

THESIS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY
IN
MACHINE AND VEHICLE SYSTEMS

MINIMISING OPERATIONAL COST WHILE
ENSURING SAFE MOTION FOR HEAVY
VEHICLES WITH MODULAR ELECTRIC AXLES

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Minimising operational cost while ensuring safe motion for heavy vehicles with modular electric axles

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Cover: Electric heavy commercial vehicle concept with powertrain system distributed on the front and rear axle. The image was generated using AI.

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*To my family,
to my teachers, whose guidance shaped my journey,
& to the departed souls, whose memories remain with me.*

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Abstract

The electrification of heavy commercial vehicles offers opportunities beyond replacing conventional powertrains with batteries and electric machines. In particular, electric axles enable propulsion, braking, and motion control to be distributed and coordinated across the vehicle. This thesis investigates how modular electric-axle powertrains can be designed and controlled to reduce operational costs while maintaining safe, robust, and predictable vehicle behaviour.

A modular powertrain architecture is proposed in which different axles or axle groups can be assigned distinct functions. A dedicated cruise axle is tailored for efficient steady-state highway operation, while a startability axle provides high traction at low speeds and on demanding grades. The joint selection of motor ratings and gear ratios, together with coordinated axle operation, is investigated to identify powertrain configurations that satisfy vehicle motion requirements while improving energy efficiency and driving range.

Building on this modular electric axle architecture, the thesis then addresses the coordination of multiple electric powertrains and friction brakes. Control-allocation methods are developed to distribute wheel-torque demands among redundant actuators while achieving the requested vehicle motion and minimising operational losses. In contrast to conventional heuristic weighting, the proposed formulations represent actuator losses as physically based cost rates, providing a direct connection between control decisions and vehicle operating cost. Tyre-friction constraints are incorporated to account for the coupled longitudinal and lateral demands during vehicle manoeuvres.

For demanding driving conditions, vehicle-level motion control is combined with lower-level actuator coordination. A model predictive control-based force generator regulates longitudinal and yaw responses within prescribed operating limits, while subsequent control allocation distributes the required forces among the available actuators. The developed methods are evaluated under conditions including low-friction surfaces and combined braking and cornering, with emphasis on stability, path tracking, and predictable vehicle response.

Finally, the optimisation framework is extended beyond energy consumption

by incorporating tyre wear alongside actuator losses. The results demonstrate that modular electric axles, combined with optimisation-based design and control, can provide new means of improving energy efficiency, reducing operating costs, and maintaining safe and predictable motion in heavy electric vehicles.

Keywords: Heavy vehicles, electric axles, electric machines, friction brakes, power loss, tyre wear, control allocation, cost minimisation.

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List of publications

This thesis comprises an extended summary and is based on the following appended papers:

Paper A

S. Janardhanan, L. Laine, M. Jonasson, B. Jacobson and M. Alaküla, "*Concept design of electric cruise and startability axles for long haul heavy vehicles to maximise driving range*," 2021 IEEE Vehicle Power and Propulsion Conference (VPPC), Gijon, Spain, 2021, pp. 1-8, doi: 10.1109/VPPC53923.2021.9699364.

Paper B

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*," 2022 IEEE Vehicle Power and Propulsion Conference (VPPC), Merced, CA, USA, 2022, pp. 1-8, doi: 10.1109/VPPC55846.2022.10003450.

Paper C

S. Janardhanan, E. Gelso, L. Laine, M. Jonasson and B. Jacobson, "*Reviewing control allocation using quadratic programming for motion control and power coordination of battery electric vehicles*," 2022 IEEE Vehicle Power and Propulsion Conference (VPPC), Merced, CA, USA, 2022, pp. 1-8, doi: 10.1109/VPPC55846.2022.10003297.

Paper D

Sachin Janardhanan, Leon Henderson, Mats Jonasson, Bengt Jacobson, Esteban R. Gelso, "*Simulation-based assessment of wheel torque allocation strategies on heavy vehicles with drivetrains on multiple axles*," Transportation Engineering, Volume 20, 2025, 100322, ISSN 2666-691X, doi: 10.1016/j.treng.2025.100322.

Paper E

Sachin Janardhanan, Jonas Persson, Mats Jonasson, Bengt Jacobson, Esteban R Gelso, Leon Henderson, "*Energy-Efficient Wheel Torque Distribution for Heavy Electric Vehicles With Adaptive Model Predictive Control and Control Allocation*", in IEEE Open Journal of Vehicular Technology, vol. 6, pp. 2909-2924, 2025, doi: 10.1109/OJVT.2025.3619823.

Paper F

T.Ghandriz, S. Janardhanan, "*Vehicle Motion Control Allocation Including Tire Wear Minimization*," in IEEE Open Journal of Vehicular Technology, vol. 7, pp. 901-916, 2026, doi: 10.1109/OJVT.2026.3666361.

Paper G

Sachin Janardhanan, Toheed Ghandriz, Mats Jonasson, Bengt Jacobson "*Integrated tyre wear and actuator power loss minimisation in heavy electric vehicle control*," submitted to 17th International Symposium on Advanced Vehicle Control (AVEC 2026).

