

THESIS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY

Safe and energy-efficient control allocation
for multi-unit heavy vehicles

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An electric tractor with two electric semitrailers, generated using Google Gemini 3 Pro Image.

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*To the children of Gaza,
who will remain children forever*

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Abstract

Electrification, increased variation, and the complexity of multi-unit vehicle combinations have created a need for novel control allocation strategies. Traditional vehicle combinations are propelled solely by the towing vehicle—trucks or tractors—with braking forces distributed in proportion to the axle loads, which is commonly the safest approach. Conversely, modern combinations may be propelled by multiple units, with regenerative braking performed by various electrified units, not necessarily in proportion to the axle loads. An optimal approach for allocating propulsion and braking forces among vehicle units is power loss minimization. This thesis presents a hierarchical, distributed control architecture that effectively manages over-actuation by decoupling upper-level combination control from lower-level unit control. This distributed framework minimizes power losses across multiple units, demonstrating that selectively disengaging electric motors reduces overall energy consumption, while integrating state-of-charge control ensures equal depletion rates among batteries across units in an energy-efficient manner.

From an energy-efficiency perspective, concentrating propulsion or regenerative braking on a single vehicle unit may be optimal. However, such allocations can compromise safety by inducing motion instabilities, such as jackknifing and trailer swing. Consequently, ensuring combination stability while allocating unit forces efficiently is essential. This thesis introduces safe operating envelopes for vehicle combinations in both the force and slip domains. Using single-track, two-track, and high-fidelity models, along with phase-portrait analyses, this work systematically establishes adaptive slip limits and safe force boundaries. These envelopes are tested in simulations and real-world vehicle tests, confirming their effectiveness in preventing yaw instabilities. Furthermore, predictive safety barriers—ranging from advanced Model Predictive Control formulations to simpler proportional-heuristic controllers—are developed and tested in simulations, demonstrating promising performance in proactively avoiding yaw instabilities.

Finally, this study compares the evaluated controllers and details the envisioned functional architecture and signal interfaces, establishing a robust foun-

dition for the safe and efficient operation of future multi-unit electrified heavy vehicles.

Keywords: Heavy vehicles, electric vehicles, electric trailers, control allocation, stability, jackknifing, trailer swing, power loss minimization.

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List of Publications

This thesis is based on the work contained in the following papers:

Paper A

U. Erdinc, M. Jonasson, M. S. Kati, B. Jacobson, J. Fredriksson, and L. Laine, “Safe operating envelope based on a single-track model for yaw instability avoidance of articulated heavy vehicles,” *Vehicle Syst. Dyn.*, vol. 62, no. 8, pp. 2138–2161, Aug. 2024, doi: 10.1080/00423114.2023.2276767.

Paper B

U. Erdinc, M. Jonasson, M. S. Kati, B. Jacobson, J. Fredriksson, and L. Laine, “Validation of high-fidelity simulation-based safe operating envelopes for articulated heavy vehicles using real test data,” *Vehicle Syst. Dyn.*, vol. 62, no. 10, pp. 2567–2589, Oct. 2024, doi: 10.1080/00423114.2023.2296595.

Paper C

U. Erdinc, M. Jonasson, M. S. Kati, B. Jacobson, J. Fredriksson, and L. Laine. (Nov. 2023). Modelling of articulated heavy vehicles for computation of accurate safe operating envelope for yaw stability. Presented at 17th Int. Symp. on Heavy Vehicle Transp. & Technol., Brisbane, Australia. [Online]. Available: hvtforum.org/wp-content/uploads/2023/12/HVTT17_paper_r_3004_Erdinc.pdf

Paper D

U. Erdinc, M. Jonasson, M. S. Kati, L. Laine, B. Jacobson, and J. Fredriksson, “Yaw stability control of vehicles using a slip polytope validated with real tests,” in *Proc. 16th Int. Symp. on Adv. Vehicle Control*, Milan, Italy, Sep. 2024, pp. 130–136, doi: 10.1007/978-3-031-70392-8_19.

Paper E

A. Bharganwar, M. V. Ramasamy, U. Erdinc, and M. Jonasson, “Safe operating envelope generation using phase portraits for a tractor-semitrailer combination,” presented at the 17th Int. Symp. on Adv. Vehicle Control, Tsukuba, Japan, Sep. 2026.

Paper F

U. Erdinc, M. Jonasson, M. S. Kati, B. Jacobson, L. Laine, and J. Fredriksson, “Distributed powertrain and battery coordination for multi-unit electrified vehicle combinations,” presented at the 17th Int. Symp. on Adv. Vehicle Control, Tsukuba, Japan, Sep. 2026.

Paper G

U. Erdinc, M. Jonasson, M. S. Kati, L. Laine, B. Jacobson, and J. Fredriksson, “Experimental validation of yaw stability control strategies for articulated vehicle combinations,” in *Proc. 35th IEEE Intell. Vehicles Symp.*, Jeju, South Korea, Jun. 2024, doi: 10.1109/IV55156.2024.10588717.

Paper H

U. Erdinc, M. Jonasson, M. S. Kati, B. Jacobson, L. Laine, and J. Fredriksson, “Safe predictive control allocation for heavy vehicle combinations,” submitted for publication.

The author of this thesis conducted the modeling, simulation, testing, analysis, and initial drafting for all appended papers, with the exception of Paper E. Co-authors provided valuable discussions, reviews, and revisions throughout the research process. For Paper E, the first two authors performed the modeling, simulation, analysis, and initial drafting, while the author of this thesis contributed through guidance, review, and revision.

The following publications by the author are relevant to the topic but not included in the thesis:

A. Hansson, E. Andersson, L. Laine, M. S. Kati, U. Erdinc, and M. Jonasson, “Safe operating envelope for limiting actuation of electric trailer in tractor-semitrailer combination,” in *Proc. 2022 IEEE 25th Int. Conf. Intell. Transp. Syst. (ITSC)*, Macau, China, Oct. 2022, pp. 3886–3893, doi: 10.1109/ITSC55140.2022.9922094.

S. van Dam, L. Wisell, K. Shingade, M. Kieu, U. Erdinc, M. S. Kati, E. Gelso, and D. Parthasarathy, “Safe control allocation of articulated heavy vehicles using machine learning,” in *Proc. 16th Int. Symp. on Adv. Vehicle Control*, Milan, Italy, Sep. 2024, doi: 10.1007/978-3-031-70392-8_1.

S. Ramesh, K. Shingade, U. Erdinc, M. S. Kati, and M. Jonasson, “Lateral performance enhancement in A-double vehicle combinations through active dolly steering,” presented at the 29th IAVSD Int. Symp. on Dyn. of Vehicles on Roads and Tracks, Shanghai, China, Aug. 2025.

U. Erdinc, O. A. S. Ali, J. Tian, E. Gelso, and M. S. Kati, “Learning efficiency maps of electric trailers for power loss minimization,” presented at the 29th IAVSD Int. Symp. on Dyn. of Vehicles on Roads and Tracks, Shanghai, China, Aug. 2025.

A. Bharganwar, M. V. Ramasamy, U. Erdinc, and M. Jonasson, “Adaptive slip control for electrified articulated heavy vehicles,” presented at the 17th Int. Symp. on Adv. Vehicle Control, Tsukuba, Japan, Sep. 2026.

In addition, 27 patent applications in vehicle motion control have been filed as a result of this thesis.

Nomenclature

The nomenclature applies only to the thesis, but not to the appended papers.

Symbols

a	Acceleration	m/s^2
$\mathbf{B}$	Control effectiveness matrix	
c_y	Normalized lateral acceleration	
$c_{tractor}$	Tractor driven axle friction utilization	
$c_{trailer}$	Trailer axle friction utilization	
g	Gravitational acceleration	m/s^2
$\mathbf{G}$	Matrix defining inequality constraints	
$\mathbf{h}$	Vector defining inequality constraints	
F	Force	N
P_L	Power loss	W
R	Wheel radius	m
S_x	Longitudinal slip	
T	Torque	Nm
$\mathbf{u}$	Input vector	
$\bar{\mathbf{u}}$	Upper limit vector for inputs	
$\underline{\mathbf{u}}$	Lower limit vector for inputs	
$\mathbf{v}$	Virtual control input vector	
V	Velocity	m/s
$\mathbf{W}_{\mathbf{u}}$	Weighting matrix for reference inputs	
$\mathbf{W}_{\mathbf{v}}$	Weighting matrix for virtual control inputs	
X	Global coordinate	m
Y	Global coordinate	m
α	Tire side-slip angle	rad
β	Body side-slip angle	rad
γ	Weighting term for virtual control inputs	
δ	Tractor steering angle	rad

θ	Articulation angle	rad
μ	Road friction coefficient	
Ω	Wheel rotational speed	rad/s
ω	Yaw rate	rad/s

Subscripts and superscripts

$\square_{des}$	Desired
$\square_{EM}$	Electric motor
$\square_{SB}$	Service brake
$\square_1$	1st unit
$\square_{11}$	1st axle or electric motor of the 1st unit
$\square_{12}$	2nd axle or electric motor of the 1st unit
$\square_{1f}$	Front axle of the 1st unit
$\square_{1r}$	Rear axle of the 1st unit
$\square_2$	2nd unit or equivalent axle of the 2nd unit
$\square_{2r}$	Rear axle of the 2nd unit
$\square_x$	Longitudinal direction
$\square_y$	Lateral direction
$\square_z$	Yaw or vertical direction
$\square_{est}$	Estimated
$\square_{lim}$	Limit
$\square_{max}$	Maximum
$\square_{mod}$	Modified
$\square_{pred}$	Predicted
$\square_{prop}$	Propulsion
$\square_{ref}$	Reference
$\square_{req}$	Request

Abbreviations

ABS	Anti-lock braking system
AHV	Articulated heavy vehicle
ASIL	Automotive safety integrity level
ATC	AL-KO trailer control
CA	Control allocator
CAN	Controller area network
CCA	Combination control allocator
DI	Driver interpreter
EBS	Electronic braking system
ECU	Electronic control unit
EM	Electric motor
ESC	Electronic stability control
GPS	Global positioning system
ICE	Internal combustion engine
IMU	Inertial measurement unit
LAPIC	Lateral acceleration proportional instantaneous control
LAPPC	Lateral acceleration proportional predictive control
ME	Motion estimator
MLD	Maneuvering limitation diagram
MPC	Model predictive control
MPHC	Model predictive heuristic control
OEM	Original equipment manufacturer
PEM	Predictive energy management
PI	Proportional-integral
PL	Power loss
PLM	Power loss minimization
QM	Quality management
QP	Quadratic program
RA	Rearward amplification
RSC	Roll stability control
SB	Service brake

SOC	State-of-charge
SOE	Safe operating envelope
TEBS	Trailer electronic braking system
TSC	Trailer safety control
TSC	Trailer sway control
UCA	Unit control allocator
UN	United Nations
VTM	Volvo transport model
Cap.	Capability
Req.	Request
Stat.	Status

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Part I

Extended summary

Chapter 1

Introduction

Heavy road vehicles play a fundamental role in global logistics and remain essential for meeting the demands of modern society. Although rail, maritime, and aviation transport have seen increased utilization in recent years, these modes typically cannot provide end-to-end delivery. The initial and final legs of a supply chain—often spanning tens to hundreds of kilometers—rely predominantly on heavy commercial vehicles. Therefore, optimizing road freight transport efficiency is critical. This optimization is driven not only by economic factors but also by the urgent need to reduce environmental impact. In response, regulations increasingly permit longer heavy vehicle combinations. At the same time, the transport sector is rapidly transitioning toward electrification, extending beyond towing trucks to include electrified trailers. The increasing complexity of these multi-unit heavy vehicles requires advanced vehicle motion control strategies. This research aims to develop control systems for multi-unit heavy vehicles that simultaneously prioritize safety and energy efficiency.

1.1 Background

In the European Union, the maximum permissible road train length is 18.75 meters, and the maximum weight is 40 tons, with some exceptions [1]. The most common vehicle combinations under these restrictions are tractor-semitrailer combinations, as shown in Fig. 1.1. In several European countries, heavy vehicle combinations are permitted to reach a maximum length of 25.25 meters and a gross weight of 60 tons. For instance, in Sweden, the Nordic combination—consisting of a rigid truck, a dolly, and a semitrailer—is widely utilized. Following Finland’s example, Sweden introduced new regulations in 2023 permitting high-capacity vehicle combinations up to 34.5 meters in length and with a gross combination weight of up to 74 tons on designated road networks [2]. A typical long combination is an A-double combination, as shown in Fig. 1.1, with a tractor, dolly,

and two semitrailers. As Nordic countries increasingly permit longer multi-unit vehicle combinations, broader European adoption of this trend is anticipated [3].

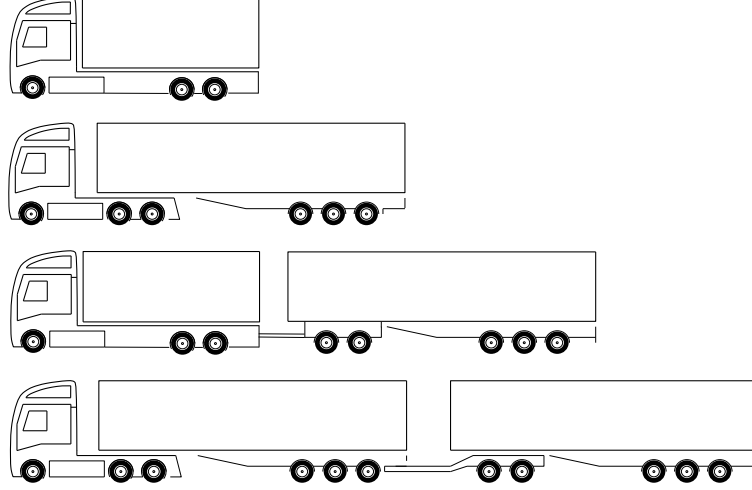


Figure 1.1: Different types of heavy vehicles (combinations): rigid truck (first), tractor-semitrailer combination (second), Nordic combination (third), A-double combination (fourth).

Vehicle electrification affects not only trucks and tractors but also trailing units such as semitrailers and dollies. Electric trailers are already becoming commercially available, such as ZF’s [4] and Trailer Dynamics’ [5] e-trailers. These trailers have batteries and electric axles, and thus can propel or brake with energy recuperation. They reduce fuel consumption and CO₂ emissions when towed by diesel trucks. When they are towed by electric trucks, they act as range extenders and traction boosters. Other examples include the Schmitz Cargobull trailer [6], where the trailer can perform regenerative braking with a smaller Electric Motor (EM), for example, to power a refrigeration system. In this case, the EM cannot provide propulsion.

Electrified units can exhibit varying degrees of control autonomy. While some e-trailers are controlled directly by the tractor, others feature independent internal controllers that require minimal data exchange, ensuring compatibility with conventional tractors with internal combustion engines. These architectural variations are outlined below and summarized in Table 1.1. Since no industry standard exists for classifying e-trailer architectures, this thesis proposes specific categories to provide a baseline understanding. Additionally, for simplicity, this work uses the term e-trailer to denote any electrified trailing unit, including e-dollies.

- **Type A:** E-trailer has EM(s) with a relatively small power capacity. This motor is used only for regenerative braking, not for propulsion. Regenerated energy is stored in a dedicated battery with relatively low capacity and used

Table 1.1: E-trailer types. The listed sensors are provided as examples; actual implementations may vary.

Type	A	B	C
Motor Power Capacity	low	high	high
Battery Capacity	low	high	high
Regenerative Braking	yes	yes	yes
Propulsion	no	yes	yes
IMU	yes	yes	yes
Articulation Angle Sensor	yes	yes	yes
Coupling Force Sensor	no	yes	no
Powering Auxiliaries	yes	yes	yes
Range Extender	no	yes	yes
Traction Booster	no	yes	yes
Startability Booster	no	yes	yes
Gradability Booster	no	yes	yes
Tractor Communication	minimal	moderate	extensive
Primary Control Node	trailer	trailer	tractor

to power auxiliaries, such as a cooling system in a refrigerated semitrailer. Communication with the tractor is minimal, and a local controller in the e-trailer is coordinating the EM(s).

- **Type B:** E-trailer has EM(s) with higher power capacity. This motor is used for both regenerative braking and propulsion. Communication with the tractor is moderate—for instance, via the ISO 11992 [7] interface—incorporating brake request but not acceleration request signals. A local controller in the e-trailer coordinates the trailer’s propulsion, possibly using data from sources such as coupling force and articulation angle sensors. The e-trailer can act as a range extender, boosting the combination’s power, gradability, startability, and traction capability.
- **Type C:** Similar to type B in which the e-trailer has EM(s) with high power capacity for regenerative braking and propulsion, and acts as a range extender, and power, gradability, startability, and traction booster. This type has the most extensive communication with the tractor, for example, via an ISO 11992 interface with additional signals such as acceleration request signals, or preferably high-speed automotive Ethernet. A control node on the tractor handles e-trailer control and provides comprehensive access to tractor and e-trailer signals. Although localized sensors—such as an artic-

ulation angle sensor—may be present on the e-trailer, a dedicated coupling force sensor is not required. The extensive state information available to the controller enables the estimation of the coupling forces directly from the respective tractor and trailer forces.

Given the variety and technological potential of electrified vehicle combinations, novel motion control strategies are essential. To address this need, this thesis focuses on Type C e-trailers.

1.2 Motivation

Octopuses are widely recognized as the most intelligent invertebrate animals. These highly capable creatures can open jars, manipulate objects, and use tools. They surprise researchers not only with their intelligence but also with the distributed architecture of that intelligence and decision-making ability across different parts of their bodies. Studies show that their arms can collect sensory information and control their movements with partial autonomy, operating without direct control from the central brain. Hence, each arm acts as if it possesses its own local processing center, or “mini-brain” [8]. In fact, around two-thirds of their neurons are located in the arms. However, for overall motion coordination and deciding the global direction of travel, the central brain retains supervisory control. Levy et al. suggest that the central brain merely decides which arm should move, but not how it executes the movement [9]. Thanks to this distributed motion control architecture featuring local “mini-brains” coordinated by a centralized main brain, the complex task of synchronizing eight arms and allocating motion among them becomes simpler and more efficient. Thus, the central brain is relieved of computationally demanding low-level decision-making, yet effectively dictates the overall motion.

Figure 1.2 illustrates an octopus alongside a Volvo FH tractor towing 20 trailers. While introducing biological imagery into an engineering study might seem unconventional, this analogy perfectly encapsulates the core motivation of this research: to establish a novel perspective on multi-unit heavy vehicle control. Ideally, the control architecture of these complex vehicle combinations should mirror the octopus’s distributed yet centrally supervised neural structure. A supervisory control allocator should distribute high-level control inputs to each vehicle unit, while each individual unit manages its own lower-level actuator coordination. The upper-level controller requires access only to the most critical sensor and actuator interfaces of each unit, offloading the lower-level computational details to the local controllers. By transitioning away from conventional control architectures and adopting a hierarchical, distributed control approach—rather than a computationally intensive, fully centralized strategy—it becomes possible to efficiently



(a) An octopus with 8 arms collecting sensory information and controlling their own motions [10].



(b) A Volvo FH tractor towing 20 trailers and 750 tons [11].

Figure 1.2: An analogy of a multi-unit heavy vehicle combination and an octopus with multiple partially self-controlling legs.

manage the complexities of modern configurations with multiple electrified units. Ultimately, this study focuses on developing safe and energy-efficient distributed control allocation frameworks for articulated heavy vehicles.

Allocating excessive propulsion or regenerative braking forces to certain units or axles to maximize energy efficiency may lead to motion instabilities such as jackknifing or trailer swing. Therefore, it is important to allocate these forces to maximize not only energy efficiency but also safety and stability. Within the envisioned distributed motion control architecture, the safe and energy-efficient distribution of forces across vehicle units must be managed at the upper level. Therefore, this study focuses on this higher-level controller, termed the “combination control allocator”.

1.3 Research questions

The following research questions are addressed in this thesis:

- RQ1.** How does the allocation of large longitudinal forces across e-axles under low-friction conditions impact the yaw stability of multi-unit heavy vehicle combinations?
- RQ2.** How can yaw stability be represented through a Safe Operating Envelope (SOE), and what are the methodological differences between force-domain and slip-domain representations?
- RQ3.** How can a hierarchical and distributed control architecture coordinate the over-actuation introduced by multiple electrified units in heavy vehicle combinations to achieve energy-efficient vehicle motion control?

- RQ4.** How can safe operating envelopes be integrated into an energy-efficient control allocation framework to prevent yaw instabilities induced by tire force saturation?
- RQ5.** How can motion predictions—varying in prediction horizons, state measurement dependencies, and computational complexity—be integrated into control allocation to proactively mitigate yaw instabilities?
- RQ6.** How should the functional architecture and unit-to-unit interfaces be designed to facilitate safe and energy-efficient allocation?

