

$$A_{\mathrm{Piston}} = 7.3 \cdot 10^{-3}\mathrm{m}^2$$

$$Q_{\mathrm{PSP}} = 4.17 \cdot 10^{-4}\,\frac{\mathrm{m}^3}{\mathrm{s}}$$

$$P_{\mathrm{HPS,active}} = 3{,}425\ \mathrm{W}$$

The maximum power of the EPS ($P_{\mathrm{EPS},max}$) at maximum static torque is calculated from the maximum current ($I_{\mathrm{EM},max}$) and the voltage (U) in equation (A-11). The maximum current ($I_{\mathrm{EM},max}$) is defined by the maximum static torque of the electric motor ($M_{\mathrm{EM,max}}$) and its torque constant (K_{M}) by equation (A-10).

$$I_{\mathrm{EM},max} = \frac{M_{\mathrm{EM,max}}}{K_{\mathrm{M}}} = \frac{6.45\ \mathrm{Nm}}{0.057\ \mathrm{Nm/A}} \approx 113\ \mathrm{A} \tag{A-10}$$

$$P_{\mathrm{EPS},max} = U \cdot I_{\mathrm{EM},max} = 24\ \mathrm{V} \cdot 113\ \mathrm{A} = 2{,}712\ \mathrm{W} \tag{A-11}$$

With the maximum power output of the EPS $P_{\mathrm{EPS,max}}$ a maximum power loss of 712 W can be saved according to (A-12) by using tracking.

$$P_{\mathrm{Save},max} = P_{\mathrm{HPS,active}} - P_{\mathrm{loss,sb}} - P_{\mathrm{EPS,max}} \approx 712\ \mathrm{W} \tag{A-12}$$

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