The author of this thesis was the first author of Papers A–E and G and the primary contributor to the research presented therein, including formulation of the research objectives, development of the proposed methods, implementation of simulation models, simulation studies, vehicle testing and data acquisition, analysis of results, and manuscript preparation. The work was conducted in close collaboration with the co-authors, who contributed through scientific supervision, discussions, research idea development, and critical review.

For Paper A, L. Laine contributed significantly to the modular axle concept, simulation studies, Python codebase, and initial manuscript. For Paper F, the author contributed to the literature review, evaluation and analysis of the proposed method, and manuscript preparation. The study was primarily conceived and led by the first author.

The list of patents generated during this research and involving the author:

- 1) L. Laine, E. Gelso, S. Janardhanan, J. Vilca, K. Tagesson, and M. Jonasson, "Method and system for controlling an electrically powered heavy commercial vehicle," WO 2023/193893 A1, 2023.
- 2) J. Vilca, E. Gelso, L. Laine, M. Askerdal, M. Jonasson, and S. Janardhanan, "A control system for controlling a retardation of a vehicle," WO 2023/174542 A1, 2023.
- 3) E. Gelso, L. Laine, J. Vilca, S. Janardhanan, and K. Tagesson, "Method and device for controlling a technical system in real time," WO 2023/156017 A1, 2023.
- 4) J. Persson, J. Åkesson, B. Jacobson, S. Janardhanan, and L. Laine, "Vehicle control allocation with relaxed global force-request," WO 2025/016526 A1, 2025.
- 5) L. Laine, J. Lindberg, S. Janardhanan, and M. Alakula, "Vehicle wheel with electrical generator for heating the tire and computer system," WO 2025/190501 A1, 2025.
- 6) L. Laine, E. Gelso, S. Janardhanan, and M. Jonasson, "A method for controlling a propulsion system," WO 2022/117706 A1, 2022.
- 7) E. Gelso, S. Janardhanan, and L. Laine, "A propulsion system for a heavy-duty vehicle, a heavy-duty vehicle and a method for controlling a propulsion system," EP 4577422 A1, 2025

Nomenclature

The nomenclature used here is applicable only for the thesis and not for the appended papers.

Acronyms

HCV	Heavy commercial vehicle
CEP	Centralised powertrain
DEP	Distributed powertrain
BEV	Battery electric heavy vehicle
TCO	Total cost of ownership
TCOP	Total cost of operations
EM	Electric machine
PMSM	Permanent magnet synchronous machine
IM	Induction machine
CAN	Controller area network
DC	Direct current
QP	Quadratic Program
OP	Operating point
CoG	Centre of gravity
MPC	Model predictive control
EBS	Electronic braking system
GVWR	Gross vehicle weight rating

Subscripts

crs	Cruise axle
stb	Startability axle
Brk	Friction brake
EM	Electric machine
w	Wheel
dr	Driven axle
tg	Tag axle

Symbols

Unless otherwise stated: All physical quantities are in SI units and radians, monetary quantities are in Euro=€.

i	Axle index
j	Actuator index
l	Wheel index
$F_{x,i}$	Equivalent longitudinal tyre force at axle i in the single-track vehicle model
$F_{y,i}$	Equivalent lateral tyre force at axle i in the single-track vehicle model
$F_{z,i}$	Equivalent normal force at axle i in the single-track vehicle model
$F_{x,f}$	Equivalent longitudinal force acting at the front axle group in the single-track model
$F_{y,f}$	Equivalent lateral force acting at the front axle group in the single-track model
$F_{x,r}$	Equivalent longitudinal force acting at the rear axle group in the single-track model
$F_{y,r}$	Equivalent lateral force acting at the rear axle group in the single-track model
δ_i	Steering angle associated with axle i in the reduced-order single-track representation
l_i	Longitudinal position of axle i relative to the vehicle CoG
l_f	Equivalent distance from the CoG to the front axle group
l_r	Equivalent distance from the CoG to the rear axle group

M_z	Yaw moment about the vehicle CoG in the reduced-order single-track model.
$M_{z,f}$	Front axle-group contribution to the yaw moment in the reduced-order single-track model
$M_{z,r}$	Rear axle-group contribution to the yaw moment in the reduced-order single-track model
n_f	Number of axles in the front axle group
n_r	Number of axles in the rear axle group
α_i	Slip angle of an axle i in the single-track model
C_i	Tyre cornering stiffness of axle i
F_x	Longitudinal force acting at the tyre–road interface in the single-wheel model
T_w	Applied wheel torque in the single-wheel model
T_{rr}	Rolling resistance torque in the single-wheel model
F_{rr}	Model-equivalent rolling resistance force
F_z	Normal load acting on the equivalent wheel in the single-wheel model
r_w	Tyre radius
c_{rr}	Rolling resistance coefficient
c_d	Aerodynamic drag coefficient
M_v	Equivalent vehicle mass
g	Gravitational acceleration
ρ	Air density
A_v	Vehicle frontal area
$\varphi_{y,r}$	Road pitch angle
μ	Tyre–road friction coefficient
a_x	Longitudinal acceleration of the vehicle in the body-fixed reference frame
a_y	Lateral acceleration of the vehicle in the body-fixed reference frame
v_x	Longitudinal vehicle velocity at the in the body-fixed reference frame
v_y	Lateral vehicle velocity at the centre of gravity in the body-fixed reference frame
ω_z	Vehicle yaw rate about the vertical axis
β	Vehicle side-slip angle at the centre of gravity
δ_f	Front axle steering angle
I_z	Vehicle yaw moment of inertia about the z-axis through the CoG
$M_{z,\text{ext}}$	External yaw moment