1.4 Limitations

Fig. 1.3 presents the key components of the envisioned Volvo functional architecture. The scope of this study is limited to the functions highlighted in blue within vehicle motion and power management: motion prediction, power management, and motion coordination.

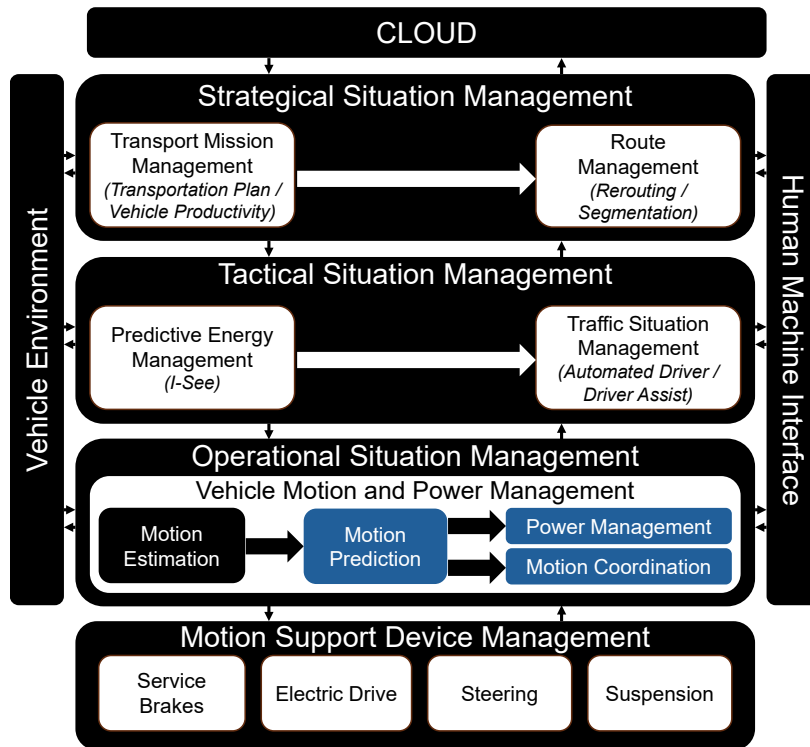


Figure 1.3: Envisioned Volvo functional architecture. The scope of this research is limited to the functions highlighted in blue.

The scope of this study is further limited to:

- control allocation without involving any predictive energy management,
- instantaneous power loss minimization, without considering tire wear, battery degradation, and other costs of ownership,
- allocation of a total force request into different units at the combination level, without detailing the allocation at the unit level to each actuator,
- allocation of longitudinal forces, excluding any lateral control variables such as lateral forces, yaw moments, or steering angles,
- preserving the yaw stability of the combination, excluding the roll stability,
- addressing divergent yaw instabilities such as jackknifing and trailer swing, but not dynamic (oscillating) yaw instabilities such as trailer sway,
- standard driving scenarios on the public road,
- pure braking or propelling, but not propelling some axles and braking some other axles at the same time,
- theoretical control allocation bounds that prioritize energy and safety performance, explicitly excluding current and pending regulatory constraints that prohibit e-trailers from pushing the tractor—such as UN Regulation 13 amendments [12]—thereby evaluating the system’s unconstrained physical potential,
- evaluating e-trailers without accounting for the cargo payload reduction caused by battery weight,
- control systems assuming available knowledge of the road friction coefficient and required vehicle states, such as articulation angle, lateral acceleration, and yaw rates, obtained via sensors or a motion estimator,
- scenarios with instantaneous signal transmission, assuming negligible delays from communication transport, sensors, and internal ECU processing,
- nominal system operation, thereby excluding system failures, fault isolation strategies, fault-tolerant functional architecture design, and protective barriers against erroneous inputs or software faults.

1.5 Scientific contributions

The scientific contributions of this thesis to the literature are shown in Fig. 1.4. The main content and contributions are listed under each paper, together with (if applicable) the section or chapter that relates to the listed content.

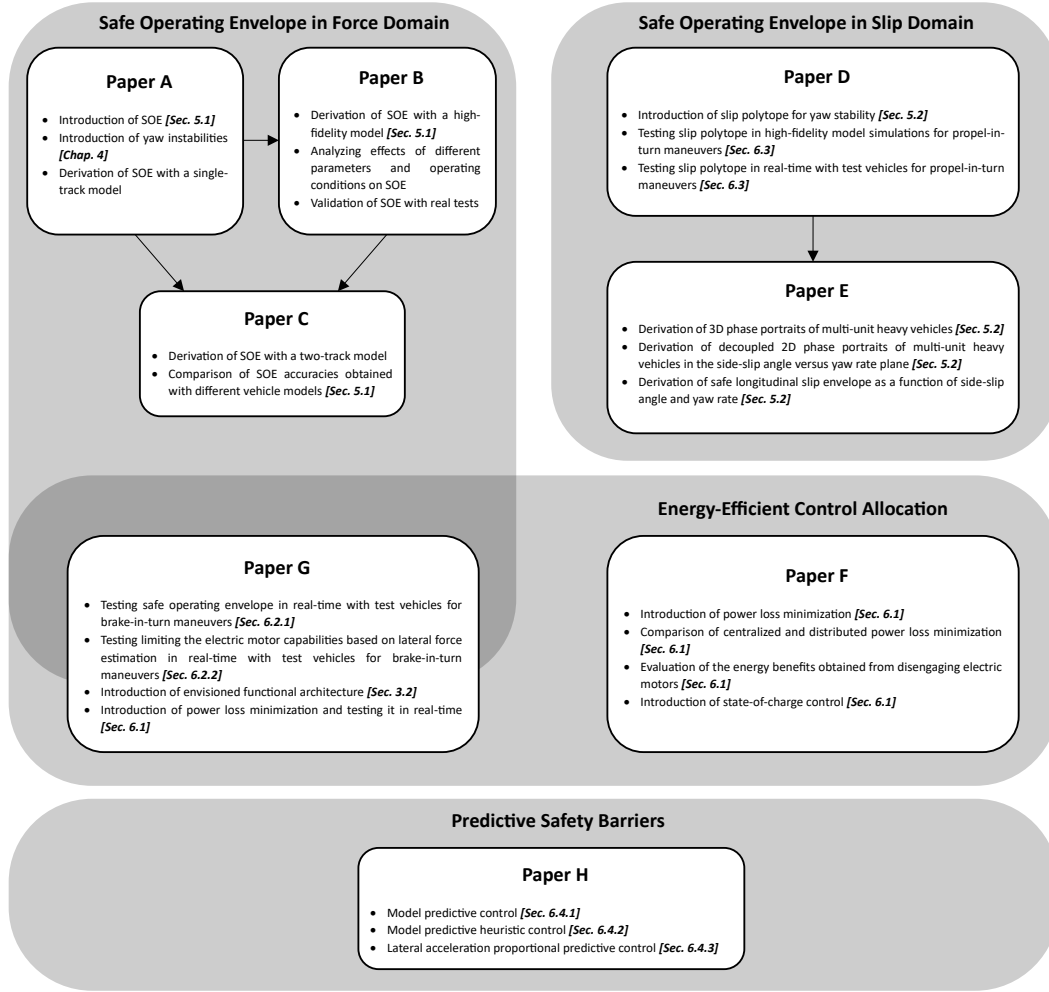


Figure 1.4: Overview of the scientific contributions presented in this thesis. The arrows indicate the recommended order for reading the appended papers.

Chapter 2

Control allocation

Road vehicles historically had a simpler set of actuators to fulfill the motion request. Internal combustion engines were used to generate longitudinal forces to accelerate the vehicle or maintain speed. A steered front axle was used to steer the vehicle in the desired direction. Service brakes were used to slow down or stop the vehicle. With this trivial set of actuators, vehicle motion control was relatively simple. Directional control and propulsion control had trivial control strategies. Unlike the propulsion and steering systems, the brake system had some redundancy, meaning there were multiple ways to achieve the same total brake force. Nevertheless, braking was performed with a fixed distribution among front and rear axle service brakes. In the 1990s, electronic stability controllers were introduced, using this redundancy to avoid instabilities by engaging only a subset of the service brakes. Modern heavy vehicles, by contrast, have much higher actuator redundancy, with multiple actively steered axles, multiple electrified axles per unit, and multiple electrified units. As a result, this over-actuation provides multiple ways to fulfill motion requests. Unlike traditional vehicles, there is now a stronger motivation to distribute actuator requests more optimally for energy efficiency across multiple electric axles or electrified units. Controlling an under-determined vehicle system and solving for more vehicle inputs (actuator requests) than the number of system equations (degrees of freedom under control), also known as “control allocation”, becomes significantly important when it comes to optimally distributing the actuator inputs while minimizing a cost function. This chapter first introduces the concept of over-actuation in heavy vehicles. It then discusses controller reconfigurability and modularity, followed by an overview of optimization-based control allocation. Finally, it details various methods for integrating safety constraints into the control allocator.

2.1 Over-actuation

Traditional road vehicles have one actuator to control each of the longitudinal and lateral dynamics: an internal combustion engine and a steered axle. For braking, there are several service brakes, but aside from electronic stability controllers, drivers or vehicle controllers merely benefit from this redundancy. Similar to traditional passenger vehicles, conventional Articulated Heavy Vehicles (AHVs) also have no redundancy in propulsion and steering, as shown in Fig. 2.1.

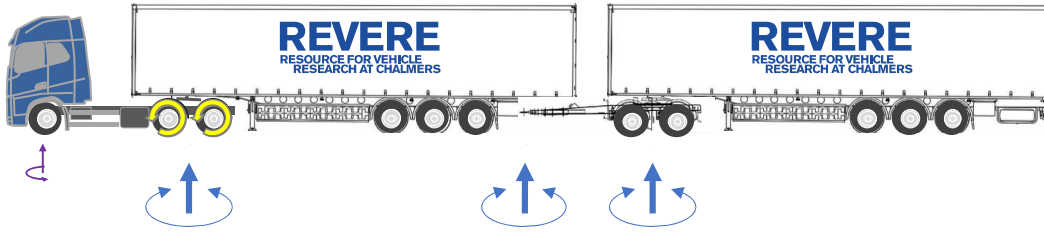


Figure 2.1: A conventional A-double vehicle combination. Purple arrows indicate steered axles, blue arrows indicate articulation points, and yellow arrows indicate diesel engine-propelled axles.

The electrification trend affected not only the towing units of heavy vehicles, namely trucks and tractors, but also the trailing units, such as semitrailers and dollies. Thus, modern AHVs may exhibit extensive over-actuation in their electric propulsion systems, as shown in Fig. 2.2. In this example, all tractor and dolly axles are electrified. Additionally, semitrailers also have electrified axles. Beyond propulsion redundancy, steering is also overactuated because all units have steered axles.

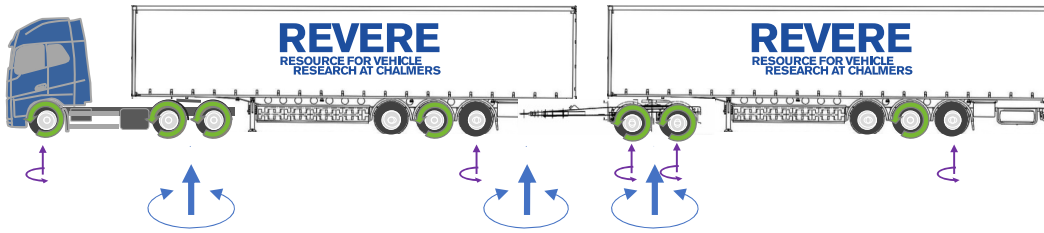


Figure 2.2: A modern A-double vehicle combination. Purple arrows indicate steered axles, blue arrows indicate articulation points, and green arrows indicate electric-propelled axles.

Currently, electric motors are usually connected to the axles through gear trains and differentials, as in Volvo FH trucks [13]. Furthermore, as wheel-hub electric motors and individually propelled wheels become more popular, redundancy in the propulsion system can also generate a yaw moment. Given this extensive over-actuation in modern AHVs, an optimal control allocation strategy is critical.

2.2 Reconfigurability

The extensive diversity in vehicle combination configurations—including the number and type of units, the autonomy levels of electrified trailers, and the available sensor and interface architectures—makes reconfigurability a critical requirement for control allocation. In this context, Laine [14] categorizes reconfigurability into online and offline adaptivity.

- Offline reconfigurability means the control system can be easily configured to accommodate different actuator types and numbers without changing the control law. Some examples are listed below.
 - The truck control system parameters can be factory-configured to accommodate various axle configurations, such as 4x2, 6x4, and 8x4.
 - Upon vehicle initialization, the combination control system configures its parameters using information transmitted by the trailing units. This modularity enables the system to adapt to various combination topologies—such as a single conventional semitrailer or an A-double equipped with two e-trailers and a dolly—and to account for configuration changes, such as the status of liftable axles on the tractor or trailers.
- Online reconfigurability, on the other hand, is the ability of the controller to handle instantaneous operating conditions and limitations of the actuators and the environment, and smooth arbitration for minimizing energy consumption and ensuring vehicle stability. Some examples are listed below.
 - The control system can adapt to changes in the road friction and other environmental parameters.
 - The control system can adapt if an actuator is faulty. The control system sets the faulty actuator’s capabilities to zero. The control system then prioritizes the remaining actuators.
 - The control system can normally minimize energy consumption, but when it detects instability, it can prioritize safety over minimizing energy consumption.

Laine proposed an offline and online reconfigurable vehicle motion control system for a passenger car [14]. The aim of the studies presented in this PhD thesis is to develop online and offline reconfigurable vehicle motion controllers for multi-unit vehicles that are easy to adapt for different vehicle and actuator combinations without changing the control law, ensuring both vehicle stability and energy efficiency.

2.3 Algorithmic architectures

Vehicles have historically had distributed motion controllers: an engine controller for internal combustion engines, an Electronic Braking System (EBS) for the service brakes, and a steering controller for the steered axle (for power-assisted steering or advanced driver-assistance systems such as lane-keeping assist). Furthermore, each subsystem may also have its own layered control architecture to control high-level dynamics or low-level actuator states. For example, a higher-level controller in EBS can distribute brake forces across different axles, while a lower-level Anti-lock Braking System (ABS) can ensure that the requested brake force is applied without causing wheel lock.

Gäfvert [15] classified algorithmic partitioning with three structures as shown in Fig. 2.3.

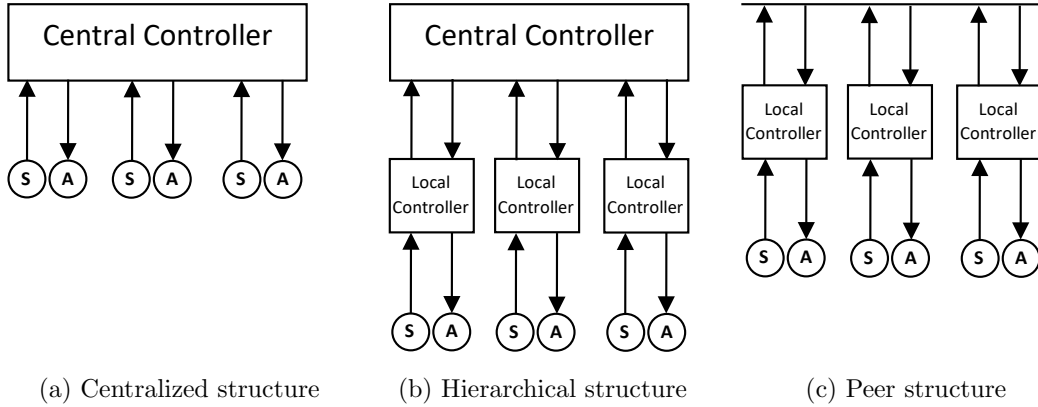


Figure 2.3: Algorithmic architectures [15]. S denotes sensor, A denotes actuator.

- **Centralized Structure:** All algorithmic blocks are merged into one central block, in which all the sensor data are received by this block, and all actuators are actuated directly from this block. Although this architecture can outperform distributed architectures [14] because all signals are simultaneously available to the controller, it lacks modularity, making it difficult to add new functionality or modify existing functions. A centralized controller is the least reconfigurable structure.
- **Hierarchical Structure:** An upper-level central controller coordinates different local controllers. Each local controller has access to actuators and sensor data. This structure offers modularity and distributed control, but the central controller handles coordination. A hierarchical architecture has high reconfigurability.

- **Peer Structure:** Algorithms are partitioned with no explicit hierarchy. No upper-level controller coordinates the local controllers, but local nodes can share information with one another. This is the most modular structure, with many distributed controllers but lacking the coordination provided by a central controller; as a result, conflicts among local nodes may occur.

Traditional vehicles with distributed motion controllers fall into the peer structure. A local engine controller controls the internal combustion engine, another local controller controls the brakes (typically an EBS system), and another local steering controller handles advanced steering functions. With the trend toward electrification, brake blending has become an important factor in minimizing energy consumption. Hence, it is important to coordinate the service braking with regenerative braking performed by the electric motors. The brake blending function in modern vehicles follows a hierarchical structure, since an upper-level controller computes the total brake force and a lower-level controller distributes it between the electric propulsion and service brake systems. Laine [14] proposed a hierarchical structure to coordinate these systems for a passenger car. The driver interpreter, path controller, and control allocator are three layers of the vehicle motion control system at the upper hierarchical level. Local controllers of each actuator system are located at the lower hierarchical level. Hence, the upper-level vehicle motion controller coordinates the lower-level controller nodes for the service brakes, the internal combustion engine, and (if autonomously steered) the steering system.

Some examples of algorithmic architectures for motion coordination of multi-unit vehicles are shown in Fig. 2.4. A centralized controller (Fig. 2.4.a) that jointly optimizes the velocity profile and torque allocation for each actuator on each unit can outperform (in terms of safety and energy efficiency) the distributed control allocators due to its access to all control signals and sensor data. However, Torinsson et al. showed that distributed predictive energy management and instantaneous control allocation with a hierarchical architecture (Fig. 2.4.b) can achieve energy savings comparable to those of a centralized controller, with an appropriate signal interface in between [16].

Following the distributed hierarchical architecture with centralized control allocation, even further algorithmic partitioning can be performed by splitting the control allocation among two levels (Fig. 2.4.c): a combination-level control allocator, which distributes the total force request coming from the predictive energy management to different units, and a lower-level control allocator for each unit that distributes each and every actuator force.

This hierarchical architecture offers high reconfigurability for different types and numbers of trailing units. Furthermore, thanks to the distributed control architecture, the local controllers can be run in different Electronic Control Units

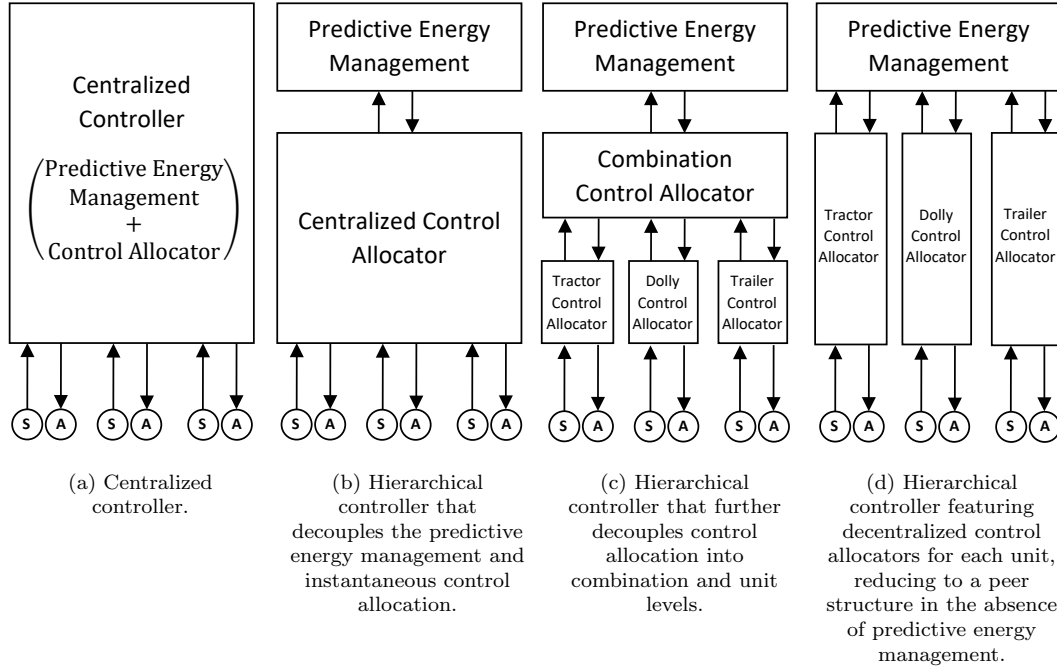


Figure 2.4: Motion coordination architectures for multi-unit vehicles. S denotes sensor, and A denotes actuator.