$T_{EM,ij}$	Delivered electromagnetic torque of electric machine j on axle i
$T_{EM,ij}^{\text{req}}$	Requested electromagnetic torque of electric machine j on axle i
$T_{EM,ij}^{\text{min}}$	Minimum (lower limit) physical torque capability of electric machine j on axle i
$T_{EM,ij}^{\text{max}}$	Maximum (upper limit) physical torque capability of electric machine j on axle i
$T_{EM,ij}^{\text{dyn}}$	Dynamically achievable electromagnetic torque of electric machine j on axle i
$\tau_{EM,ij}$	Time constant of electric machine actuator dynamics j on axle i
$\omega_{EM,ij}$	Angular speed of electric machine j on axle i
$\eta_{trmn,i}$	Transmission efficiency of driven axle i
gr_i	Transmission gear ratio of driven axle i
P_{Batt}^{cons}	Battery power consumption during motoring operation
P_{Batt}^{gen}	Battery regenerative power during regenerative (braking) operation
$P_{EM,ij}$	Mechanical power of the electric machine j on axle i
$P_{EM,ij}^{\text{loss}}$	Electric machine losses associated with electric machine j on axle i
η_{el}^{mot}	Lumped motoring efficiency including battery and inverter conversion losses
η_{el}^{gen}	Lumped regenerative efficiency including battery and inverter conversion losses
$P_{EM,ij}^{\text{loss}}$	Power loss associated with electric machine j on axle i
$P_{Brk,ij}^{\text{loss}}$	Power dissipated by friction brake actuator j on axle i
$T_{Brk,ij}$	Delivered torque by friction brake actuator j on axle i
$T_{Brk,ij}^{\text{req}}$	Requested torque of friction brake actuator j on axle i
$T_{Brk,ij}^{\text{min}}$	Minimum (lower limit) physical torque capability of friction brake actuator j on axle i
$T_{Brk,ij}^{\text{max}}$	Maximum (upper limit) physical torque capability of friction brake actuator j on axle i
$T_{Brk,ij}^{\text{dyn}}$	Dynamically achievable brake torque at axle i by friction brake j
$\tau_{Brk,ij}$	Time constant of brake actuator j on axle i
$\omega_{w,il}$	Wheel angular speed of wheel l on axle i
κ_{il}	Longitudinal wheel slip ratio of wheel l on axle i
κ_i^{lim}	Maximum allowable wheel slip ratio for ABS activation
F_x^{req}	Requested global longitudinal force to be distributed among the actuators

$F_{x,crs}$	Longitudinal force produced by the cruise axle
$F_{x,stb}$	Longitudinal force produced by the startability axle
$F_{x,crs}^{\text{req}}$	Requested longitudinal force contribution from the cruise axle transmitted through the wheel–road contacts
$F_{x,stb}^{\text{req}}$	Requested longitudinal force contribution from the startability axle transmitted through the wheel–road contacts
$F_{x,Brk}^{\text{req}}$	Requested equivalent longitudinal force contribution from the friction brake system transmitted through the wheel–road contacts
$F_{x,crs}^{\text{min}}$	Minimum (lower limit) longitudinal force capability of the cruise axle
$F_{x,crs}^{\text{max}}$	Maximum (upper limit) longitudinal force capability of the cruise axle
$F_{x,stb}^{\text{min}}$	Minimum (lower limit) longitudinal force capability of the startability axle
$F_{x,stb}^{\text{max}}$	Maximum (upper limit) longitudinal force capability of the startability axle
$F_{x,Brk}^{\text{min}}$	Minimum (lower limit) longitudinal braking force capability of the friction brake system
$F_{x,Brk}^{\text{max}}$	Maximum (upper limit) longitudinal braking force capability of the friction brake system
$\omega_{w,crs}$	Wheel angular speed of the cruise axle
$\omega_{w,stb}$	Wheel angular speed of the startability axle
SOC_{end}	Battery state of charge at the end of the drive cycle
$T_{EM,crs}^{\text{req}}$	Requested electric machine torque from the cruise axle
$T_{EM,stb}^{\text{req}}$	Requested electric machine torque from the startability axle
T_{Brk}^{req}	Requested equivalent lumped friction brake torque
$F_{x,il}$	Longitudinal tyre force at wheel l on axle i
$F_{y,il}$	Lateral tyre force at wheel l on axle i
F_y^{req}	Requested global lateral force from the vehicle motion controller
M_z^{req}	Requested yaw moment from the vehicle motion controller
$F_{x,i}^{\text{lim}}$	Available longitudinal tyre force capacity of axle i
$F_{y,i}^{\text{pred}}$	Predicted lateral force demand of axle i
a_x^{ref}	Reference longitudinal acceleration request
v_x^{ref}	Reference longitudinal velocity
ω_z^{ref}	Reference yaw rate at the centre of gravity
β^{ref}	Reference vehicle sideslip angle at the centre of gravity
δ_{SWA}	Driver steering wheel angle request
K_u	Vehicle understeer gradient

k_{st}	Steering ratio
n_{conf}	Confidence factor for tyre force constraints
ω_z^{max}	Maximum allowable yaw rate
ω_z^{min}	Minimum allowable yaw rate
β^{max}	Maximum allowable vehicle side-slip angle calculated at the centre of gravity
β^{min}	Minimum allowable vehicle side-slip angle calculated at the centre of gravity
α_f^{max}	Maximum allowable front axle tyre slip angle in a single-track model
α_r^{max}	Maximum allowable rear axle tyre slip angle in a single-track model
$T_{w,il}^{\text{lim}}$	Torque limit of a wheel on axle i
$t_{w,i}$	Track width of an axle i
$C_{\text{pl},ij}$	Power-loss cost rate associated with actuator j on axle i (€/s)
$C_{tr,il}^{\text{wear}}$	Tyre wear cost rate for tyre side l on axle i (€/s)
C_{el}	Electricity price €/kWh
M_{tr}^{wear}	Accumulated tyre tread mass loss (kg)
$M_{tr,x}^{\text{wear}}$	Accumulated tyre mass loss caused by longitudinal slip only
K_{tr}	Experimentally calibrated tyre wear coefficient
w_{tr}	Tyre tread width
d_{tr}	Tyre outer diameter
D_{tr}	Tyre rubber density
$h_{tr,il}$	Usable tread depth of tyre on axle i , side l
p_{tr}	Tyre price (€)
$\Delta h_{tr,x}$	Reduction in tyre tread depth caused by longitudinal wear
$n_{w,l}$	Number of tyres on axle side l .

Definition

Operational cost	Cost which accumulates over travelled distance or transport mission time. In general, this may include energy consumption, tyre wear, battery degradation, driver cost, maintenance, tolls, and depreciation costs. In this thesis, the operational cost is limited to components that are directly related to vehicle motion and can be expressed as instantaneous cost rate terms, namely electrical power and tyre wear rate arising from motion. Costs associated with auxiliary or task-specific heavy-vehicle operations are excluded.
Vehicle configuration	Total number of wheels and the number of driven wheels on a vehicle unit and the type of vehicle (tractor / rigid). A pair of driven wheels are usually on the same axle. For example, 4x2, 4x4, 6x4 etc. In a vehicle combination, the vehicle unit defines only the configuration of the leading vehicle unit.
Powertrain configuration	Specification of the powertrain components, like gear ratio, EM power etc.
Heavy commercial vehicle	A commercial road vehicle with a GVWR above 3,500 kg. This includes medium- and heavy-duty trucks as well as vocational vehicles designed either for goods transport or for task-specific operations.

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Chapter 1

Introduction

Global freight transport is expected to grow substantially in the coming decades, driven by expanding trade, urbanization, and the rapid rise of e-commerce [42, 44]. Heavy commercial vehicles (HCVs) are projected to remain the on-road mode of long-haul freight transport due to their high load capacity and operational flexibility, leading to a significant increase in vehicle-kilometers traveled worldwide [43]. Although HCVs constitute only a small fraction of the total vehicle fleet, they contribute disproportionately to greenhouse-gas emissions and air pollutants such as nitrogen oxides NO_x and particulate matter, posing significant environmental and public-health challenges [19, 26, 37]. Without effective mitigation measures, the continued growth in freight demand could jeopardize the achievement of national and international climate targets. In response, governments have strengthened regulatory frameworks through increasingly stringent emission standards, low-emission zones, and zero-emission vehicle mandates [19, 25]. Among the proposed decarbonization pathways, electrification, hybridization, and hydrogen fuel-cell technologies have emerged as promising solutions. Battery-electric HCVs, in particular, offer zero tailpipe emissions, high drivetrain efficiency, and the potential for substantially reduced lifecycle emissions when supplied with low-carbon electricity [85, 101]. Advances in battery technology, charging infrastructure, and electric powertrain technologies have expanded the applicability of electric trucks beyond urban and regional operations, while regulatory incentives, sustainability goals, and the prospect of lower operating costs have encouraged their adoption by logistics operators [82, 5, 101, 28]. Nevertheless, battery-electric HCVs still account for only a small share of new truck sales in Europe, and their widespread deployment remains constrained by challenges including high upfront costs, charging infrastructure availability, charging times, driving range limitations, and payload penalties associated with battery mass. Consequently, significant technological and operational developments are still required to enable the large-scale electrification of freight transport and realize its full decarbonization potential.

1.1 Background

1.1.1 Evolution of electric powertrain architectures in HCVs



Figure 1.1: Illustration of a 4x2 vehicle configuration, with chassis mounted prime mover and rear axle drive; (left): Historical diesel engine layout on Volvo ÖV4 TV pickup vehicle from 1927 [110]. (right): DAF LF Battery Electric Innovation Truck exhibited at IAA 2018 [35].

In the 21st century, HCV powertrain architectures have historically been shaped by the operating characteristics of diesel engines, which dictated vehicle geometry, component placement, and functional integration. Narrow efficiency ranges, high torque transmission requirements, vehicle dimensions and substantial cooling and exhaust needs led to standardized layouts featuring longitudinal engines, multi-speed transmissions, mechanical drivelines, and rear differentials [40, 39]. As manufacturing infrastructure, supplier networks, regulatory frameworks, and maintenance practices co-evolved with diesel technology, early electrification efforts by established HCV manufacturers largely focused on adapting existing ICE-based platforms rather than pursuing clean-sheet designs [15, 98]. Initial hybrid, battery-electric, and fuel-cell HCVs therefore followed retrofit-oriented development strategies, integrating electric machines into conventional chassis with minimal geometric modification [15, 91, 111]. While this approach reduced development cost and industrial disruption, it imposed inherent compromises.