(ECUs), which may be located in the same or different vehicle units.

The algorithmic architecture illustrated in Fig. 2.4.d reduces to a peer structure if the predictive energy management layer is omitted. This configuration is exemplified by a conventional Internal Combustion Engine (ICE) tractor and e-trailer hybrid combination under manual driver control, operating without high-level predictive energy management. In this scenario, the e-trailer utilizes a local control allocator, eliminating the need for a centralized combination controller or an extensive signal interface with the tractor.

The research presented in this PhD thesis excludes predictive energy management and focuses exclusively on the instantaneous, safe, and energy-efficient allocation of actuator forces across vehicle units and axles. Consequently, the vehicle's velocity profile is not optimized over a future horizon; instead, it is assumed to be known *a priori*. The studies in this PhD thesis adopt the architecture shown in Fig. 2.4.c and focus specifically on the combination control allocator.

2.4 Optimization-based control allocation

Solving a control problem for an over-actuated dynamical system and distributing control inputs among several actuators to track a set of reference signals is called “control allocation”. For a set of reference control signals (such as total longitudinal and lateral force, and yaw moment on the vehicle), $\mathbf{v}$, also named as “virtual control input vector”, and for a linear control problem that outputs the actuator requests (such as electric motor and service brake torques and steering angle), $\mathbf{u}$, the problem can be formulated as:

$$\mathbf{v} = \mathbf{B}\mathbf{u} , \quad (2.1)$$

where $\mathbf{B}$ is called the “control effectiveness matrix” and $\mathbf{v} \in \mathbb{R}^{k \times 1}$, $\mathbf{u} \in \mathbb{R}^{m \times 1}$, $\mathbf{B} \in \mathbb{R}^{k \times m}$. If $k = m$, which means the number of virtual control inputs and the actuator requests are equal (hence the system is fully actuated), and $\mathbf{B}$ is invertible, then (2.1) can be solved for the solution $\mathbf{u}$ as:

$$\mathbf{u} = \mathbf{B}^{-1}\mathbf{v} . \quad (2.2)$$

In this trivial case, $\mathbf{B}$ is a square matrix, and the system has no redundancy. In the case of over-actuation with $k < m$ for an underdetermined system, the matrix $\mathbf{B}$ is not square and not invertible. Hence, a more advanced method is needed instead of (2.2). There are multiple options for control allocation, such as pseudo-inverse, direct allocation, daisy-chaining, and optimization-based methods [18]. Here, optimization-based methods will be elaborated.

Solving the control allocation problem via optimization allows using multiple objectives while ensuring the virtual control input signals are met. Härkegård [19] and Laine [14] solved the control allocation problem for flight control and vehicle control, respectively, with the following formulation:

$$\mathbf{u} = \arg \min_{\mathbf{u} \leq \mathbf{u} \leq \bar{\mathbf{u}}} \|\mathbf{W}_{\mathbf{u}}(\mathbf{u} - \mathbf{u}_{\text{des}})\|_2^2 + \gamma \|\mathbf{W}_{\mathbf{v}}(\mathbf{B}\mathbf{u} - \mathbf{v})\|_2^2 . \quad (2.3)$$

The first term $\|\mathbf{W}_{\mathbf{u}}(\mathbf{u} - \mathbf{u}_{\text{des}})\|_2^2$ is for tracking the desired reference control inputs, $\mathbf{u}_{\text{des}}$. Desired control inputs can be sent from, e.g., the energy management and steering controllers. Even if no reference signal is sent from such controllers, this term (with $\mathbf{u}_{\text{des}} = 0$) penalizes the use of any actuators, which can, e.g., be useful for avoiding actuator fights, such as front-axle propulsion and rear-axle braking. $\mathbf{W}_{\mathbf{u}}$ is the weighting matrix for setting the weights of each reference input tracking term. The second term $\|\mathbf{W}_{\mathbf{v}}(\mathbf{B}\mathbf{u} - \mathbf{v})\|_2^2$ is for fulfilling the desired motion of the vehicle. $\mathbf{W}_{\mathbf{v}}$ is the weighting matrix for setting the weight of each virtual control input term. γ is another weighting factor that determines the prioritization of the second term over the first. $\underline{\mathbf{u}}$ and $\bar{\mathbf{u}}$ are the minimum and

maximum actuator capabilities, respectively. The formulation uses the squared 2-norm; hence, it becomes a quadratic problem. Alternatively, the second term can be written as an equality constraint, and only the first term can be optimized.

For vehicles with multiple electric motors, a power loss minimization problem can be solved by distributing the motor torques in an energy-efficient way [20], [21]:

$$\begin{aligned} \mathbf{u} &= \arg \min_{\mathbf{u} \leq \mathbf{u} \leq \bar{\mathbf{u}}} P_L(\mathbf{u}) , \\ \text{s.t. } \mathbf{v} &= \mathbf{B}\mathbf{u} . \end{aligned} \quad (2.4)$$

In this formulation, instead of reference input tracking term, $\|\mathbf{W}_u(\mathbf{u} - \mathbf{u}_{\text{des}})\|_2^2$, the total power loss (due to electric motor losses, inverter losses, driveline, and battery losses), $P_L(\mathbf{u})$, is minimized. These losses are usually approximated as a quadratic power loss function for a given motor speed; hence, the formulation becomes a quadratic problem that is more computationally efficient to solve. Fulfilling the reference virtual control inputs, $\mathbf{v}$, is done via the equality constraint.

In the literature, control allocation is also combined with predictive energy management, considering a road profile over a 5–20 km horizon [17]. However, predictive energy management is outside the scope of the studies presented in this PhD thesis.

2.5 Stability constraints

While allocating the actuator inputs, it is important to maintain stability. Applying excessive force to the tractor's driven axle may cause jackknifing, especially on slippery, low-friction roads and in sharper curves. Likewise, excessive forces on the trailer axles may cause trailer swing. Both incidents pose significant risks and may lead to fatal accidents. Therefore, it is important to avoid risky force allocation.

A set of inequality constraints can be added to the control allocation problem given in (2.4) as follows:

$$\mathbf{G}\mathbf{u} \leq \mathbf{h} , \quad (2.5)$$

where matrix $\mathbf{G}$ and vector $\mathbf{h}$ represent the inequality conditions imposed on the allocated actuator requests $\mathbf{u}$. Matrix $\mathbf{G}$ and vector $\mathbf{h}$ can be functions of certain vehicle states, such as lateral acceleration. Inequality constraints formulated in (2.5) can be used, e.g., to avoid excessive tractor braking when the trailer is not braking, and vice versa, at high lateral accelerations. This method, namely “safe operating envelope”, will be elaborated in Chapter 5.

If the actuator force constraints are straightforward and do not relate to each other, they can simply be implemented by using upper and lower bounds for $\mathbf{u}$: $\bar{\mathbf{u}}$ and $\underline{\mathbf{u}}$. As an example, longitudinal forces applied by the service brakes and electric motors can be limited considering the lateral force estimations and a friction circle as explained in Section 6.2.2. Likewise, maximum and minimum actuator forces can be limited as a function of certain vehicle states, such as lateral acceleration.

Chapter 3

Envisioned functional architecture and interface

Historically, communication between different units in heavy vehicle combinations has been performed via the ISO 11992 CAN network. Because traditional trailers lack a propulsion system, this network's signal database is limited; consequently, it primarily manages trailer braking. Furthermore, trailer control systems are very limited and typically consist only of a Trailer Electronic Brake System (TEBS) and air suspension controllers. Electrification of modern heavy vehicle combinations requires a novel, standardized interface and functional architecture. This chapter first details the current functional architecture and interfaces used in vehicle combinations, then introduces the envisioned architecture and interface.

3.1 Current functional architecture and interface

European regulations mandate the use of the Anti-lock Braking System (ABS) on trailing units within heavy vehicle combinations. Typically, ABS is embedded within a broader Electronic Braking System (EBS) and operates in conjunction with electronic brake-force distribution and Roll Stability Control (RSC). While air suspension systems are also standard on most heavy trailers, these units rarely incorporate any other actuators or advanced control functions, aside from specific body-builder equipment. As a result, trailers have significantly lower operational complexity and fewer Electronic Control Units (ECUs) than towing vehicles such as rigid trucks or tractors.

A simplified functional architecture of a Nordic combination consisting of a truck, dolly, and semitrailer is shown in Fig. 3.1. Although the detailed computational nodes and interfaces are simplified and some actuators and control systems

(such as air suspension systems) are omitted for clarity, the main control systems and actuators are illustrated. Sensors are depicted in orange, and actuators are shown in blue.

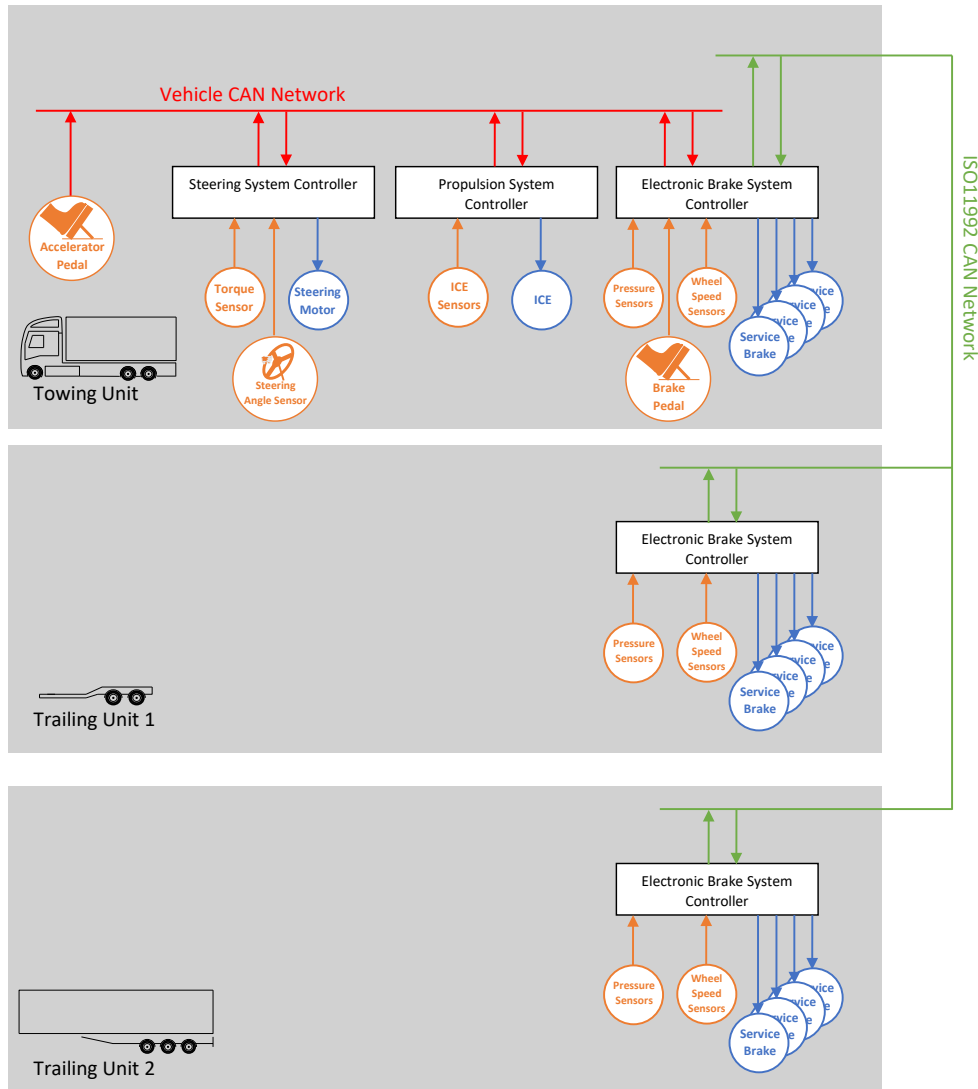


Figure 3.1: Functional architecture of traditional European heavy vehicle combinations.

The steering controller in an advanced steering system, e.g., with lane-keeping assist (and thus a steering motor to generate torque), has access to steering wheel angle and torque measurements at the steering column. Using these signals, the steering controller regulates the steering motor.

The propulsion system controller can access various Internal Combustion Engine (ICE) sensors and control the ICE based on accelerator pedal position. The accelerator pedal signal is not only communicated to the propulsion system but is also available to many other controllers. Hence, in the figure, it is shown as connected to the vehicle CAN network.

The EBS is typically not developed by automotive Original Equipment Manufacturers (OEMs); instead, it is outsourced, along with the Service Brake (SB) actuators and sensors. Heavy vehicles are equipped with pneumatic brakes. Among others, pressure sensor and wheel speed sensor measurements are communicated to the EBS controller. Furthermore, the brake pedal sensor is typically read directly by the EBS. The EBS then controls the SBs accordingly. Brake forces are distributed proportionally to the axle loads.

TEBS is functionally similar to truck EBS; however, because it lacks direct input from the brake pedal, the truck EBS communicates the overall retardation request to the trailer via the standardized ISO 11992 CAN network [7]. The truck EBS can slightly modify this request for coupling force control, though these adjustments are limited in magnitude. The only exception occurs during stability control events, where the truck can actively intervene and request trailer braking independent of the truck's own retardation target.

ISO 11992 CAN network is defined by the corresponding ISO standard and has limited content. For example, it contains no information about the accelerator pedal position or the truck's acceleration target. However, for conventional trailers, this extensive set of signals is unnecessary, as they are not propelled.

As shown in Fig. 3.1, the algorithmic architectures of traditional vehicle combination systems are mostly peer structures, as explained in Section 2.3. Steering, propulsion, and braking systems control the motion actuators without an upper-level coordinator, although they communicate with one another via the vehicle CAN network. Furthermore, the overall combination control is also similar to the peer structure: each unit's EBS controls the SBs of its corresponding unit, with no upper-level coordinator, yet they communicate with each other via the ISO 11992 CAN network.

Integrating electrified powertrains and advanced steering systems into the trailing units requires a more advanced functional architecture and interface. In the following sections, the envisioned architecture and interface are elaborated.

3.2 Envisioned functional architecture and interface

The envisioned functional architecture for a representative Nordic combination—consisting of an electric truck, an actively steered dolly, and an electric semi-

trailer—is depicted in Fig. 3.2. While the complete architecture includes lateral control strategies such as trailer steering or yaw-moment generation via differential braking or propulsion, the primary focus of this study is on longitudinal force allocation (propulsion and SBs). However, the proposed longitudinal motion control incorporates lateral motion awareness, ensuring that longitudinal actuator force allocation does not compromise vehicle stability.

A Driver Interpreter (DI) receives inputs from the accelerator pedal, brake pedal, and steering wheel, translating them into a positive or negative acceleration request sent to the Combination Control Allocator (CCA). Alternatively, if predictive cruise control is active, the Predictive Energy Management (PEM) system determines an optimal velocity profile based on upcoming road information, such as slope and curvature. The PEM then sends an acceleration request to the CCA, which establishes a feedback loop by returning acceleration status and capability signals to the PEM.

Typically, the CCA converts the incoming acceleration request into a total longitudinal force request. However, in an alternative architecture, the DI or PEM can perform this conversion directly. In this setup, the total force request is sent to the CCA, which then feeds total force capability and status signals back to the PEM.

The Motion Estimator (ME) processes data from the Global Positioning System (GPS), Inertial Measurement Unit (IMU), wheel speed sensors, and other sensors to calculate and communicate estimated vehicle states and parameters, such as unit speeds, yaw rates, side-slip angles, articulation angles, unit masses, total driven and non-driven axle loads, and road friction.

The CCA distributes the total longitudinal force request across the units as total Electric Motor (EM) and SB forces per unit, while Unit Control Allocators (UCAs) manage lower-level actuator allocation. To enable this allocation, the UCAs send status and capability signals back to the CCA. When power loss minimization is utilized in the control system, another important signal is the set of coefficients for the unit-wise lumped power loss. This lumped representation aggregates the unit's total power loss into a single model, accounting for multiple EMs, gear trains, inverters, and the battery. The power loss is approximated as a quadratic function of the applied force or torque at any given instantaneous EM operating speed. Consequently, the signal communicated simply consists of the three coefficients that define this curve (the quadratic, linear, and constant terms).

The UCA of each unit allocates the requested actuator forces to the EMs and SBs. While the EBS controller and propulsion system controller remain present in Fig. 3.2, their roles involve lower-level actuator control, such as regulating the voltage and current for the EMs and managing the pneumatic pressures for the SBs. In this architecture, the EBS is configured to execute individual external

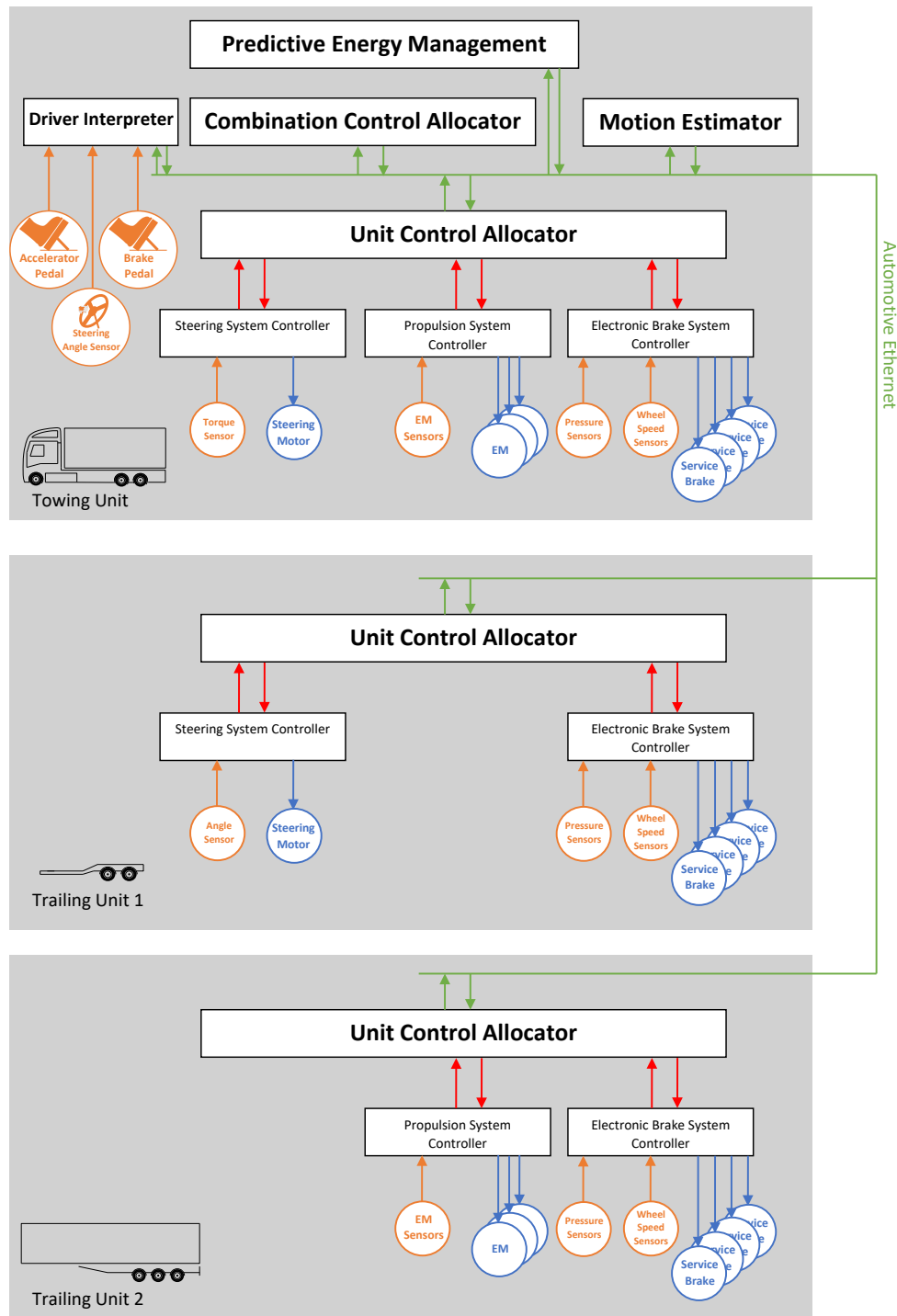


Figure 3.2: Envisioned functional architecture for heavy vehicle combinations.

braking requests from an upper-level controller, such as the UCA. Additionally, for units with multiple EMs, the UCA may perform local power loss minimization.