Electric machines deliver high torque from standstill and maintain high efficiency over wide speed ranges, reducing the need for multi-speed transmissions [71]. A powertrain concept exploiting these features already existed in the 1900s, using wheel hub motors, as shown in Figure 1.2. Nevertheless, early electric trucks often retained conventional gearboxes due to chassis compatibility and manufac-

turing legacy, introducing unnecessary mass and efficiency losses [70, 96]. Battery integration was similarly constrained by ICE-oriented chassis layouts, resulting in distributed battery installations that increased thermal management complexity and reduced usable payload compared to centralized, purpose-designed configurations [10, 54, 65, 56].

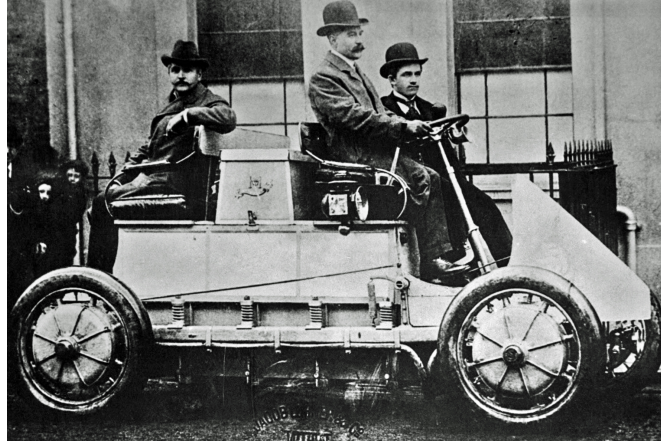


Figure 1.2: Lohner–Porsche Electromobile mixed hybrid using electric wheel hub motors [69]. Photograph from 1902.

As electric machine performance, battery energy density, and power electronics matured, battery-electric HCVs increasingly adopted purpose-designed architectures optimized for electric powertrain [10, 107]. Parallely, new OEMs and electric-first manufacturers such as Tesla Inc, BYD Auto Co., Ltd, and Windrose Technology, among others launched new vehicle platforms [41, 11, 116]. Alongside them established manufacturers including Daimler and Iveco also pursued platform-level redesign rather than ICE-derived adaptations [17, 45]. Electric motor placement evolved from centralized motor–reducer units toward multi-motor and distributed powertrain configurations enabling independent torque control, improved traction, and enhanced vehicle stability [107, 117]. This evolution reflects a broader shift toward vehicle architectures incorporating multiple electric powertrains. In this context, new market entrants have predominantly pursued clean-sheet electric vehicle designs unconstrained by legacy architectures, while established vehicle manufacturers have tended to follow evolutionary, ICE-centric pathways to leverage existing assets.

More recently, some of the established HCV OEMs, new entrants and Tier-1 suppliers have introduced integrated axle electric drive units, commonly referred to as e-axles [115]. These architectures distribute compact electric machines with integrated transmissions across multiple axles and have been demonstrated on limited vehicle configurations with promising performance and efficiency results.

This development signals a clear shift away from diesel-centric adaptations toward purpose-built electric platforms for future heavy-duty transport, although broader adoption remains influenced by regulatory frameworks, service practices, and industry standards rooted in conventional vehicle design [79, 99].

1.1.2 Design and configuration complexity of HCVs

Designing HCVs is a complex engineering task due to the wide diversity of vehicle configurations and application-specific requirements [16, 112]. HCVs operate in multiple roles, each demanding unique performance, durability, and operational characteristics. These variations strongly influence chassis layout, axle arrangement, suspension design, load distribution, and overall vehicle stability.

The number and arrangement of axles and deciding which should be propelled, (e.g., 4×2 , 6×4 , 8×4 , or 10×4) directly affect load distribution, traction, turning radius, and packaging constraints [16]. Multi-axle vehicles improve payload capacity and stability but introduce challenges in suspension design, steering geometry, and integration of components such as fuel tanks, batteries, or e-axle systems [6, 109]. Additional axles often reduce available space for equipment, increasing trade-offs in vehicle layout and overall design.

Depending on the application, the HCVs can be broadly classified as the following:

- **Rigid Trucks:** Typically operate in urban or regional routes, requiring frequent acceleration and braking. Design priorities include maneuverability, high efficiency, and regenerative braking capability [122].
- **Tractor-Trailers:** Long-haul freight vehicles demand high payload capacity, large energy storage systems (e.g., 550–650 kWh for electric variants), and optimized aerodynamics to maximize range [122, 106].
- **Mining and Construction Vehicles:** Operate in harsh environments with low-speed, high-load conditions. Design must ensure extreme torque, robust cooling, and redundancy for safety and reliability [6, 109].

Overall, the interplay between axle configuration and application type creates a highly complex design environment for HCVs. Balance of features like payload capacity, stability, maneuverability, and operational efficiency while meeting regulatory requirements and infrastructure constraints is an engineering challenge.