Communication between the CCA and UCAs can be facilitated via automotive Ethernet, which offers significantly higher bandwidth. This added capacity enables the transmission of a larger volume of signals at higher resolutions and update frequencies. Alternatively, the condensed signal set presented in Table 3.1 can be transmitted via a standard CAN. However, this approach requires an updated version of the ISO 11992 standard tailored specifically for electrified vehicle combinations.

Table 3.1: Envisioned signal interface to and from the combination control allocator for the longitudinal motion coordination. Req.=Request, Stat.=Status, Cap.=Capability.

Signal	Req.	Stat.	Cap.	from	to
Steering angle		✓		DI	CCA
Longitudinal acceleration req.	✓			DI/PEM	CCA
Longitudinal acceleration		✓	✓	CCA	DI/PEM
Total unit service brake force req.	✓			CCA	UCA
Total unit service brake force		✓	✓	UCA	CCA
Total unit electric motor force req.	✓			CCA	UCA
Total unit electric motor force		✓	✓	UCA	CCA
Unit lumped EM power loss map		✓		UCA	CCA
SOC of the unit's battery		✓		UCA	CCA
Longitudinal slips limits on e-axles	✓			CCA	UCA
Longitudinal slips on e-axles		✓	✓	UCA	CCA
Unit mass		✓		ME	CCA
Unit non-driven axles normal load		✓		ME	CCA
Unit driven axles normal load		✓		ME	CCA
Unit longitudinal velocity		✓		ME	CCA
Unit lateral velocity		✓		ME	CCA
Unit longitudinal acceleration		✓		ME	CCA
Unit lateral acceleration		✓		ME	CCA
Unit yaw rate		✓		ME	CCA
Unit side-slip angle		✓		ME	CCA
Unit articulation angle		✓		ME	CCA
Unit coupling force		✓		ME	CCA
Road friction		✓		ME	CCA

In the envisioned functional architecture and signal interface proposed in this section, a steer-by-wire system is considered. Hence, the DI receives the steering wheel angle input and converts it to a curvature request sent to the CCA. The

CCA allocates the road wheel steering angle (or steering angle rate) requests into different units. If the steering system is not a steer-by-wire type, then the steering angle sensor can also be read by the local steering system controller of the truck, the CCA cannot request a steering angle from the truck CCA, but instead, the steered angle is directly controlled by the driver manually. The steering system controller in the truck can still apply additional torque to the steering column for driver assistance systems. In the case of an active steering system in trailing units, the CCA can still request steering angles of the trailing units. If there are multiple steered axles in the units, the CCA can alternatively request a lateral force or curvature from the UCAs. Then UCAs allocate the steering angles to various steered axles. In another alternative, the CCA can request an equivalent (lumped) steering angle considering a kinematic vehicle model and the UCA allocates the steering angles to each steered axle locally. In addition, CCA can allocate yaw moments into the units, and UCAs can allocate differential braking or propulsion forces into individual wheels to fulfill those yaw moment requests. The steering and yaw moment coordination together with longitudinal dynamics coordination at the combination level via the CCA can bring advantages such as improved stability, reduced swept path, and improved maneuverability. However, it is important to note that steering and yaw moment control are beyond the scope of this study.

The envisioned functional architecture proposed in this section assumes a steer-by-wire system. In this configuration, the DI receives the steering wheel angle input and converts it into a curvature request for the CCA. The CCA then allocates road wheel angle (or angle rate) requests to the various units. If the truck does not feature a steer-by-wire system, the road wheel angle is controlled manually by the driver. In this case, the CCA cannot request a steering angle for the truck; however, the truck's local steering controller can still read the steering angle sensor to apply supplementary torque for driver assistance systems. Even with manual steering on the towing unit, the CCA can still request steering angles for any actively steered trailing units.

For units equipped with multiple steered axles, the CCA can request an overall lateral force or curvature from the UCA, which then distributes the appropriate steering angles to the individual axles. Alternatively, the CCA can request an equivalent (lumped) steering angle based on a kinematic vehicle model, leaving the local axle-wise allocation to the UCA. Furthermore, the CCA can distribute yaw moment requests to the units, which the UCAs fulfill via differential braking or propulsion at the individual wheels. Coordinating lateral and yaw dynamics with longitudinal dynamics at the combination level yields significant advantages, including enhanced stability, a reduced swept path, and improved maneuverability. However, lateral control strategies—specifically steering and yaw moment allocation—remain outside the scope of this study.

The energy-efficiency and safety aspects of the control allocators can be decoupled to create a more modular functional architecture. Consequently, only the safety-critical components of the control allocation must undergo rigorous Automotive Safety Integrity Level (ASIL) certification, while the energy-efficiency algorithms can be developed under the baseline Quality Management (QM) designation [22]. Further details on the control allocators' internal functional partitioning are explained in Section 6.

The primary focus of this thesis is the Combination Control Allocator (CCA), with a secondary emphasis on the Unit Control Allocator (UCA). The Driver Interpreter (DI), Predictive Energy Management (PEM), Motion Estimator (ME), and lower-level actuator controllers fall outside the scope of this work.

Chapter 4

Unsafe modes of articulated heavy vehicles

In recent years, longer combinations with multiple articulation points have become more common due to their commercial and environmental benefits. In general, these vehicles tend to exhibit motion instabilities, such as jackknifing, trailer swing, combination spin-out, trailer sway, and rollover. The first four relate to the yaw degree of freedom, and the last one relates to the roll degree of freedom. Jackknifing and trailer swing occur when the towing or trailing units lose their lateral grip, usually due to excessive braking or propulsion forces and locked wheels on slippery surfaces. Combination spin-out is defined as the simultaneous occurrence of the two aforementioned instability modes, resulting in all vehicle units sliding sideways. Trailer sway, in contrast, is a more dynamic problem related to the rearward amplification of multiple units. The lateral velocity or yaw rate of the trailing units is amplified relative to the towing unit and may result in fatal accidents. Finally, rollover occurs at high lateral accelerations if the payload's center of gravity is high. This chapter explains these motion instabilities in detail.

4.1 Jackknifing

Jackknifing is defined as the loss of directional stability of the towing unit. This results in excessive articulation angle and a large heading-angle deviation of the towing unit from the intended direction of travel. It happens when the towing unit loses lateral tire grip and starts to slide sideways. The loss of lateral tire grip is typically caused by excessive combined tire slip, which involves high longitudinal and lateral tire forces. Fig. 4.1.a illustrates a typical jackknifing of an Articulated Heavy Vehicle (AHV). The tractor and semitrailer are depicted by the blue and gray units, respectively.

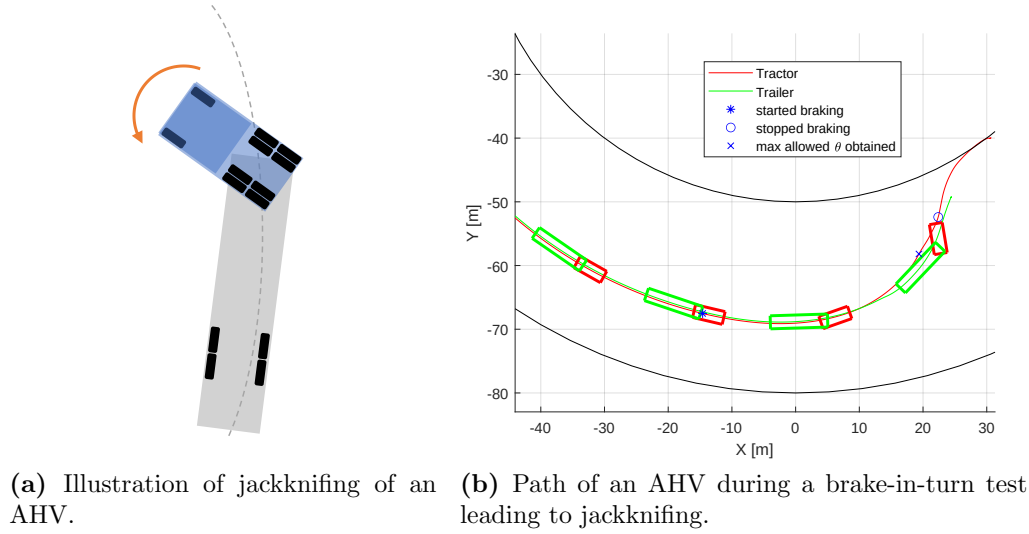


Figure 4.1: Illustration of jackknifing and the corresponding path.

In Fig. 4.1.b, the path followed by a real tractor-semitrailer combination during a brake-in-turn test on snow is illustrated. The tractor is braked extensively with the drive axle at the point marked with the blue asterisk. This results in the activation of the slip controller, and thus brake forces are limited. After braking, the AHV exhibits jackknifing: the tractor loses grip on the drive axles and starts to slide sideways, causing the articulation angle to increase undesirably.

Without timely intervention, jackknifing can cause the tractor cab to collide with the semitrailer, potentially resulting in a fatal accident. Reducing propulsion or brake forces, or applying a corrective yaw moment via differential braking, can mitigate jackknifing. Trucks and tractors have Electronic Stability Control (ESC) systems, which detect yaw instability and take preventive actions, such as differential braking, to apply a corrective yaw moment.

4.2 Trailer swing

Trailer swing is defined as the loss of directional stability of the trailing unit, resulting in excessive articulation angle and a large heading-angle deviation of the trailing unit from the intended direction of travel. This happens when the trailing unit loses lateral tire grip and starts to slide sideways. Losing lateral tire grip is typically caused by excessive combined tire slip, which involves high longitudinal and lateral tire forces. A typical trailer swing of an AHV is illustrated in Fig. 4.2.a.

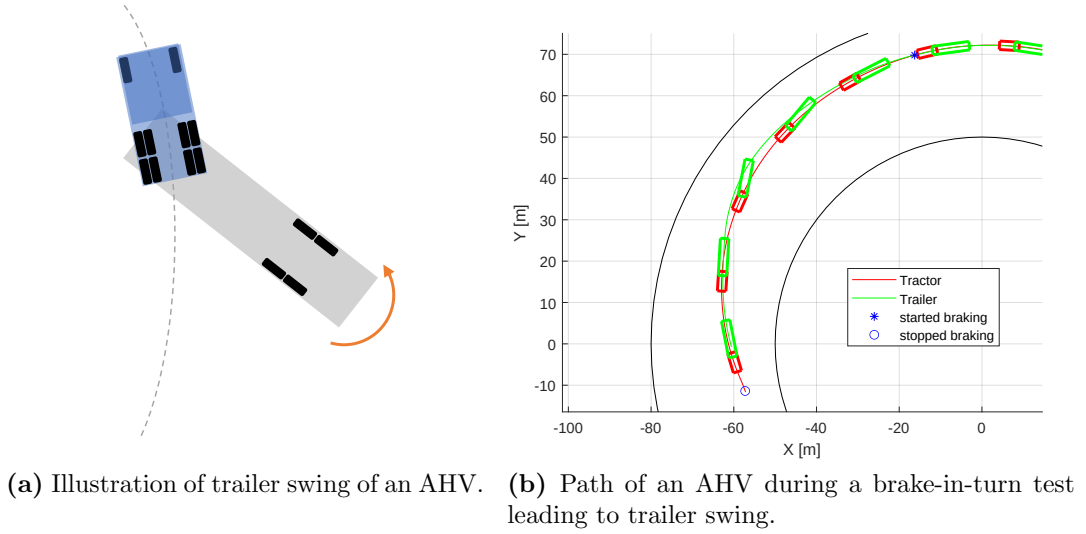


Figure 4.2: Illustration of trailer swing and the corresponding path.

The trajectory of a real tractor-semitrailer combination during a brake-in-turn test on snow is depicted in Fig. 4.2.b. The semitrailer is extensively braked at the marked point indicated by the blue asterisk, activating the slip controller and thereby limiting brake forces. After the braking point, the AHV experiences a trailer swing: the semitrailer loses its grip and starts to slide sideways, and the articulation angle increases undesirably.

Real tests show that trailer swing severity during braking may decrease over time as speed decreases, unlike jackknifing. Nevertheless, it is important to avoid excessive trailer swing before the trailer leaves its lane and potentially collides with vehicles in adjacent lanes by reducing braking or propulsion forces.

Trailers typically lack yaw-stability controllers, such as ESC, found in towing trucks and tractors, to prevent trailer swing. However, trailers are equipped with ABS, which helps prevent excessive wheel slip and thus reduces the risk of trailer swing.

4.3 Combination spin-out

In this study, combination spin-out is defined as the simultaneous loss of directional stability in both the towing and trailing units. Unlike jackknifing or trailer swing, the articulation angle may not grow significantly because both units have large heading-angle deviations from the intended direction of travel. A combination spin-out occurs when both the towing and trailing units lose their lateral tire

grip due to high combined slip and start to slide sideways. A typical combination spin-out of an AHV is illustrated in Fig. 4.3.a.

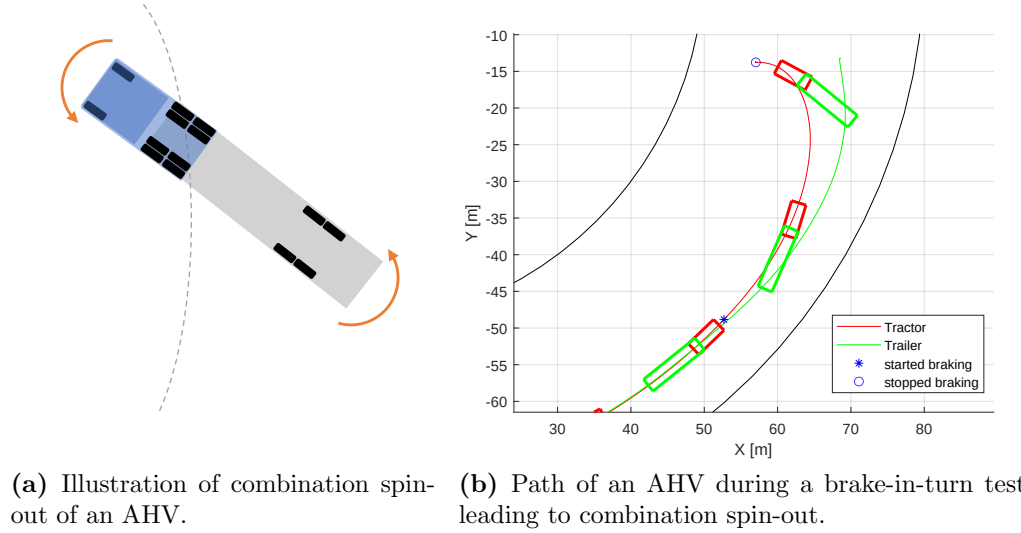


Figure 4.3: Illustration of combination spin-out and the corresponding path.

In Fig. 4.3.b, the trajectory of a real tractor-semitrailer combination during a brake-in-turn test on snow is illustrated. Both units are braked extensively at the point indicated by the blue asterisk. After the braking point, the AHV exhibits a combination spin-out, causing both units to lose their grip and slide sideways.

4.4 Trailer sway

Yaw instabilities explained so far are divergent instabilities in which one or more motion states (such as yaw rate) grow without oscillations. However, a vehicle's states may also exhibit oscillatory behavior with growing amplitude, which is called "dynamic instability". Trailer sway is a dynamic instability related to the combination's yaw dynamics.

Rearward Amplification (RA) is the ratio of the maximum lateral dynamics state observed in the first unit to that in the last unit of a combination. The state is typically lateral acceleration or yaw rate. RA is used to evaluate the risk of rollover or swing-out of the last unit relative to what the driver experiences [23]. The maximum RA allowed for vehicle combinations in Sweden is 2.4 [24]. RA is typically investigated using lane-change maneuvers or sinusoidal steering inputs.

Kharrazi et al. investigated the RA of yaw rate across many AHV combinations and reported RA values of 2.12 for the truck and double center-axle trailer,

1.84 for the A-double, 1.65 for both the Nordic combination and AB-double, and 1.08 for the tractor-semitrailer [25]. Consequently, the truck and double center-axle trailer is the most susceptible to dynamic instability.

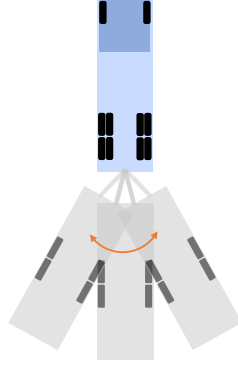


Figure 4.4: Illustration of trailer sway. The trailing unit (gray) continuously sways sideways around the coupling.

Evaluating dynamic stability using RA alone is not always sufficient. Yaw damping, defined as the damping of lateral excitation over time, decreases at higher speeds. If the yaw damping of a combination reaches zero at a certain high speed (defined as the zero-damping speed), lateral oscillations will not be damped over time, leading to self-excitation and continued growth. Combinations without a center-axle trailer normally have sufficient yaw damping [26]. Combinations with center-axle trailers, on the other hand, may become self-excited at certain speeds due to insufficient yaw damping caused by improper loading, coupling distances, or wheelbases. An example of this scenario, where the corresponding combination exceeds the zero-damping speed of 89 km/h, is depicted in Fig. 4.5 [26]. In this case, the units' yaw rates continue to increase after a lane change because yaw damping is insufficient.

Combinations exceeding the zero-damping speed continue to sway even after a small excitation, such as a lane change or side wind. This swaying continues unless it is mitigated by applying, e.g., a corrective yaw moment through differential braking or stretch braking. For passenger cars towing small trailers or caravans, there are several stability control systems that corrective actions may be applied by the towing unit (such as Toyota Trailer Sway Control system with differential braking, TSC [27]), or trailing unit (such as Bosch Trailer Safety Control system with differential braking, TSC [28], or AL-KO Trailer Control system with stretch braking, ATC [29]). If self-excitation is not mitigated promptly, trailer sway can lead to trailer rollover or an excessively large swept path, eventually causing the combination to fold, such as jackknifing [30].

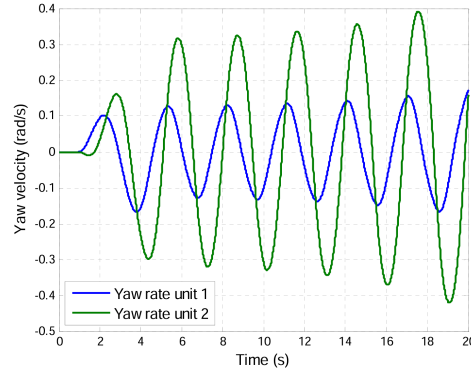


Figure 4.5: Self-excited yaw rate response for a 6x4 truck and center axle trailer [26].

4.5 Rollover

An important instability mode for heavy vehicles is rollover, which typically occurs in vehicle combinations with a high center of gravity, such as timber trailers, tankers, or concrete mixers. Rollover usually occurs during maneuvers involving high lateral accelerations, such as sharp turns or lane changes. However, it can also occur at a standstill under static conditions due to high bank angles or strong side winds.



Figure 4.6: A tractor-semitrailer combination rolling over during electric truck crash safety tests [31].

During dynamic maneuvers with high lateral accelerations, rollover typically starts at the trailers, causing the inner wheels to lift off. Because the trailers lack high roll stiffness due to their long wheelbases, the rollover occurs progressively, advancing from the rear of the combination toward the front and ultimately pulling the towing units into the rollover. Although rollover usually begins with the trailers due to their high center of gravity and mass, it may also first occur on the towing units with high centers of gravity, such as rigid trucks, especially

during clothoid maneuvers with a progressively increasing turning radius, since the towing units experience higher lateral acceleration than the trailing units. Fig. 4.6 illustrates a tractor-semitrailer combination experiencing rollover [31].

Modern trucks and trailers are often equipped with a Roll Stability Control (RSC) system. These controllers brake the combination to reduce speed, thereby reducing lateral acceleration. Some passenger cars are equipped with active rollover prevention systems that monitor the vehicle's state to prevent rollovers using active suspensions and anti-roll bars [32], or by braking a subset of the wheels to induce a loss of lateral grip, thereby reducing the lateral forces acting on the vehicle [33]. Additionally, further studies explore rollover prevention via active steering [34].