1.1.3 Technological advances in HCV actuator systems

HCVs also face increasing demands for safety, energy efficiency, and adaptability to variable payloads. These challenges have led to exploration toward electri-

cally and electronically actuated architectures, from the conventional mechanical and passive actuated systems. This shift has been accelerated by the widespread adoption of transport electrification, advanced driver assistance systems, and autonomous vehicle initiatives, which require higher levels of actuator precision, responsiveness, and integration. As a result, modern heavy vehicles are increasingly relying on coordinated actuator control and exhibiting over-actuated architectures, enabled by redundant and distributed powertrain actuators, braking, and suspension actuators [113, 123, 119]. This added flexibility improves traction and vehicle stability and lays the foundation for distributed drive architectures to optimize energy efficiency, stability, comfort and overall system coordination.

Electric machines, due to their inherent symmetric four-quadrant torque–speed operation, enable control strategies that treat axle-level actuators as independent. This over-actuation capability allows torque to be allocated dynamically according to vehicle states, load conditions, and traction limits [117]. In HCV applications, several types of electric machines are employed, each offering distinct advantages. Permanent magnet synchronous machines (PMSMs) are widely used due to their high torque density, high efficiency, and excellent low-speed performance, making them well suited for traction applications [48, 21]. Electrically excited synchronous machines (EESMs) offer comparable controllability while eliminating the need for permanent magnets, thereby reducing dependency on rare-earth materials and enabling adjustable field excitation for extended constant-power operation and improved high-speed efficiency. Induction machines (IMs) provide robust operation, lower material cost, and high fault tolerance, which are attractive characteristics for heavy-duty and harsh operating environments [21, 59]. Switched reluctance machines (SRMs) have also gained increasing attention due to their simple and rugged construction, absence of permanent magnets, and high thermal tolerance, although they require advanced control strategies and may exhibit higher torque ripple [32]. The availability of multiple electric machine technologies provides flexibility in powertrain design and supports over-actuated architectures by enabling independently controllable propulsion actuators across multiple axles.

The enhanced controllability provided by electrification extends to suspension, steering and braking systems. Active and semi-active suspension actuators, such as electro hydraulic struts, electro mechanical dampers and hydro pneumatic suspensions, enable independent modulation of vertical and roll forces, improving ride comfort, load distribution, and stability [119, 63]. Electrically actuated steering systems, including electric power steering and steer-by-wire architectures, further enhance vehicle controllability by enabling precise and software-configurable lateral force generation, while providing the redundancy and fault tolerance required for automated driving and safety-critical control, thereby contributing to overall vehicle over-actuation [77]. Next generation braking systems, including brake-by-wire (BBW) and electromechanical braking (EMB) actuators, allow precise

and rapid brake force distribution across axles, supporting stability interventions, anti-rollover strategies, and coordinated vehicle motion control [34, 120]. By treating powertrain, suspension, and braking actuators as an integrated system, over-actuated architectures can dynamically reallocate actuator efforts to satisfy multiple objectives—safety, comfort, and energy efficiency—even under varying payloads or in the presence of actuator constraints or faults [123, 63].

1.2 Research motivation for electrified HCVs

Despite their promising potential, the transition from conventional diesel-powered heavy commercial vehicles to zero-emission alternatives remains a significant challenge. Battery-electric, hybrid, and fuel-cell-powered HCVs face a range of technical, operational, and economic constraints, including high capital costs, limited energy storage capacity, charging infrastructure requirements and load capacity challenges. Addressing these challenges is crucial for enabling the widespread adoption of zero-emission technologies across a broad spectrum of freight transport applications.

1.2.1 Operational and economic factors in electric HCV design

Driving range and payload capacity remain central performance metrics for electrical HCVs. While diesel vehicles have historically dominated the operational fleet, due to high fuel energy density and mature refuelling infrastructure, tightening emission regulations across the EU and globally are accelerating the shift toward battery-electric and hydrogen fuel-cell technologies [4, 25].

For BEVs, achieving competitive range typically requires large battery packs. However, increased battery capacity adds significant mass, directly reducing payload—the primary source of freight revenue [65, 90]. Range is further influenced by battery efficiency, aerodynamics, terrain, temperature, charging availability, and the overall efficiency of the powertrain components [92, 108].

Beyond performance, total cost of operations (TCOP) is a principal driver of BEV adoption. TCOP usually includes energy use, maintenance, battery replacement, tyre wear, charging costs, driver salary, and operational reliability [4, 74]. Even small improvements in drivetrain efficiency or energy utilization can yield substantial cost savings over the high annual mileages typical of freight transport. Key factors affecting TCOP include:

- **Energy and Efficiency:** Optimized torque distribution, reduced electrical and mechanical power losses, and effective regenerative braking improve energy

efficiency and reduce long-term operating costs [53, 121, 108]. Efficiency improvements also reduce required battery size for a given duty cycle, lowering capital cost.

- **Battery–Payload Trade-off:** Larger batteries extend range but reduce payload and increase depreciation costs. Studies show that battery mass has a measurable impact on freight capacity and revenue potential [65, 75]. Improved energy efficiency and route optimization enable smaller battery packs without compromising range.
- **Maintenance and Tyres:** Although BEVs have fewer moving parts than ICE vehicles, high torque levels and increased vehicle weight can accelerate tyre wear [67, 68]. Advanced control strategies—such as optimal traction control and axle load management—help reduce wear and maintenance needs [87].
- **Charging costs:** High-power and megawatt-scale charging infrastructure requirements are strongly influenced by vehicle efficiency. Improved efficiency reduces peak charging power, operational downtime, and infrastructure investment [76, 124].