Chapter 5

Safe operating envelope

Traditional heavy vehicle combinations have internal combustion engines only in the towing unit, making it trivial to control the propulsion and braking forces. To propel the combination, the driver's request is received via the accelerator pedal, and a simple control strategy is applied to control engine torque. The service brake request, received via the brake pedal, is fulfilled by braking forces proportional to axle loads, which is the safest allocation for vehicle stability. When retarders are used, brake blending is performed between the engine's exhaust brake, retarder, and service brakes. With the electrification of both towing and trailing units in heavy vehicle combinations, novel control allocation strategies are needed to distribute propulsion and regenerative braking forces across the units. These strategies should minimize energy consumption, considering the efficiency maps of various electric motors and other power losses. Meanwhile, the controller should ensure the safe distribution of the actuator requests. If too much force is allocated to the towing unit, jackknifing may occur. If excessive force is allocated to the trailing unit, the trailer can swing. To ensure safe allocation, a Safe Operating Envelope (SOE), also known as a Maneuvering Limitation Diagram (MLD), safe maneuvering envelope, or safe set, can be utilized. It is defined as a set of operational limits and conditions on vehicle states and actuator requests for safe vehicle operation. A control allocation using the SOE increases safety and reduces the risk of instabilities such as jackknifing and trailer swing. Furthermore, the SOE is an important diagram for understanding vehicle combination instabilities and safety limits. An accurate vehicle dynamics model, such as a two-track model, can be used to generate a precise SOE. Once validated through high-fidelity simulations or real tests, this model becomes a suitable candidate for integration into a model-based control design, such as model predictive control.

5.1 Safe operating envelope in force domain

SOE is defined as a set of safe limits on actuator requests and formulated as a function of vehicle states (such as lateral acceleration) and environmental parameters (such as road friction coefficient). Conceptually, the SOE presented in this section is analogous to traditional brake force distribution diagrams for two-axle vehicles, but generalized to define safe limits for both braking and propulsion force distributions across multi-unit combinations. In **Paper A**, SOEs for both propulsion and braking are obtained as a function of normalized lateral acceleration, c_y , which is defined as:

$$c_y = \frac{a_y}{\mu g} , \quad (5.1)$$

where a_y is the lateral acceleration of the towing unit, μ is the friction coefficient, and g is the gravitational acceleration.

For each c_y , hence for different combinations of road frictions and lateral accelerations, a 2-dimensional SOE is obtained by simulating brake-in-turn or propel-in-turn maneuvers with various combinations of tractor and semitrailer actuator forces. In the simulations, a single-track vehicle model is employed. Tire forces are limited with respect to the corresponding friction circles [35]. The applied propulsion or braking forces, F_{1rx} for the tractor drive axle and F_{2x} for the semitrailer's axle group, are defined as:

$$\begin{cases} F_{1rx} &= c_{tractor} \mu F_{1rz} , \\ F_{2x} &= c_{trailer} \mu F_{2z} , \end{cases} \quad (5.2)$$

where F_{1rz} and F_{2z} are the normal loads on the tractor's drive axle and the semitrailer's axle group, respectively. $c_{tractor}$ and $c_{trailer}$ denote the friction utilization coefficients for the tractor and semitrailer, respectively. By varying these coefficients in the interval of $[-1,0]$ for braking and $[0,+1]$ for propulsion, each maneuver is classified as safe or unsafe. For defining a safe maneuver, the following criteria are used:

$$\max(\Delta\alpha_{1r}) < 5^\circ \ \& \ \max(\Delta\alpha_2) < 3^\circ , \quad (5.3)$$

where $\max(\Delta\alpha_{1r})$ denotes the maximum deviation of the side-slip angle of the tractor's drive axle from its quasi-steady-state value, and $\max(\Delta\alpha_2)$ represents the same for the semitrailer axle group. Quasi-steady-state side-slip angles are the values of these states just before the application of actuator forces, describing a well-behaved and stable combination. For a semitrailer with a length of 10 m, a limit of 3° side-slip angle deviation results in approximately 0.5 m lateral displacement. This limit is chosen to prevent excessive lateral displacement, which could cause the semitrailer to leave its lane and collide with other vehicles in

adjacent lanes. For the tractor, a larger limit of 5° is chosen because towing units are shorter than trailing units and can tolerate greater side-slip angle deviations without leaving the lane.

A generic 2-dimensional SOE for a specific c_y is shown in Fig. 5.1 for a braking tractor-semitrailer combination. Typically, trailer swing occurs above a certain $c_{trailer}$ level. This limiting $c_{trailer}$ correlates weakly with tractor forces, meaning trailer swing occurs almost regardless of how much the tractor is braking or propelling. Similarly, beyond a certain level of $c_{tractor}$, jackknifing occurs. Jackknifing, unlike trailer swing, is more probable when the semitrailer is not braked, which is reflected by the inclined limit for jackknifing in Fig. 5.1. Therefore, braking the semitrailer helps prevent jackknifing to a certain extent by introducing a corrective yaw moment on the towing unit. This feature, known as “stretch braking”, is applied in heavy vehicle combinations traveling downhill slopes by braking the trailer to enhance vehicle safety. In propulsion, on the other hand, applying less propulsion to the semitrailer helps avoid tractor jackknifing. The common characteristic of the two cases is that a higher tension force in the coupling enhances the towing unit’s yaw stability.

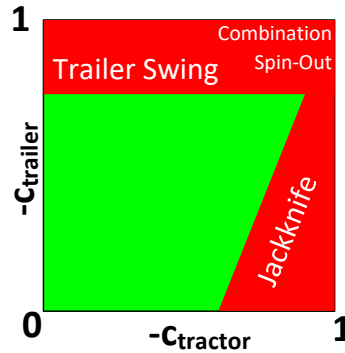


Figure 5.1: Different yaw instability modes shown on the SOE for a braking vehicle combination. From Paper A.

For combinations of the larger (in the absolute sense) values of $c_{tractor}$ and $c_{trailer}$, a combination spin-out happens, indicating instability in both the towing and trailing units.

In **Paper B**, a high-fidelity model referred to as the Volvo Transport Model (VTM) [36] is employed to derive the SOE of a tractor-semitrailer combination. This simulation model also incorporates a simple slip controller to keep wheel slips under control. Numerous simulations are conducted for a brake-in-turn maneuver to obtain an SOE, as shown in Fig. 5.2.

Simulations are conducted for four different c_y values, and a 2-dimensional SOE is obtained for each c_y as depicted in Fig. 5.2a. At lower lateral accelera-

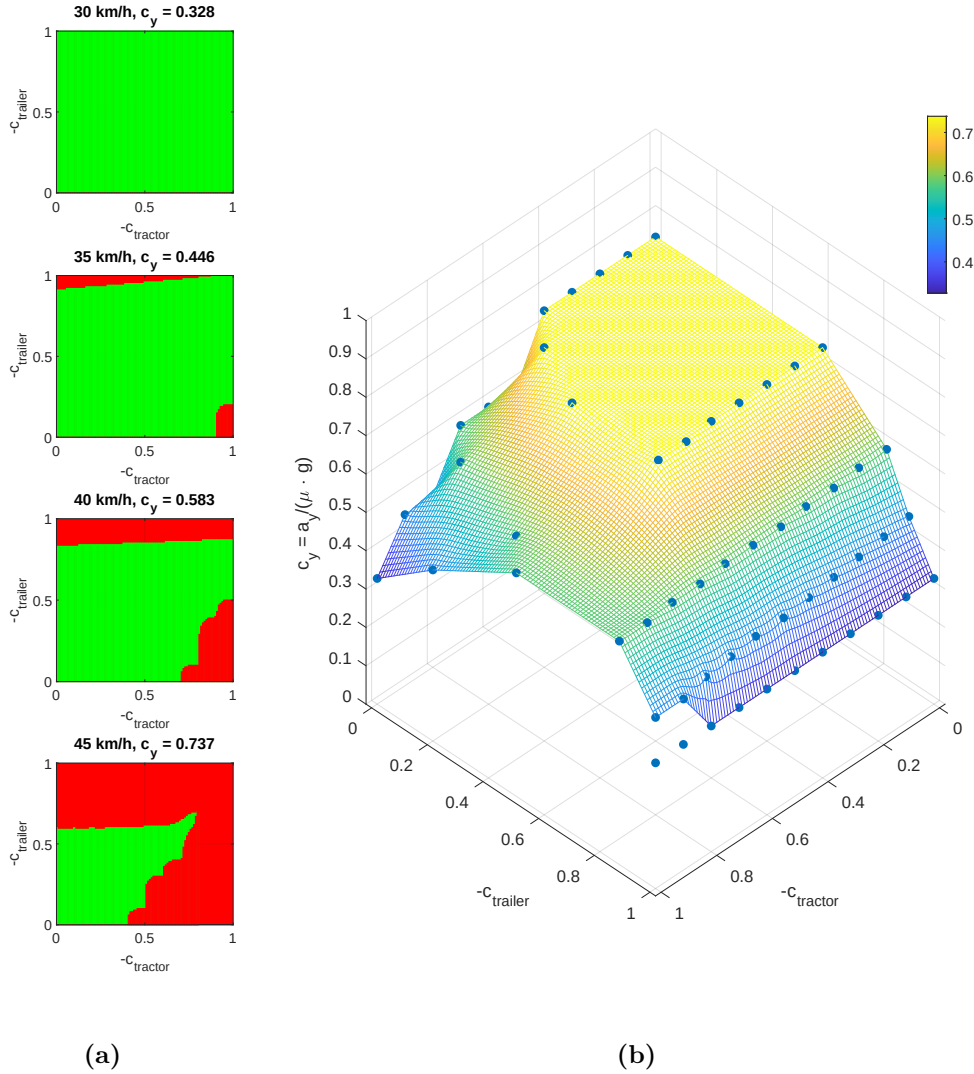


Figure 5.2: SOE obtained from the high-fidelity simulations for a tractor-semitrailer combination braking in turn. Adapted from Paper B.

tions, all combinations of tractor and semitrailer braking forces are safe (shown in green). At higher lateral accelerations, however, excessive braking forces lead to yaw instabilities, such as jackknifing and trailer swing (shown in red). The 2-dimensional SOEs are used to derive a 3-dimensional SOE, as presented in Fig. 5.2b, where the vertical axis represents c_y . Envelopes for the intermediate c_y values that are not simulated are interpolated, yielding a 3-dimensional surface. The maximum c_y value simulated is 0.737, corresponding to the highest speed (45 km/h) that the combination can reach for the simulated turning radius of 72 m,

friction coefficient of 0.3, and loading conditions. Any combination of c_y , $c_{tractor}$, and $c_{trailer}$ values under the obtained SOE surface results in a stable maneuver, and any combination of those above the SOE surface results in an unstable maneuver, leading to jackknifing, trailer swing, or combination spin-out. The SOE obtained with high-fidelity model simulations is validated with real vehicle tests.

In **Paper C**, an alternative two-track model with roll dynamics is presented to obtain an SOE. This model also incorporates a friction-circle-inspired combined slip model, which is lacking in the single-track model presented in **Paper A**. SOEs obtained with the single-track and two-track models are compared with the SOE obtained from high-fidelity simulations. All SOEs for four different speeds (and c_y values) with three different models are shown in Fig. 5.3.

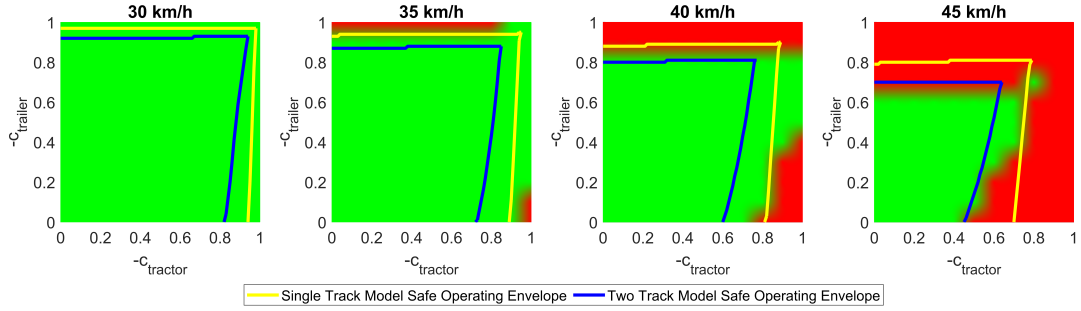


Figure 5.3: SOEs obtained with single-track model (yellow line), two-track model (blue line) and high-fidelity VTM model with slip controller (red/green background). From Paper C.

Validations against high-fidelity simulations show that a two-track model with roll dynamics, lateral load transfer, and combined slip outperforms the single-track model in SOE accuracy. Therefore, the combined slip model for simulating large tire forces near tire limits and the inclusion of lateral load transfer under higher lateral accelerations are important when deriving an SOE. The SOE obtained from the two-track model closely matches that from the VTM without a slip controller (a more appropriate comparison than the VTM with a slip controller, since the two-track model also lacks a slip controller). In Fig. 5.3, as the VTM is simulated with a slip controller (which is a more realistic scenario since modern vehicles are equipped with ABS), the SOE obtained with the two-track model is smaller at lower lateral accelerations compared to the one obtained with the VTM. However, this is because the slip controller in the high-fidelity model reduces brake forces while leaving sufficient lateral force capability to maintain yaw stability. The two-track model, on the other hand, has no slip controller, but the lateral tire forces are limited by the friction circle, resulting in reduced lateral force capability for maintaining yaw stability. Nevertheless, the SOE obtained with the two-track model is more conservative and is mostly smaller than the SOE obtained with the

VTM, as shown in Fig. 5.3. Thus, using this SOE for vehicle control would result in safe control allocation.

The SOE obtained from any of these methods can serve as inequality constraints in the control allocation formulation as demonstrated in Section 2.5. Alternatively, the two-track model, validated against a high-fidelity model, can be used in a model-based control framework, such as Model Predictive Control (MPC), with additional safety constraints on critical vehicle states (such as yaw rate or side-slip angle) to ensure stability.

5.2 Safe operating envelope in slip domain

As modern electric vehicles have interfaces not only to control motor torque but also to limit or control motor speed (and thus longitudinal tire slip), the SOE can be extended to cover longitudinal slip. In Fig. 5.4, lateral tire forces as a function of longitudinal slip for a number of constant side-slip angles are depicted [37]. For larger side-slip angles, the tires generate significantly less lateral force when longitudinal slip is high. Hence, it is important to limit longitudinal slip in electrically driven wheels when a large lateral force is required to maintain yaw stability at high lateral accelerations, especially on low-friction surfaces.

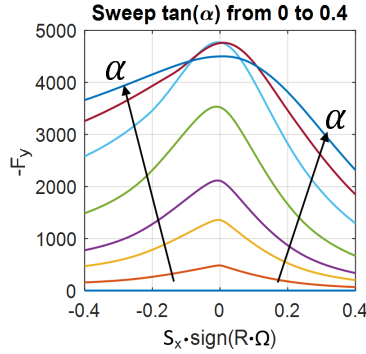


Figure 5.4: Lateral force F_y as a function of longitudinal slip S_x for a number of constant side-slip angles α . Adapted from [37].

In **Paper D**, an SOE in the longitudinal slip limit versus side-slip angle domain is presented, referred to as “slip polytope”. In Fig. 5.5, the adaptive longitudinal slip limit, S_x^{lim} , is shown as a function of the deviation of the side-slip angle α from the reference side-slip angle α^{ref} . α^{ref} is estimated via a vehicle dynamics model for quasi-steady-state conditions. Therefore, α^{ref} represents the side-slip angle of the axle for the specific speed and turning radius, for zero longitudinal and yaw acceleration, and with zero longitudinal forces on the wheels (no braking or propulsion). If the measured or estimated side-slip angle of the axle, α , is equal

to α^{ref} , then positive and negative slip limits for the unit's electric motors are set as the default limits $S_x^{lim,prop}$ and $S_x^{lim,brake}$, respectively. These can typically be in the range of $\pm 5\%$ to $\pm 15\%$, depending on the conditions. Any deviation of the side-slip angle α from the reference side-slip angle α^{ref} results in a reduction in the longitudinal slip limit S_x^{lim} . A side-slip angle deviation of $\Delta\alpha$ reduces S_x^{lim} to 0, indicating that at this level of side-slip angle deviation, no longitudinal force can be applied by the actuators, and all the tire force capability is exclusively reserved for the lateral direction to maintain yaw stability. This slip envelope is applied to the electric motors of the units, but not to the service brakes. This means that service brakes are always available to slow down the vehicle. By imposing slip polytope constraints, excessive regenerative braking with electric axles or any propulsion that leads to yaw instabilities can be avoided.

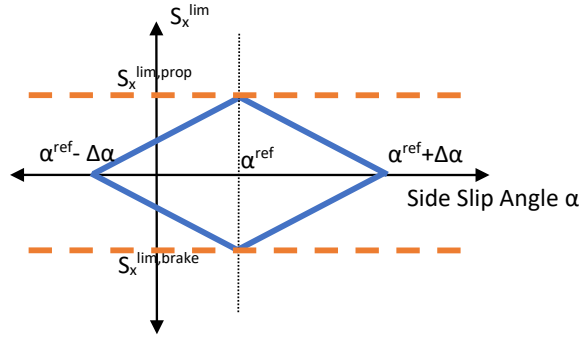


Figure 5.5: Longitudinal slip limit S_x^{lim} represented as a function of side-slip angle α deviation from the reference side-slip angle α^{ref} . Adapted from Paper D.

While a simple linear decrease in the slip limits provides a practical baseline, a more precise envelope can be implemented. This concept can also be extended to higher dimensions by incorporating additional states, such as yaw rate. To systematically construct these higher-dimensional boundaries, state-space analysis is commonly employed. Using this approach, phase-plane trajectories of the vehicles can be generated for specific operating conditions, parameterized by factors such as road friction and payload. These phase-plane trajectories can be plotted for different state pairs, such as side-slip angle versus side-slip angle rate [38], lateral velocity versus yaw rate [39], or side-slip angle versus yaw rate [40].

In Fig. 5.6, a phase-plane trajectory for a coach is shown [38], focusing on side-slip angle and side-slip angle rate. The upper and lower boundaries for vehicle stability are shown in red. The domain of attraction lies between these red lines, where the vehicle is stabilized over time. Outside these boundaries, on the other hand, the domain of repulsion occurs, where the vehicle's side-slip angle increases, eventually causing loss of grip and vehicle instability.

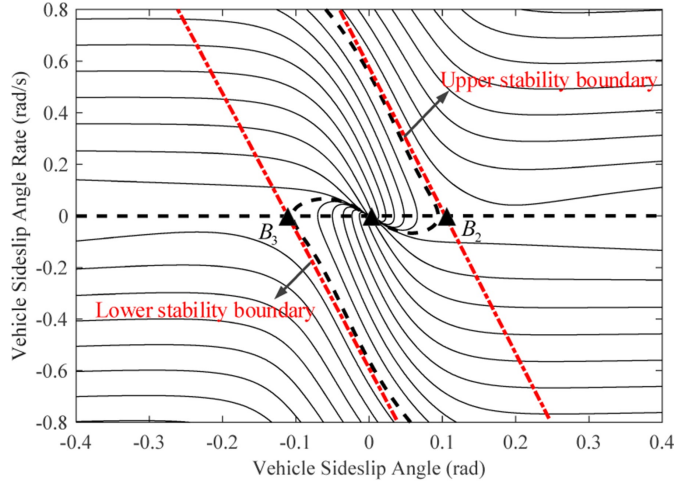


Figure 5.6: Vehicle side-slip angle versus side-slip angle rate phase-plane trajectory for a coach [38] .

Extending these envelopes to multi-unit vehicles is possible, though it involves greater complexity than single-unit configurations. This complexity arises from the high-dimensional dynamics of articulated vehicles, where the tractor and trailer states are tightly coupled. Consequently, generating and analyzing phase portraits across such a multi-dimensional space becomes both computationally demanding and difficult to visualize.

In **Paper E**, a 3-dimensional phase portrait for a tractor-semitrailer combination is presented within the domain of tractor side-slip angle, tractor yaw rate, and trailer yaw rate, as shown in Fig. 5.7. For this analysis, two states (articulation angle and longitudinal velocity) and all inputs (steering angle and wheel slips) are held constant. While maintaining a constant longitudinal velocity is a standard simplification in phase portrait analysis, the articulation angle inherently changes during any actual dynamic maneuver. Consequently, as the physical system progresses in time, the true trajectory will deviate from the depicted streamlines and exit this fixed subspace. Fixing these two states, however, remains a useful method to visualize the instantaneous vector field and the local region of attraction. Therefore, the streamlines illustrate the vector field's shape rather than the system's true time-domain trajectories.

While 3-dimensional phase portraits offer a comprehensive overview, their complexity makes interpretation and automated boundary extraction challenging. Consequently, a reduction in dimensionality through 2-dimensional projections is introduced. The second and third projections in Fig. 5.7 demonstrate that coupled projections—those combining states from both units—exhibit a strong dependency on the omitted third state. Thus, the resulting 2-dimensional portraits

are misrepresentative and difficult to interpret. Conversely, the first projection involving single-unit states (e.g., tractor side-slip angle versus tractor yaw rate) remains highly consistent regardless of the adjacent unit's yaw rate. This decoupling characteristic motivates a dimensionality-reduction strategy that yields two separate phase portraits: one for the tractor and one for the semitrailer.