1.2.2 Safety and dynamic performance in electric HCVs

HCVs are critical for freight transport but are associated with disproportionately severe accidents. Despite representing less than 3% of the European vehicle fleet, HCVs are involved in approximately 15% of road fatalities [24]. Their large mass, high payloads, and dynamic load characteristics make collisions inherently more dangerous. Conventional diesel-powered HCVs rely on mechanical systems - diesel engine, retarders, differentials and pneumatic brakes - which limit stability features and dynamic control under high load, low-adhesion, or emergency conditions [1, 105]. Such limitations contribute to rollovers, skidding, trailer sway, and loss-of-control incidents.

Electrification introduces a transformative potential for dynamic vehicle safety. Electric traction motors offer rapid torque response, fine controllability, and independent wheel- or axle-level torque distribution [22, 33]. These capabilities enable torque vectoring, actively adjusting driving and braking torque at each wheel to improve stability, traction, and handling. In heavy vehicles, torque vectoring can counteract understeer or oversteer, prevent rollovers and trailer sway, mitigate jackknifing in articulated vehicles, and enhance traction on wet, snowy, or uneven terrain — all without sacrificing speed or efficiency [3, 51].

Additional benefits include multi-axle regenerative braking and coordinated brake-blending algorithms, which smoothen deceleration, reduce thermal stress

on mechanical brakes, and maintain load stability during heavy braking or downhill operations [118]. Future HCV platforms with integrated e-axles or in-wheel motors enable wheel-level control, simplify drivetrain design, enhance fault tolerance, and support advanced driver-assistance systems (ADAS) and autonomous functions [22].

Given these advances, a systematic integration of such technologies into electric HCVs is warranted, with a focus on dynamic performance and safety, to evaluate how electrification and torque vectoring can reduce accident risk and enhance vehicle stability beyond the limits of conventional mechanical systems.

1.2.3 Opportunities of integrated actuator coordination in HCVs

The transition toward electrified, automated, and software-defined heavy vehicles will increase the number and diversity of onboard actuators, including independently controllable powertrain actuators, brake-by-wire and electromechanical braking systems, electrically actuated steering, and active suspension. These technologies enhance vehicle safety, controllability, and efficiency but introduce significant actuation redundancy, resulting in over-actuated architectures.

In such systems, multiple actuators can influence the same vehicle motion objectives, such as longitudinal force, lateral force, or yaw moment. Control allocation¹ addresses this challenge by systematically distributing desired vehicle-level forces among redundant actuators while respecting actuator constraints, dynamics, and saturation limits. Classical and optimization-based control allocation methods, including pseudo inverse and quadratic programming approaches, provide the theoretical foundation for actuator coordination, while model predictive control allocation (MPCA) extends these concepts to explicitly handle constraints and actuator dynamics in real time [97, 94, 36].

In the context of distributed electric vehicles, control allocation has been widely applied to torque vectoring and energy-efficient wheel force distribution. Optimal torque distribution strategies for overactuated electric vehicles have demonstrated improvements in stability and energy efficiency [55, 95], while adaptive and energy-aware MPCA frameworks further enhance performance under varying operating conditions and payloads [103].

For heavy vehicles in particular, integrated actuator coordination using control allocation enables the simultaneous management of propulsion, braking, steering, and suspension systems. This improves stability, energy efficiency, and fault tolerance, and allows high-level motion controllers to focus on trajectory and vehicle-

¹Mathematically, over-actuation leads to an underdetermined system, with more actuator inputs than independent control objectives. Control allocation selects a feasible or optimal actuator input vector from the infinitely many solutions while satisfying system constraints.

level objectives without being burdened by actuator-level constraints. The importance of such strategies is further amplified in autonomous or highly automated vehicles, where precise trajectory tracking, fail-operational behavior, and robustness under varying road friction and load conditions are essential.

1.3 Research questions

The design of heavy vehicles is inherently complex, involving trade-offs between operational efficiency, cost, and safety. The transition to electrified powertrains introduces new opportunities by enabling propulsion to be distributed across multiple axles, increasing actuation redundancy and flexibility in control. This over-actuated architecture allows additional objectives—beyond conventional motion control—to be addressed, such as energy efficiency, component wear, and overall system performance. Effectively exploiting these opportunities, however, requires a systematic and concept-driven design approach supported by suitable modeling and control strategies.

To this end, this thesis simplified system-level representations of electric heavy vehicle platforms and investigates the following research questions (RQs):

- **RQ1:** How should the electric powertrain of on-road electric HCVs be configured to satisfy vehicle motion requirements across different vehicle configurations having multiple electrically propelled axles ? For example: On a 3-axle truck which axles should be configured with electric machines to increase driving range ?
- **RQ2:** Which control strategy is best suited to instantaneously coordinate the axles and actuators of a proposed powertrain system while improving driving range and tracking motion requests ?
- **RQ3:** How can lateral tyre grip be maintained during coordinated axle and actuator control on multiple electrically propelled axles to ensure laterally stable and predictable vehicle behaviour ? Additionally, how can longitudinal safety be ensured during combined deceleration and cornering manoeuvres ?
- **RQ4:** How can operational cost rates, such as energy consumption and tyre wear, be incorporated into coordinated axle and actuator control to minimise the total operational costs of the vehicle ?