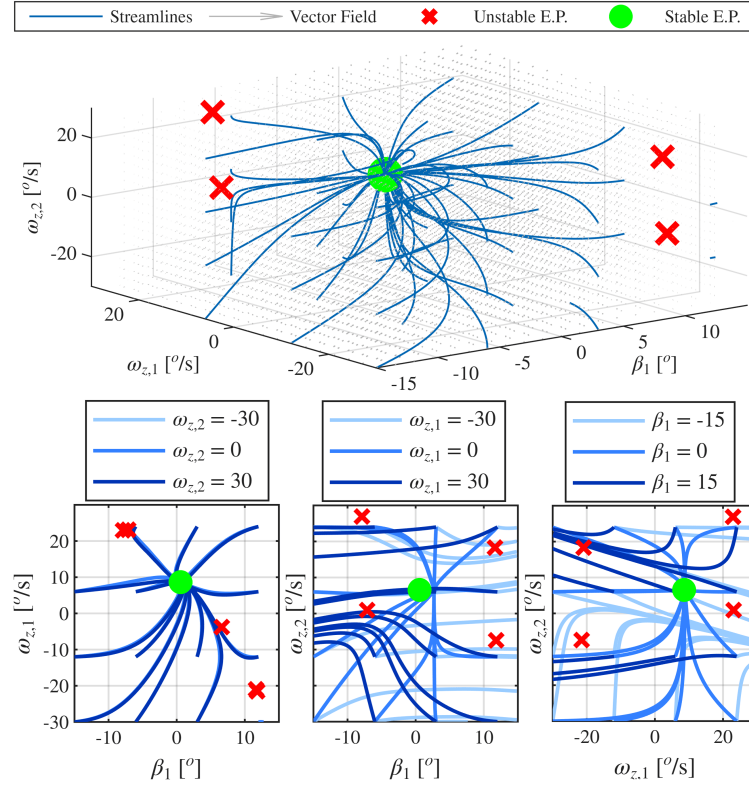


Figure 5.7: 3D phase portrait in the $(\beta_1, \omega_{z,1}, \omega_{z,2})$ state space and its three 2D projections for a tractor-semitrailer combination. From Paper E.

Kinematic relationships under steady-state turning conditions are used to isolate two distinct phase portraits while preserving the units' underlying physical coupling. Under steady-state turning conditions, the side-slip angle versus yaw rate envelope for one unit is generated by varying its states while keeping the coupled unit's states constant. The SOEs are established by evaluating both the system's theoretical stability and its operational feasibility, with the latter verified by overlaying the lateral slip limits corresponding to peak lateral forces. The SOEs obtained for the tractor and semitrailer are shown in Fig. 5.8.

2-dimensional SOEs, generated in the side-slip angle versus yaw rate plane for various longitudinal slips, can be stacked along a third axis to construct a comprehensive 3-dimensional SOE, as shown in Fig. 5.9. While not explored

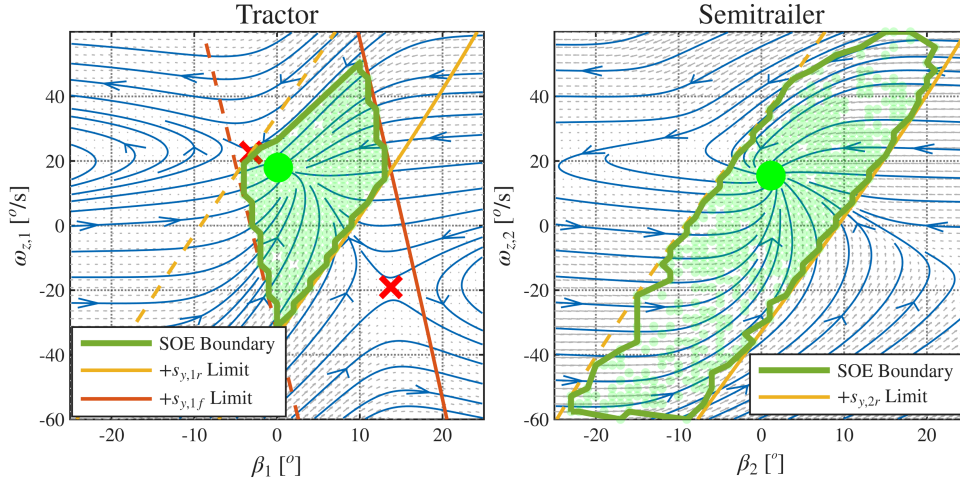


Figure 5.8: 2D phase portraits and SOEs in the (β, ω_z) state space for a tractor and semitrailer. Adapted from Paper E.

further in this PhD thesis, extracting adaptive longitudinal slip limits from these multi-dimensional envelopes facilitates stable vehicle operation [41].

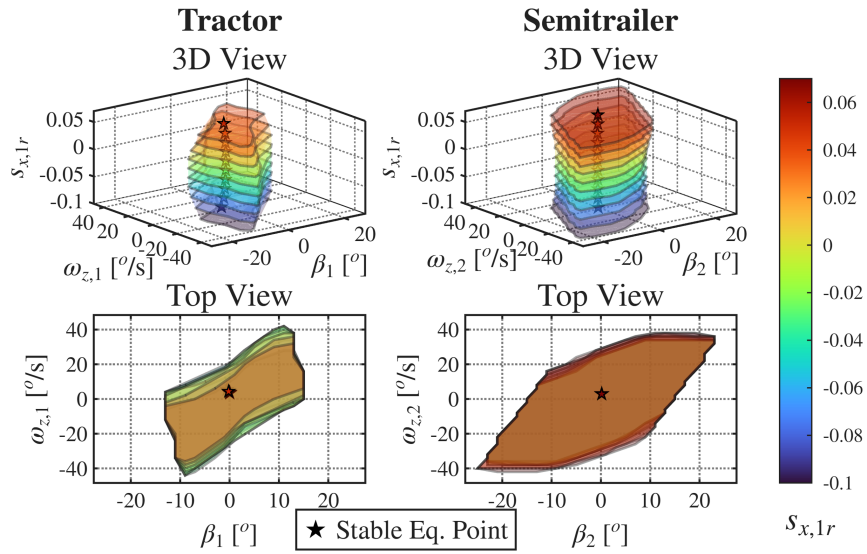


Figure 5.9: SOEs with varying longitudinal slip on tractor rear axle $s_{x,lr}$ (-0.10 to $+0.06$). Adapted from Paper E.

Chapter 6

Safe and energy-efficient control allocation

Driven by the transport industry’s shift toward electrification, multi-unit electrified powertrains for heavy commercial vehicles have become a major focal point. Notably, trailer electrification is rapidly emerging as a priority, with manufacturers actively developing these advanced systems. By equipping both the towing vehicle and its connected trailers with dedicated Electric Motors (EMs) and battery packs, these distributed architectures offer substantial performance advantages. However, coordinating the energy flow between physically separated units remains a complex challenge. Fully realizing their potential requires in-depth research into optimal control logic and advanced energy management frameworks.

To optimize energy efficiency, applying regenerative braking on a single axle or unit is strongly justified—unlike traditional service braking, which is distributed across all axles and units in proportion to their normal loads. Furthermore, electrification of trailing units enables trailer-only propulsion, which inherently carries a higher risk of instability than towing-unit propulsion. Therefore, allocating propulsion and regenerative braking forces across the axles and units of heavy combinations must carefully balance energy efficiency with stability. While Electronic Stability Control (ESC) systems can mitigate certain yaw instabilities, the underlying Control Allocator (CA) must proactively request a safe force distribution to avoid inducing jackknifing or trailer swing. This preventive safety margin is especially critical on low-friction, high-curvature roads where ESC interventions may be insufficient.

This chapter first presents an energy-efficient control allocation strategy focused on power loss minimization, evaluating how balancing battery levels and selectively disengaging propulsion systems across multiple units affect energy consumption. Subsequently, various approaches for safe control allocation are detailed. While the broader vehicle architecture may include supplementary safety

algorithms that request differential braking (such as ESC), the combination CA proposed in this chapter exclusively distributes longitudinal forces across the different units. It does not allocate lateral forces or yaw moments; instead, its primary aim is to manage these longitudinal forces safely without inducing yaw instability.

6.1 Energy-efficient control allocation with power loss minimization

In the context of energy-efficient control allocation, Power Loss Minimization (PLM) [20, 21] has received considerable attention. In **Paper F**, PLM is applied to an electric tractor and electric semitrailer combination vehicle.

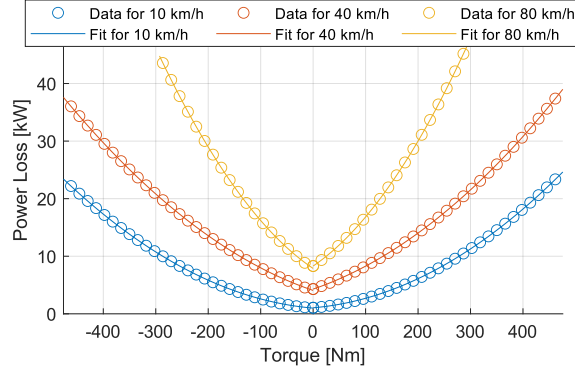


Figure 6.1: Battery-to-wheel power losses as a function of motor torque for an induction motor. From Paper F.

Battery-to-wheel Power Losses (PLs) in a simulated heavy vehicle, including battery, inverter, EM, and gear losses, shown in Fig. 6.1, can be approximated as a quadratic polynomial of the EM torque (T):

$$\begin{aligned}
 P_{L,EM,i} &\approx a_i T_{EM,i}^2 + b_i T_{EM,i} + c_i, \\
 \text{where } i &\in \{11, 12, 2\}, \\
 P_{L,SB,j} &= -\omega_j T_{SB,j}, \\
 \text{where } j &\in \{1, 2\}.
 \end{aligned} \tag{6.1}$$

In this example, tractor has two EMs (EM_{11} and EM_{12}), and semitrailer has one (EM_2). PL of Service Brakes (SBs), on the other hand, is formulated as the product of wheel rotational speed, ω_j , and the applied brake torque, $T_{SB,j}$ ($j = 1$ for the tractor and $j = 2$ for the semitrailer), implying that SBs dissipate all brak-

ing energy as heat. Total PLs can be minimized with a quadratic programming algorithm:

$$\begin{aligned} \mathbf{u} = \arg \min_{\mathbf{u}} & \left(\sum_i P_{L,EM,i}(\mathbf{u}) + \sum_j P_{L,SB,j}(\mathbf{u}) \right), \\ \text{s.t. } & \mathbf{B} \mathbf{u} = \mathbf{v}, \quad \underline{\mathbf{u}} \leq \mathbf{u} \leq \bar{\mathbf{u}}. \end{aligned} \quad (6.2)$$

Here, the input vector, $\mathbf{u}$, consists of allocated EM and SB forces to the actuators. The equality constraint ensures that the allocated actuator torques fulfill the total longitudinal force request. The inequality constraints ensure the allocation stays within actuator capabilities.

Quadratic program (6.2) minimizes PLs of all actuators across the combination in a single formulation and allocates the actuator forces. Hence, it is a centralized CA, as shown in Fig. 6.2.a. A primary limitation of centralized CA is its reliance on direct access to low-level actuator interfaces and specific PL parameters. In many real-world applications—such as articulated vehicles utilizing electrified trailers from diverse manufacturers—this level of system transparency may be unavailable. Consequently, a hierarchical control architecture is necessary. In this approach, an upper-level controller manages force demands at the combination level, while the lower-level allocation among multiple EMs is delegated to the local controllers within each unit. Such a distributed CA is shown in Fig. 6.2.b: A combination CA allocates unit-wise EM and SB forces into each unit, and unit CAs can allocate actuator forces to each actuator. The combination CA, in this case, receives the lumped PL parameters a_i , b_i , c_i (given in (6.1)) from the unit CAs.

Paper F evaluates centralized and distributed PLM against a baseline rule-based CA strategy that distributes EM forces equally across the vehicle units. It also investigates energy savings from selectively disengaging EMs to eliminate idle losses and introduces a State-of-Charge (SOC) control strategy to balance battery depletion rates. The results indicate that PLM reduces energy consumption by about 1% compared to the rule-based approach, with centralized PLM offering only marginal energy benefits of about 0.2% over the distributed architecture. Conversely, selectively disengaging EMs yields substantially greater energy savings of up to 5% by mitigating idle losses. The study also highlights that CA without SOC management disproportionately depletes the more efficient unit's battery. Implementing the proposed SOC strategy synchronizes depletion rates while incurring an energy consumption penalty of up to 1%. However, because this penalty is comparable to the overhead of towing a conventional trailer, the overall system remains highly efficient. Ultimately, given the substantial benefits—such as extended range, superior startability, and improved gradeability—the findings demonstrate that electrified trailers can be utilized without a net energy penalty

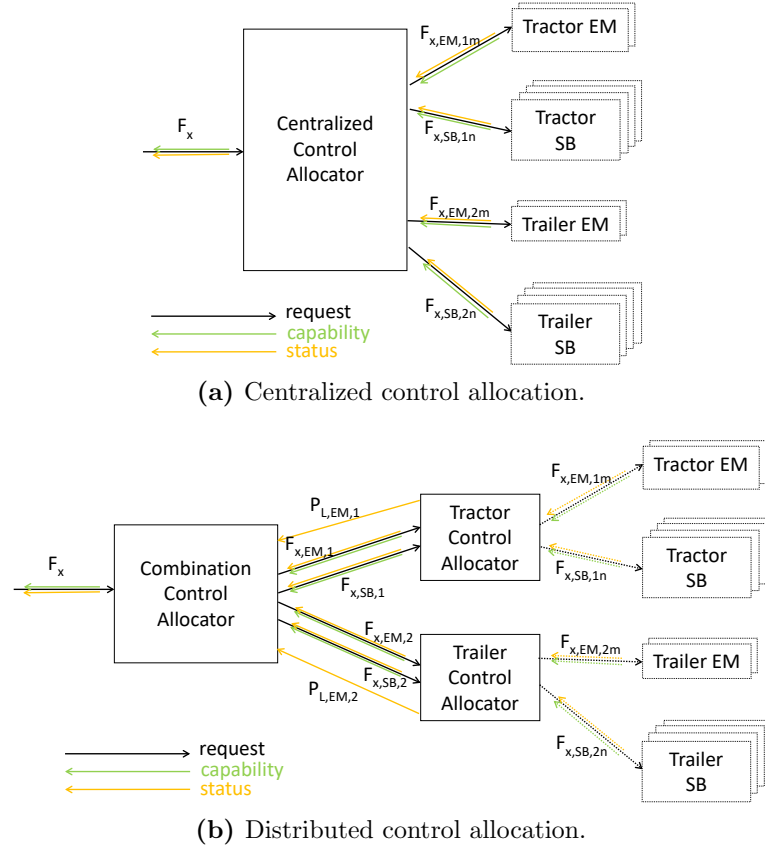


Figure 6.2: Centralized and distributed control allocation functional architecture and signal interfaces. Adapted from Paper G.

compared to traditional combinations.

In **Paper G**, PLM is tested within a real-time controller on a physical tractor-semitrailer combination. Because only the tractor is electrified, the optimization problem is trivial, routing all propulsion and regenerative braking requests solely to the tractor. However, real brake-in-turn tests conducted on a circular frozen lake track at the Colmis Proving Ground [42] reveal a critical limitation: utilizing PLM in CA without additional safety constraints can induce jackknifing on low-friction, high-curvature roads. Therefore, integrating robust safety mechanisms is vital to guarantee the yaw stability of the articulated vehicle. The following sections explain potential safety mechanisms.

6.2 Safety constraints in force domain

As demonstrated by the real vehicle test results in Fig. 6.3, a brake-in-turn maneuver on a low-friction surface leads to jackknifing when braking is applied to the tractor's drive axle without safety constraints. To address this, this section presents two stability control methods from **Paper G** that can be integrated within the quadratic program (6.2) introduced in Section 6.1.

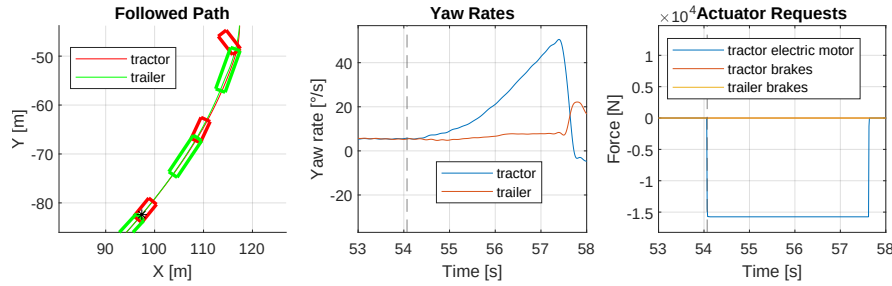


Figure 6.3: State plots for a tractor-semitrailer combination during a brake-in-turn maneuver on a low-friction surface. Unconstrained braking on the tractor's drive axle leads to jackknifing. From Paper G.

6.2.1 Safe operating envelope

In **Paper G**, a Safe Operating Envelope (SOE), presented in Section 5.1, is used with the PLM algorithm for brake-in-turn tests. SOE is introduced as an inequality constraint, given in (2.5), on the quadratic program (6.2) minimizing PLs.

When the tractor drive axles' friction utilization, $c_{tractor}$, reached the boundaries defined by the SOE, the total brake force request was met by applying additional braking on the semitrailer using its SBs, as shown in Fig. 6.4. After slowing down through braking, lateral acceleration decreased. Because the SOE was larger at lower lateral accelerations, the amount of regenerative braking from the tractor's EMs increased while the combination continued braking. Consequently, the brake-in-turn maneuver was completed, and the combination came to a safe standstill without jackknifing, owing to the SOE.

6.2.2 Limiting the longitudinal force capabilities based on lateral force estimates

In **Paper G**, a lateral tire force estimator is presented. These estimated forces are then used to limit the longitudinal force capabilities of each unit based on

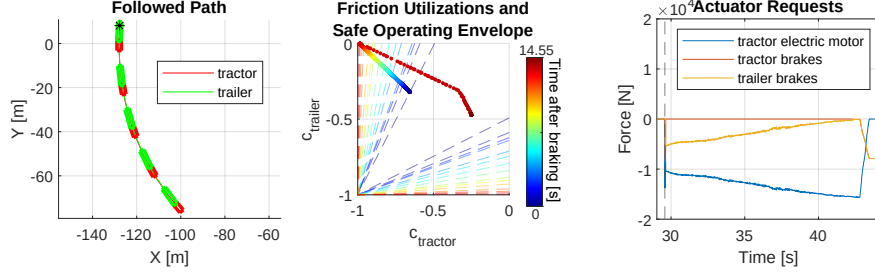


Figure 6.4: State plots for a tractor-semitrailer combination during a brake-in-turn maneuver on a low-friction surface. The SOE ensures stable braking on the tractor’s drive axle by redistributing braking forces to the semitrailer. From Paper G.

the friction circle concept [35]. Longitudinal force limits are used as actuator capabilities, $\underline{u}$ and $\bar{u}$, in the quadratic program (6.2) minimizing PLs.

As a result, a smaller longitudinal actuator force is allowed at higher lateral accelerations, since most of the tire force capability is used in the lateral direction to keep the combination stable. In brake-in-turn tests, this approach ensured safe braking and coming to a standstill without the risk of jackknifing, as shown in Fig. 6.5. As the longitudinal force capability of the tractor’s electric axle is reduced, the total braking force is supplied mainly by additional braking via semitrailer SBs, with a small contribution from tractor SBs.

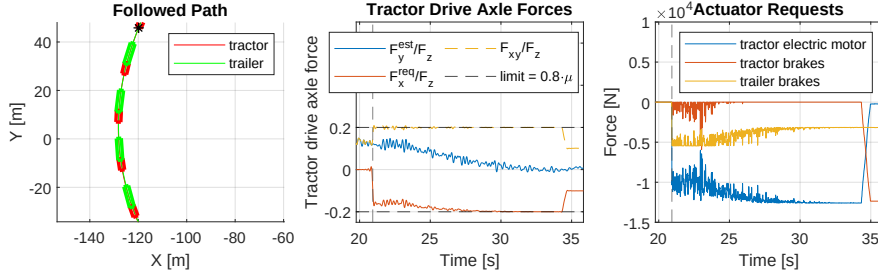


Figure 6.5: State plots for a tractor-semitrailer combination during a brake-in-turn maneuver on a low-friction surface. Redistributing the braking forces to the tractor and semitrailer SBs ensures stable braking on the tractor’s drive axle. From Paper G.

6.3 Adaptive longitudinal slip limit as a function of side-slip angle

An alternative approach to ensuring yaw stability is to limit wheel slips rather than EM forces, as discussed in Section 5.2. In **Paper D**, a slip polytope is first introduced and subsequently evaluated using a high-fidelity simulation model of a

6.3. Adaptive longitudinal slip limit as a function of side-slip angle 53

tractor-semitrailer combination during propel-in-turn maneuvers on low-friction surfaces. A quasi-steady-state side-slip angle for the tractor is estimated using a single-track model. Then, the longitudinal slip limit for the slip controller is reduced proportionally to the deviations from this reference angle. The results demonstrate that a vehicle combination utilizing such an adaptive longitudinal slip limit can accelerate to higher speeds on low-friction surfaces more stably than one with a constant slip limit. While the combination with a fixed slip limit jackknifed, the one with an adaptive limit maintained small side-slip angles.

The same scenario was also tested with a real vehicle combination on a circular ice track, where the driver was instructed to steer the vehicle to maintain directional stability. Under constant slip limits, the vehicle repeatedly slid sideways, leading to high side-slip angles, as shown in Fig. 6.6. To regain control, the driver counter-steered significantly, adjusting the steering angle to compensate for the tractor's side-slip. Consequently, with a constant slip limit, the vehicle could not accelerate to higher speeds because it lost traction.

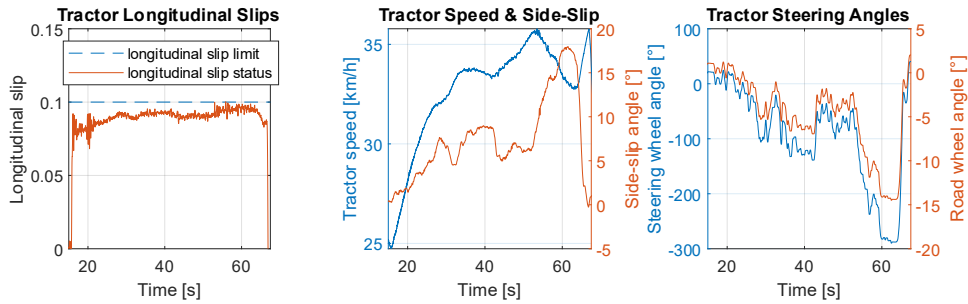


Figure 6.6: State plots for tractor-semitrailer combination during a propel-in-turn test with fixed slip limits. From Paper D.

With an adaptive slip limit, as shown in Fig. 6.7, the vehicle combination accelerated smoothly to higher speeds while maintaining stability and requiring minimal steering effort.

Fig. 6.8 depicts the longitudinal slip versus side-slip angle scatter plot for slip controllers with both fixed and adaptive slip limits. While the fixed limits produced significantly larger side-slip angles, the adaptive strategy constrained longitudinal slip and side-slip angle within the predefined slip polytope to ensure stable operation. In the left-hand turning maneuver presented in this study, the right-hand side of the slip polytope limited the slip values (due to the sign convention). If a right-hand turning maneuver were to be performed, the reference side-slip angle would be negative, and thus the left-hand side of the slip polytope would act as the limiting envelope.

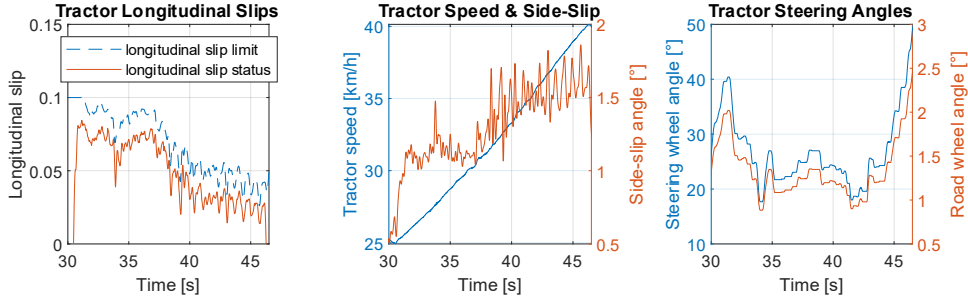


Figure 6.7: State plots for tractor-semitrailer combination during a propel-in-turn test with adaptive slip limits. From Paper D.

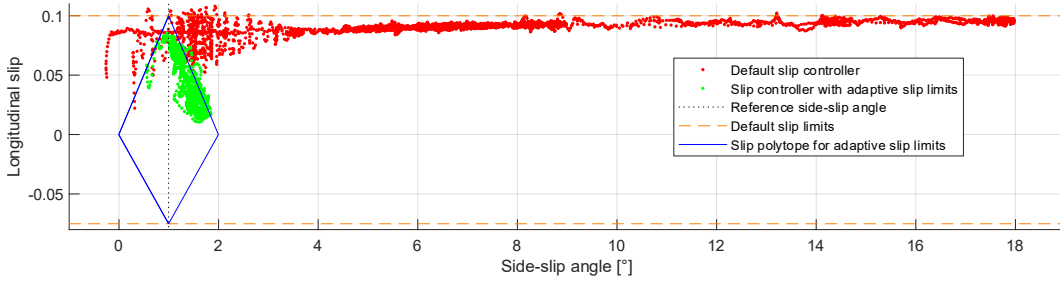


Figure 6.8: Longitudinal slip versus side-slip angle scatter plot for slip controllers with fixed and adaptive slip limits. From Paper D.

6.4 Predictive safety barriers

This section presents a hierarchical control allocation architecture that effectively decouples energy efficiency from vehicle safety. In this framework, a higher-level energy-efficient CA distributes the required actuator forces to optimize energy efficiency. Complementing this, a lower-level safety barrier monitors and transmits the actuators' current capabilities to the CA, dynamically updating them as operational constraints change. Under normal driving conditions—which constitute the majority of operation—the safety barrier simply passes through these optimal requests. However, during safety-critical situations, it acts as a fallback controller, overriding or modifying the distributed actuator forces to ensure vehicle stability. Utilizing a predictive formulation, the controllers in this section predict future states to proactively prevent yaw instabilities induced by suboptimal wheel force distribution.

Fig. 6.9 illustrates the functional architecture of the predictive safety barriers studied in **Paper H**. The three controllers shown are Model Predictive Control (MPC), Model Predictive Heuristic Control (MPHC), and Lateral Acceleration Proportional Predictive Control (LAPPC). Although they are grouped together

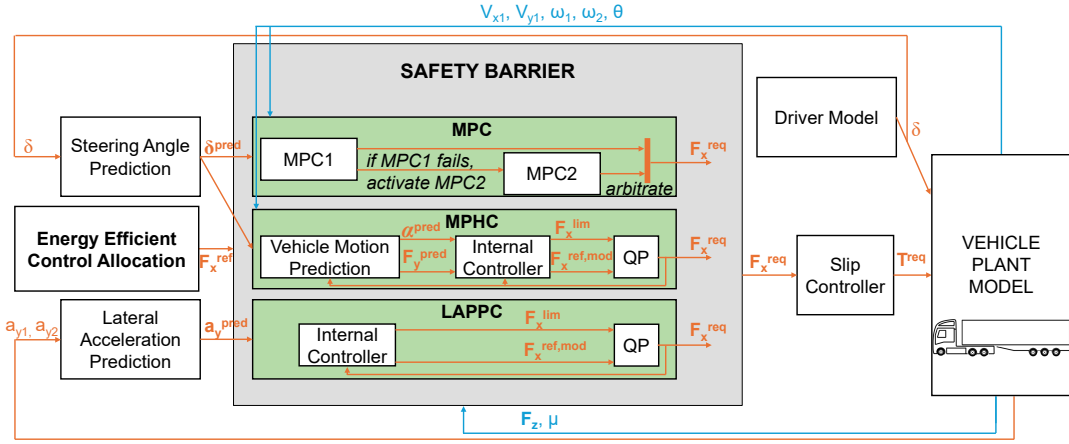


Figure 6.9: Functional architecture of the predictive safety barriers. Adapted from Paper H.

under the safety barrier function, only one operates at a time.

All controllers receive the reference longitudinal axle force distribution vector, $\mathbf{F}_x^{\text{ref}}$, from an upper-layer energy-efficient CA. The active controller subsequently sends modified or original longitudinal force requests, $\mathbf{F}_x^{\text{req}}$, to the slip controller. The slip controller, in turn, sends the actuator torque requests, $\mathbf{T}^{\text{req}}$, to the vehicle plant model.

Furthermore, all controllers have access to the axle normal loads, $\mathbf{F}_z$, and the road friction, μ . Finally, a driver model utilizes pure-pursuit control [43] with Proportional-Integral (PI) corrections—based on lateral deviation from the reference trajectory—to request the steering angle, δ .

The performance of the controllers is evaluated using sine-with-dwell (as defined in ISO 18375:2016 [44]) and highway-exit maneuvers. Fig. 6.10 presents the results for the sine-with-dwell test at an initial speed of 70 km/h, where the tractor’s drive axle is braked at a reference force of $F_x^{\text{ref}} = 0.6\mu_x F_z$. Sine-with-dwell tests involve open-loop sinusoidal steering at 0.2 Hz, with a 1 s dwell at the second peak. Steering angle amplitude varied from 1° to 15° in 1° increments across tests. Each test point shown in Fig. 6.10 represents the outcome of a complete test maneuver. Each simulation is terminated as soon as one of three conditions is met: the elapsed time reaches 18 s, the speed of either the tractor or the semitrailer falls below 10 km/h, or the articulation angle exceeds 88° (which denotes a severe jackknifing or trailer swing event).

Alongside the MPC, MPHC, and LAPPC strategies, an F_z proportional braking case is evaluated as a baseline, distributing the same total reference brake force, F_x^{ref} , in proportion to the axle loads. Finally, a Lateral Acceleration Proportional Instantaneous Control (LAPIC) approach is included to investigate the performance benefits of predictive formulations over instantaneous control.

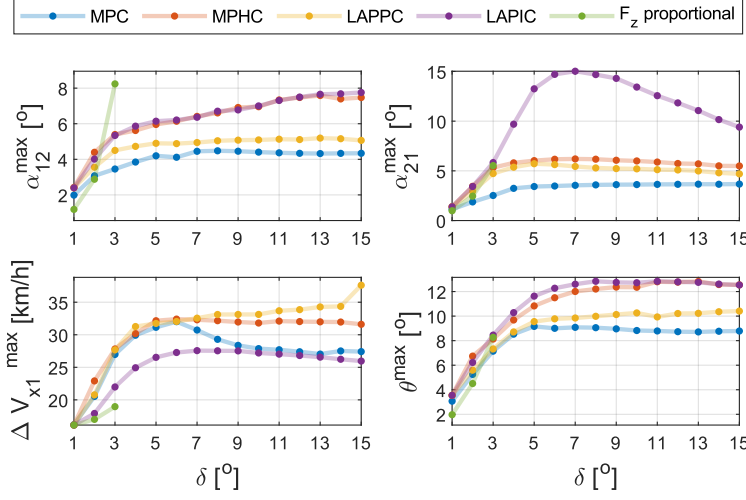


Figure 6.10: Sine-with-dwell test results involving braking with the tractor’s drive axle for predictive safety barrier controllers. Each point represents a performance metric for a single maneuver. The points are connected with lines for visual clarity. Adapted from Paper H.

The proposed controllers operate across three distinct intervention thresholds. Under normal driving conditions, $\mathbf{F}_x^{\text{ref}}$ is assigned directly as $\mathbf{F}_x^{\text{req}}$ without modification. During hazardous maneuvers, the controllers intervene by modulating the force allocation to preserve stability. Under critical threat of instability, the controllers supplement this reallocation with additional emergency braking to rapidly reduce vehicle speed.

6.4.1 Model Predictive Control

MPC is an ideal candidate for safe control allocation, as it integrates prediction and closed-loop optimization into a model-based design. An MPC framework for safe control allocation is presented in **Paper H**. An MPC formulation, MPC1 in Fig. 6.9, that uses steering-angle predictions with a 2.5 s prediction horizon and a single-track vehicle model ensures stable vehicle operation by enforcing soft tire-force limits with respect to a friction circle and hard side-slip-angle limits on the driven axles of the electric tractor and electric semitrailer. Soft limits can be violated with a large slack cost, and hard limits cannot be violated. MPC1 is formulated to track the reference axle forces, $\mathbf{F}_x^{\text{ref}}$, with minimal deviation possible. When MPC1 reaches the maximum number of iterations or becomes infeasible, the second MPC formulation, MPC2, is activated. MPC2 is formulated to minimize the side-slip angles and $\mathbf{F}_x^{\text{req}}$ by applying a maximum total force of

$\sum \mathbf{F}_x^{\text{ref}}$, which allows it to request a smaller longitudinal force (i.e., a larger brake force). Both MPC formulations use the same five vehicle states: longitudinal and lateral speed of the tractor, V_{x1} and V_{y1} , yaw rates of the tractor and semitrailer, ω_1 and ω_2 ; and articulation angle, θ .

Among the tested controllers, MPC framework ensured the smallest amount of tractor's drive axle side-slip angle, α_{12}^{max} , trailer's drive axle side-slip angle, α_{21}^{max} , and articulation angle, θ^{max} during the simulations, as shown in Fig. 6.10. The MPC framework performed the least emergency braking at larger steering angle magnitudes compared to other predictive controllers, resulting in a smaller tractor speed reduction, ΔV_{x1} , while the emergency braking amount was similar at lower steering angle magnitudes. Although LAPIC performed less emergency braking than MPC, it resulted in significantly higher side-slip angles.

The baseline case with F_z proportional braking could not maintain vehicle stability at steering angles greater than 3° due to the absence of additional emergency braking and predictive control, resulting in jackknifing.

6.4.2 Model Predictive Heuristic Control

MPHC presented in **Paper H** effectively decouples the predictive and optimal control phases of standard MPC. While it retains the same vehicle model for predicting future side-slip angles and lateral forces, it replaces the optimization step with a heuristic algorithm. This algorithm redistributes axle forces in response to high predicted side-slip angles and triggers additional emergency braking when these predictions reach critical thresholds. The modified longitudinal axle reference forces, $\mathbf{F}_x^{\text{ref,mod}}$, and the longitudinal force limits derived from the friction circle, $\mathbf{F}_x^{\text{lim}}$, are subsequently fed into a Quadratic Program (QP) for final control allocation. Due to the lack of closed-loop optimization over the prediction horizon (since the predictions assume constant longitudinal forces), the horizon is restricted to 1 s, which is shorter than that used in the MPC. A primary advantage of MPHC is its significantly lower computational burden than standard MPC.

MPHC successfully maintained vehicle stability during the sine-with-dwell tests, although the resulting side-slip and articulation angles were larger than those observed with the MPC, as shown in Fig. 6.10.

6.4.3 Lateral Acceleration Proportional Predictive Control

LAPPC, detailed in **Paper H**, uses lateral acceleration predictions with a 0.5 s prediction horizon. For large lateral acceleration predictions, it redistributes the reference longitudinal axle forces or applies an additional emergency braking.

Similar to MPHC, final allocation is performed within a QP by considering the modified longitudinal axle reference forces, $\mathbf{F}_x^{\text{ref,mod}}$, and the longitudinal force limits derived from the friction circle, $\mathbf{F}_x^{\text{lim}}$. The primary advantages of LAPPC are its reliance on fewer vehicle state measurements—requiring only the lateral accelerations of the units, which are readily available in modern heavy vehicle combinations—and a substantially lower computational load compared to both MPC and MPHC.

LAPPC successfully maintained vehicle stability during the sine-with-dwell maneuvers. As depicted in Fig. 6.10, the resulting side-slip and articulation angles were larger than those achieved with MPC, but smaller than those with MPHC. However, for larger steering amplitudes, LAPPC requested slightly more emergency braking than MPHC.

To determine whether prediction is strictly necessary for maintaining vehicle stability, the LAPIC strategy was implemented. LAPIC employs control logic similar to LAPPC, but relies solely on instantaneous lateral acceleration rather than predicted values. Although LAPIC requires the least additional emergency braking, it induces unacceptably large semitrailer side-slip angles, indicating trailer swing. Consequently, it cannot be considered a viable alternative to LAPPC.

Chapter 7

Conclusions and future outlook

This final chapter presents conclusions and recommendations for future research.

7.1 Conclusions

The studies presented in this thesis have investigated safe and energy-efficient control allocation for multi-unit heavy vehicles. Based on this work, conclusions are drawn for each research question as follows:

RQ1. *How does the allocation of large longitudinal forces across e-axes under low-friction conditions impact the yaw stability of multi-unit heavy vehicle combinations?*

The eigenvalue stability analysis of the tractor-semitrailer combination presented in **Paper A** demonstrated that excessive force allocation on the tractor or semitrailer axles under low friction and high lateral acceleration makes the combination unstable. Specifically, if the tractor drive axle is propelled or braked with large forces, jackknifing typically occurs, whereas if the semitrailer axle is propelled or braked with large forces, trailer swing typically occurs. Simulations evaluating different levels of friction utilization revealed that any force allocation that generates compressive force on the coupling—such as trailer propulsion or tractor braking—increases risk of instability compared to force allocations that generate tensile force on the coupling—such as trailer braking or tractor propulsion. Furthermore, if excessive forces are applied simultaneously to both the tractor and semitrailer, both units lose yaw stability, leading to a combination spin-out.

Real test results presented in **Paper B** confirmed these findings. Furthermore, it was shown that if only one axle within an axle group is excessively braked or propelled, the risk of yaw instability is substantially

reduced compared to scenarios where all axles in the group are subjected to significant forces. This indicates that e-trailers with a single driven axle tend to remain stable overall, except under conditions of high lateral accelerations approaching the friction limits.

RQ2. *How can yaw stability be represented through a Safe Operating Envelope (SOE), and what are the methodological differences between force-domain and slip-domain representations?*

A 3-dimensional SOE in the force domain that limits the braking and propulsion forces of the combination as a function of lateral acceleration is obtained with a single-track model in **Paper A**, with a two-track model in **Paper C**, and with a high-fidelity vehicle model validated with real vehicle tests in **Paper B**. The latter study further investigated the effects of different environmental parameters (such as friction and slope) and vehicle states (such as load distribution or speed) on the SOE. It showed how the SOEs expand or contract depending on the parameters. SOEs obtained using a two-track vehicle model and a tire model with combined slip were more accurate than those obtained with a single-track model.

As an alternative to the SOE in the force domain, **Paper D** introduced a slip polytope that limits the longitudinal slip of the driven wheels. This limit is formulated as a function of the side-slip angle deviation from a reference value, which is obtained via a single-track vehicle model estimator.

Building upon the 2-dimensional slip polytope framework that formulates longitudinal slip limits as a function of side-slip angle, **Paper E** presented a methodology to obtain safe longitudinal slip limits as a function of both side-slip angles and yaw rates using phase portrait analysis. To address the high dimensionality of multi-unit vehicle phase portraits, this study proposed a method to effectively decouple them across different units.

RQ3. *How can a hierarchical and distributed control architecture coordinate the over-actuation introduced by multiple electrified units in heavy vehicle combinations to achieve energy-efficient vehicle motion control?*

In **Paper F**, for a combination of an electric tractor and electric trailer, both centralized and distributed Power Loss Minimization (PLM) strategies are evaluated, concluding that distributed PLM at the combination and unit levels resulted in only marginally higher energy consumption than the centralized approach. The distributed architecture aggregates the Power Loss (PL) of multiple Electric Motors (EMs) into a single representation per unit. This eliminates the need for the combination-level

controller to access lower-level actuator details or request individual actuator forces, allowing it to allocate force requests strictly on a per-unit basis. This approach is particularly beneficial when trailing units from different manufacturers are equipped with multiple EMs, where low-level actuator interfaces—such as force requests or PL data—remain inaccessible to the upper-level controller.

Although PLM yielded approximately 1% energy reduction compared to rule-based force allocations to units, a more significant energy benefit is achieved by enabling selective disengagement of EMs, resulting in up to 5% energy reduction by mitigating idle losses.

Minimizing instantaneous PLs leads to disproportionate depletion rates across multiple units' batteries. This happens because the controller tends to use the vehicle unit with the most efficient EMs for propulsion and regenerative braking. To address this, an SOC control strategy was developed in the same study, resulting in a 1% energy penalty compared to using PLM alone. However, this penalty is comparable to the inherent energy benefit of using an e-trailer over a conventional trailer. Hence, it was shown that e-trailers can be utilized without a net energy penalty while achieving equal depletion rates across multiple batteries. Balancing these rates enables simultaneous charging while fully leveraging the significantly extended range, superior startability, gradeability, and traction benefits of e-trailers.

RQ4. *How can safe operating envelopes be integrated into an energy-efficient control allocation framework to prevent yaw instabilities induced by tire force saturation?*

Paper G utilized the SOE formulated in the force domain as inequality constraints in the Quadratic Program (QP) that minimizes PLs. The envelope contracts as lateral acceleration increases. This method was tested with real vehicles during brake-in-turn maneuvers on snow, resulting in stable vehicle operation, whereas braking without the SOE led to jackknifing.

Additionally, the same study presented an alternative method for estimating lateral tire forces using a single-track vehicle model and limiting longitudinal axle forces with respect to a friction circle. This approach was also demonstrated through real vehicle tests, resulting in safe braking. In this formulation, the longitudinal force limits are imposed as lower and upper bounds on the inputs.

The slip polytope presented in **Paper D** was evaluated with high-fidelity vehicle model simulations, followed by real-vehicle tests during propel-

in-turn maneuvers on ice. Adaptive slip limits were sent to the slip controller, which limits the longitudinal slip of the tractor's electric axle. The results showed that adaptive longitudinal slip limits improve yaw stability, traction, and acceleration performance compared to fixed longitudinal slip limits.

RQ5. *How can motion predictions—varying in prediction horizons, state measurement dependencies, and computational complexity—be integrated into control allocation to proactively mitigate yaw instabilities?*

In **Paper H**, three predictive safety barriers are evaluated: Model Predictive Control (MPC), Model Predictive Heuristic Control (MPHC), and Lateral Acceleration Proportional Predictive Control (LAPPC), featuring prediction horizons of 2.5 s, 1.0 s, and 0.5 s, respectively. While MPC and MPHC rely on single-track vehicle models to predict states, LAPPC uses direct predictions of lateral acceleration. Methodologically, MPC performs a full closed-loop optimization of actuator forces and predicted vehicle states, whereas MPHC and LAPPC use heuristic force-reallocation strategies based on their predictions. Despite these computational differences, all three controllers effectively decouple upper-level energy-efficient control allocation from lower-level, stability-preserving interventions. Under normal driving conditions, the safety barriers act transparently, strictly preserving the energy-efficient actuator force references. However, during critical maneuvers in which force reallocation alone cannot maintain yaw stability, all three architectures can trigger emergency braking.

All three controllers maintained vehicle stability during the simulated highway exit and sine-with-dwell maneuvers. While MPC showed the most robust performance with minimal emergency braking and the smallest side-slip angles, LAPPC stands out for its computational efficiency and minimal state-measurement requirements—relying solely on lateral acceleration, unlike MPC and MPHC, which require longitudinal and lateral tractor velocities, yaw rates, and the articulation angle.

RQ6. *How should the functional architecture and unit-to-unit interfaces be designed to facilitate safe and energy-efficient allocation?*

Paper G introduced an envisioned hierarchical control allocation architecture. Control allocation is divided into two layers: an upper level for combination control allocation and a lower level for unit control allocation. These layers may employ either PLM or a predetermined distribution strategy to allocate EM force requests. If PLM is adopted, units must

transmit lumped PL information to the combination Control Allocator (CA). Furthermore, as explained in **Paper F**, designing the e-trailers to disengage their EMs upon receiving a zero-force request (or another disengage signal) from the combination CA yields significant energy benefits by mitigating idle losses. Likewise, all electric units should communicate their SOC levels to the combination CA to equalize depletion rates among the batteries.

Addressing the safety aspects of control allocation involves multiple options. Safety constraints in the force domain, as explained in Section 6.2, can be implemented as inequality constraints or box constraints on the inputs directly within the QP formulated to minimize PLs. To utilize an SOE as a function of lateral acceleration, the units' lateral acceleration measurements must be communicated to the combination CA. To limit actuator longitudinal forces using lateral force estimates, a more extensive set of state measurements may be required, depending on the fidelity of the lateral force estimation model, including longitudinal and lateral tractor accelerations and the articulation angle.

If adaptive slip limits defined as a function of the side-slip angle are adopted, the slip controller imposes these limits by constraining the force requests allocated by the combination CA. This approach effectively decouples the controller logic: energy-efficient control allocation takes place in the force domain, while adaptive slip limits safeguard stability. In this method, accurate estimation of side-slip angles by a motion estimator becomes crucial.

Alternatively, if predictive safety barriers are employed, the combination CA allocates the reference unit forces, while the safety barriers conditionally modify these requests to maintain stability at a lower level. Similar to the adaptive slip limit strategy, this architecture maintains a strict separation between energy optimization and safety logic. However, implementing MPC and MPHC requires an extensive set of state measurements—such as the longitudinal and lateral velocities of the tractor, yaw rates, articulation angle, and steering angle—because these methods rely on a single-track vehicle model. In contrast, if LAPPC is employed, measuring only the units' lateral accelerations is sufficient, significantly simplifying the required sensor setup.

The friction coefficient is assumed to be available via a motion estimator for all methods. In addition to detailed sensor measurements, methods that rely on a vehicle model require vehicle parameters such as wheelbase, mass, and inertia. Section 3.2 details the envisioned functional architecture alongside a comprehensive list of the necessary signal interfaces.

Although the SOEs detailed in Section 5.1 provide valuable insights into the stability limits of vehicle combinations, using predetermined SOEs in real-time control requires an extensive set of offline-generated databases. Because numerous factors affect these envelopes—such as load distribution, friction, and slope—relying on SOEs for real-time control can become computationally expensive, consume significant memory, and ultimately become impractical. Alternatively, rather than using a vehicle model to predetermine the SOE, the same model can be integrated into an MPC framework to maintain stability in real time, eliminating the need for an offline-generated database. However, as previously noted, MPC requires an extensive set of state measurements and vehicle parameters. Furthermore, while it delivers robust safety performance, this approach inherently shifts the computational burden of offline SOE generation to real-time execution on the ECU. While these computational and signaling requirements are technically achievable, the significantly simpler LAPPc approach—relying solely on lateral accelerations—delivers strong performance and effectively prevents instabilities. Given its minimal computational footprint, LAPPc emerges as a prime candidate for immediate industrialization. It successfully achieves the industry’s required balance between superior performance and near-term practical realizability. Consequently, LAPPc can be deployed as a pragmatic, immediate solution, paving the way for longer-term MPC adoption once the necessary sensor interfaces and computational capacities are fully secured.

As an alternative or supplement to safety barriers in the force domain, adaptive longitudinal slip limits can ensure stable vehicle operation, as this method has shown strong performance in both simulations and real-world vehicle tests. This approach is relatively straightforward to implement, provided that the EMs feature a slip or speed limit interface and that side-slip angle estimation is available. Consequently, it emerges as a strong candidate for immediate industrialization.

While the theoretical benefits of PLM are clear, standardizing the transmission of PL maps from e-trailers remains challenging because of multiple manufacturers, the need for compatibility across tractor and trailer manufacturers, and broader regulatory and standardization challenges related to signal interfaces. Nevertheless, as long as the tractor and trailer belong to the same manufacturer, PLM remains a strong candidate. In such cases, PLM can be implemented online in real-time or used offline to generate a predetermined power distribution map. However, regardless of the tractor-trailer manufacturer combination, it is crucial for e-trailers to transmit their SOC status signals to the tractor. Furthermore, enabling disengagement capabilities for the trailers’ electrified powertrains is essential for reducing energy consumption by eliminating idle losses.

7.2 Future outlook

While this study primarily addresses divergent yaw instabilities, such as jackknifing and trailer swing, future research should expand this scope to include dynamic yaw instabilities, particularly trailer sway, as well as roll stability.

Additionally, this thesis assumes ideal state measurements and estimates. The impact of inaccuracy, uncertainty, and low resolution in measured states (e.g., vehicle speed and acceleration) and environmental parameters (e.g., road friction) on controller performance remains a critical area for subsequent investigation.

Finally, current and pending regulatory constraints that prohibit e-trailers from pushing the tractor—such as the UN Regulation 13 amendments [12]—fall outside the scope of this work. Incorporating these legislative boundaries into the control architecture represents an important direction for practical implementation.

Bibliography

- [1] I. Katsarova. “Weights and dimensions of road vehicles in the EU.” European Parliamentary Research Service. 2014. Accessed: Jul. 16, 2026. [Online]. Available: europarl.europa.eu/RegData/bibliotheque/briefing/2014/140796/LDM_BRI%282014%29140796_REV1_EN.pdf
- [2] S. Kharrazi et al. (Nov. 2023). A review of more than a decade of research and field tests on longer vehicle combinations in Sweden that supported road approval. Presented at 17th Int. Symp. on Heavy Vehicle Transp. & Technol., Brisbane, Australia. [Online]. Available: hvttforum.org/wp-content/uploads/2024/02/HVTT17_paper_7891_Kharrazi.pdf
- [3] European Parliament News. “Parliament to decide on longer and heavier green trucks.” europarl.europa.eu. Accessed: Jul. 16, 2026. [Online]. Available: europarl.europa.eu/news/en/agenda/briefing/2024-03-11/18/parliament-to-decide-on-longer-and-heavier-green-trucks
- [4] ZF. “Electrified Trailer Solution by ZF.” zf.com. Accessed: Jul. 16, 2026. [Online]. Available: zf.com/products/en/cv/stories_content_pages/electrified_trailer_solution.html
- [5] Trailer Dynamics. “The eTrailer defines a new way to power the entire tractor-trailer system efficiently and sustainably.” trailerdynamics.de. Accessed: Jul. 16, 2026. [Online]. Available: trailerdynamics.de
- [6] Schmitz Cargobull. “S.KOe COOL.” cargobull.com. Accessed: Jul. 16, 2026. [Online]. Available: cargobull.com/en/products/box-body/box-semi-trailer/skoe-cool
- [7] *Road vehicles — Interchange of digital information on electrical connections between towing and towed vehicles*, ISO Standard 11992-2:2023, International Organization for Standardization, 2023. [Online]. Available: iso.org/standard/79621.html

- [8] N. Bradford. “How Octopus Arms Bypass the Brain.” [scientificamerican.com](https://scientificamerican.com/article/how-octopus-arms-bypass-the-brain/). Accessed: Jul. 16, 2026. [Online]. Available: scientificamerican.com/article/how-octopus-arms-bypass-the-brain/
- [9] G. Levy, T. Flash, and B. Hochner, “Arm coordination in octopus crawling involves unique motor control strategies,” *Current Biol.*, vol. 25, no. 9, pp. 1195–1200, May 2015, doi: 10.1016/j.cub.2015.02.064.
- [10] G. Barathieu. “UPY 2017 Award Winners - Underwater Photographer of the Year.” underwaterphotographeroftheyear.com. Accessed: Jul. 16, 2026. [Online]. Available: underwaterphotographeroftheyear.com/archive/2017-winning-images/award-winners/1/
- [11] Volvo Trucks. *Volvo Trucks - Volvo Trucks vs 750 Tonnes: An extreme heavy haulage challenge*. (Apr. 12, 2016). Accessed: Jul. 16, 2026. [Online Video]. Available: youtube.com/watch?v=5UUz1zTZfU&ab_channel=VolvoTrucks
- [12] United Nations Economic Commission for Europe (UNECE), “UN Reg 13 Amendment for E-Axles and Self-Propelled Trailers,” UNECE Working Doc. GRVA-21-41e, 2025. Accessed: Jul. 16, 2026. [Online]. Available: unece.org/sites/default/files/2025-01/GRVA-21-41e.pdf
- [13] Volvo Trucks. “Volvo FH Electric.” volvotrucks.com. Accessed: Jul. 16, 2026. [Online]. Available: volvotrucks.com/en-en/trucks/electric/volvo-fh-electric.html
- [14] L. Laine, “Reconfigurable motion control systems for over-actuated road vehicles,” Ph.D. dissertation, Dept. Appl. Mechanics, Chalmers Univ. of Technol., Gothenburg, Sweden, 2007. [Online]. Available: research.chalmers.se/en/publication/41492
- [15] M. Gäfvert, “Topics in modeling, control, and implementation in automotive systems,” Ph.D. dissertation, Dept. Autom. Control, Lund Inst. of Technol., Lund, Sweden, 2003. [Online]. Available: portal.research.lu.se/files/4840790/8571803.pdf
- [16] J. Torinsson, M. Jonasson, D. Yang, B. Jacobson, and T. Ghandriz, “Energy consequences of separating velocity planning and torque distribution in over-actuated electric vehicles,” *Transp. Eng.*, vol. 23, 2026, Art. no. 100419, doi: 10.1016/j.treng.2025.100419.
- [17] T. Ghandriz, “Transportation mission-based optimization of heavy combination road vehicles and distributed propulsion, including predictive energy and motion control,” Ph.D. dissertation, Dept. Mechanics and Maritime Sci.,

- Chalmers Univ. of Technol., Gothenburg, Sweden, 2020. [Online]. Available: research.chalmers.se/en/publication/?id=520358
- [18] J. Torinsson, “Power loss minimization in electric cars by wheel force allocation,” Lic. thesis, Dept. Mechanics and Maritime Sci., Chalmers Univ. of Technol., Gothenburg, Sweden, 2020. [Online]. Available: research.chalmers.se/en/publication/520503
- [19] O. Härkegård, “Backstepping and control allocation with applications to flight control,” Ph.D. dissertation, Dept. Elect. Eng., Linköping Univ., Linköping, Sweden, 2003. [Online]. Available: diva-portal.org/smash/get/diva2:243491/FULLTEXT01.pdf
- [20] J. Torinsson, M. Jonasson, D. Yang, and B. Jacobson, “Energy reduction by power loss minimisation through wheel torque allocation in electric vehicles: a simulation-based approach,” *Vehicle Syst. Dyn.*, vol. 60, no. 5, pp. 1488–1511, 2022, doi: 10.1080/00423114.2020.1858121.
- [21] S. Janardhanan, L. Laine, M. Jonasson, B. Jacobson, and E. Gelso, “Motion control and power coordination of electric propulsion and braking distributed on multiple axles on heavy vehicles,” in *Proc. 2022 IEEE Vehicle Power and Propulsion Conf. (VPPC)*, Merced, CA, USA, pp. 1–8, doi: 10.1109/VPPC55846.2022.10003450.
- [22] *Road vehicles — Functional safety*, ISO Standard 26262-1:2018, International Organization for Standardization, 2018. [Online]. Available: [iso.org/standard/68383.html](https://www.iso.org/standard/68383.html)
- [23] J. Aurell and C. B. Winkler. (1995). Standard test procedures for the lateral stability of heavy vehicle combinations. Presented at 4th Int. Symp. on Heavy Vehicle Weights and Dimensions, Ann Arbor, MI, USA. [Online]. Available: hvtforum.org/wp-content/uploads/2019/11/Standard-Test-Procedures-For-The-Lateral-Stability-Of-Heavy-Vehicle-Combinations-Aurell-.pdf
- [24] Transportstyrelsen, “TSFS 2018:40 Transportstyrelsens föreskrifter och allmänna råd om tekniska krav på fordonståg med bruttovikt över 64 ton och längd om högst 25,25 meter,” Transportstyrelsen, 2018. Accessed: Jul. 16, 2026. [Online]. Available: transportstyrelsen.se/TSFS/TSFS%202018_40k.pdf
- [25] S. Kharrazi, B. Augusto, and N. Fröjd, “Assessing dynamics of heavy vehicles in a driving simulator,” *Transp. Res. Part F: Traffic Psychol. and Behav.*, vol. 65, pp. 306–315, 2019, doi: 10.1016/j.trf.2019.07.007.

- [26] J. Aurell and T. Wadman, "Vehicle combinations based on the modular concept," Nordiska Vägtekniska Förbundet, Report no. 1/2007, 2007. Accessed: Jul. 16, 2026. [Online]. Available: nvfnorden.org/wp-content/uploads/2020/10/2007-Vehicle-combinations-based-on-the-modular-concept-Background-and-analysis.pdf
- [27] Toyota. "Hilux How To: Trailer Sway Control." toyota.co.uk. Accessed: Jul. 16, 2026. [Online]. Available: mag.toyota.co.uk/hilux-trailer-sway-control/
- [28] Bosch. "Trailer safety control." bosch-mobility.com. Accessed: Jul. 16, 2026. [Online]. Available: bosch-mobility.com/en/solutions/driving-safety/trailer-safety-control/
- [29] AL-KO. "ATC: AL-KO Trailer Control." alko-tech.com. Accessed: Jul. 16, 2026. [Online]. Available: alko-tech.com/en/Vehicle-types-products/atc-al-ko-trailer-control_pdt_227
- [30] Q. Wang and Y. He, "A study on single lane-change manoeuvres for determining rearward amplification of multi-trailer articulated heavy vehicles with active trailer steering systems," *Vehicle Syst. Dyn.*, vol. 54, no. 1, pp. 102–123, 2016, doi: 10.1080/00423114.2015.1123280.
- [31] Volvo Trucks. "Full-scale crash tests of Volvo's electric trucks." volvotrucks.com. Accessed: Jul. 16, 2026. [Online]. Available: volvotrucks.com/en-en/news-stories/stories/2022/dec/full-scale-crash-tests-of-volvo-s-electric-trucks.html
- [32] Volkswagen. "Active roll compensation." volkswagen-newsroom.com. Accessed: Jul. 16, 2026. [Online]. Available: volkswagen-newsroom.com/en/the-new-touareg-offroad-media-drive-in-morocco-4291/active-roll-compensation-4301
- [33] B. Chen and H. Peng, "Differential-braking-based rollover prevention for sport utility vehicles with human-in-the-loop evaluations," *Vehicle Syst. Dyn.*, vol. 36, no. 4–5, pp. 359–389, 2001, doi: 10.1076/vesd.36.4.359.3546.
- [34] V. F. Dal Poggetto and A. L. Serpa, "Vehicle rollover avoidance by application of gain-scheduled LQR controllers using state observers," *Vehicle Syst. Dyn.*, vol. 54, no. 2, pp. 191–209, 2016, doi: 10.1080/00423114.2015.1125005.

- [35] B. Jacobson, “Combined longitudinal and lateral slip,” in *Vehicle Dyn. Compendium*. Gothenburg, Sweden: Chalmers University of Technology, 2020, ch. 2, sec. 2.5, pp. 133–134. [Online]. Available: research.chalmers.se/publication/520229/file/520229_Fulltext.pdf
- [36] N. Fröjd. “Handling analysis and control development of commercial trucks with Volvo transport models.” [mathworks.com](https://mathworks.com/videos/handling-analysis-and-control-development-of-commercial-trucks-with-volvo-transport-models-1622035211192.html). Accessed: Jul. 16, 2026. [Online]. Available: se.mathworks.com/videos/handling-analysis-and-control-development-of-commercial-trucks-with-volvo-transport-models-1622035211192.html
- [37] B. Jacobson, “Anisotropic brush model with independent bristles,” in *Vehicle Dyn. Compendium*. Gothenburg, Sweden: Chalmers University of Technology, 2020, ch. 2, sec. 2.5.1, pp. 134–137. [Online]. Available: research.chalmers.se/publication/520229/file/520229_Fulltext.pdf
- [38] R. Fu, H. Zhang, Y. Guo, F. Yang, and Y. Lu, “Real-time estimation and prediction of lateral stability of coaches: a hybrid approach based on EKF, BPNN, and online autoregressive integrated moving average algorithm,” *IET Intell. Transp. Syst.*, vol. 14, no. 13, pp. 1892–1902, 2020, doi: 10.1049/iet-its.2020.0385.
- [39] J. Funke, M. Brown, S. M. Erlien, and J. C. Gerdes, “Collision avoidance and stabilization for autonomous vehicles in emergency scenarios,” *IEEE Trans. Control Syst. Technol.*, vol. 25, no. 4, pp. 1204–1216, Jul. 2017, doi: 10.1109/TCST.2016.2599783.
- [40] S. M. Erlien, S. Fujita, and J. C. Gerdes, “Shared steering control using safe envelopes for obstacle avoidance and vehicle stability,” *IEEE Trans. Intell. Transp. Syst.*, vol. 17, no. 2, pp. 441–451, Feb. 2016, doi: 10.1109/TITS.2015.2453404.
- [41] A. Bharganwar, M. V. Ramasamy, U. Erdinc, and M. Jonasson, “Adaptive Slip Control for Electrified Articulated Heavy Vehicles,” presented at the 17th Int. Symp. on Adv. Vehicle Control, Tsukuba, Japan, Sep. 2026.
- [42] Colmis AB. “Test Tracks.” colmis.com. Accessed: Jul. 16, 2026. [Online]. Available: colmis.com/test-tracks/
- [43] R. C. Coulter, “Implementation of the pure pursuit path tracking algorithm,” Robot. Inst., Carnegie Mellon Univ., Pittsburgh, PA, USA, Tech. Rep. CMU-RI-TR-92-01, Jan. 1992.

- [44] *Heavy Commercial Vehicles and Buses — Test Method for Yaw Stability — Sine with Dwell Test*, ISO Standard 18375:2016, International Organization for Standardization, 2016. [Online]. Available: [iso.org/standard/62291.html](https://www.iso.org/standard/62291.html)

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