1.4 Limitations

The scope of research was conducted with certain assumptions and limitations. Here a list of important limitations are noted:

- The studies are limited to on-road vehicle applications. The analysis is restricted to forward driving conditions, and standstill operations are not considered. Driving in extreme weather conditions like heavy rain, side winds and off-road conditions are neglected.
- The primary focus of this thesis is on single vehicle units, namely solo tractors and rigid trucks. The only exception is Paper D, which includes a tractor–semi-trailer combination to represent a realistic freight transport application while avoiding complex manoeuvres and vehicle combination dynamics, such as yaw instability, rollover, and jack-knifing. Furthermore, only towing vehicles with steering actuation on the front axle group are considered. Thus, this thesis focuses on axle and actuator coordination within individual vehicle units, whereas the work in [23] focuses on combination coordination between vehicle units.
- While configuring the powertrain system, geometric size of powertrain components, battery and packaging constraints were not explicitly considered. However, unrealistic or extreme powertrain specifications were avoided based on engineering judgement. In addition, the effect of component sizing on mass and inertia of components are also neglected which is a simplification but assumed acceptable for conceptual studies.
- The control algorithms developed in this thesis are restricted by only considering vehicle motion in the road plane, neglecting roll-over effects. For the development of the control systems, it is assumed that the desired vehicle states are available, although the estimation or accessibility of these states is not addressed within the scope of this work. Furthermore, only instantaneous actuator control is considered; consequently, long-horizon predictive control strategies are excluded except in Paper 5. Within the assumed hierarchical control architecture, the upper control layers generate motion requests, while the lower-level controller is responsible for allocating the actuator commands.
- Predictive motion control is not considered in this thesis. Instead, the instantaneous velocity and acceleration demands are assumed to be given by a higher-level controller, such as a predictive cruise controller, or by the driver. This separation is motivated by the findings of Torinsson et al.[104], who

showed that velocity planning and torque distribution can be formulated as separate hierarchical problems, with predictive velocity planning followed by instantaneous torque distribution. Since this thesis considers a single energy source and does not address hybrid drivetrains or the blending of multiple energy sources, the focus is therefore placed on the instantaneous torque distribution problem for a prescribed velocity and acceleration demand.

- Since the purpose of thesis was to evaluate conceptual powertrain design there are no physical prototype available for the different configurations presented. Thus, the main results presented in this thesis are obtained through simulation studies. Consequently, the scalable models of the electric machines and algorithms developed were not validated as part of this work. Only electric machines and friction brakes are considered as controllable actuators in this thesis. Suspension and steering control are not included in the developed control algorithms. Steering actuation is assumed to be regulated by the driver model. Furthermore, losses associated with thermal management systems and auxiliary loads are excluded from the analysis.
- The proposed control strategies in thesis are not intended to replace standard active safety systems, such as traction control (TC), anti-lock braking system (ABS) and electronic stability control (ESC). Instead, it is assumed that these systems remain active and available during vehicle operation and override the proposed controllers.

1.5 Main Contributions

A summarized list of the scientific contributions of this thesis is presented below.

- A modular distributed electric powertrain architecture for long-haul heavy electric vehicles is established. The key idea is to separate cruise and startability functions across dedicated axles, enabling a more flexible operating-mode-based propulsion system design. On top of this, a joint hardware-control optimisation framework is introduced for selecting motor ratings and gear ratios. The results show clear potential for improved energy efficiency and extended driving range. (**Paper A and Chapter 3**).
- An optimisation-based actuator coordination framework for motion control and power coordination is formulated. The main contribution is the systematic use of constrained power-loss minimisation, e.g. through quadratic programming, to coordinate multiple actuators on multiple axles. This allows propulsion and braking forces to be distributed while explicitly respecting

actuator limits and reducing power losses. (**Paper B & C and Chapter 4**)

- A simulation-based evaluation framework for wheel torque allocation strategies in heavy electric vehicles is presented. It enables structured comparison of different strategies in terms of energy efficiency, longitudinal tyre slip, and actuator usage. The results further highlight that including electric machine idle losses in the coordination can yield measurable energy savings compared to safety and wear oriented approaches. (**Paper D and Chapter 4**)
- A hierarchical wheel torque coordination framework for heavy electric vehicles is introduced. It combines adaptive model predictive control (MPC) at the vehicle level with a lower-level control allocation layer. Together, this enables energy-efficient torque distribution while maintaining stability and predictable steering behaviour, even in low-friction and highly dynamic scenarios. (**Paper E and Chapter 5**)
- A unified operational-cost-based actuator coordination framework for heavy electric vehicles is presented. The main idea is to extend energy-based actuator and axle coordination framework by including tyre wear in the optimisation objective. Both tyre degradation and energy losses are expressed in monetary terms representing cost-rates, enabling a consistent trade-off. The framework shows how torque allocation can reduce total operating cost by balancing energy consumption and tyre wear. (**Paper F & G and Chapter 6**)

1.6 Outline

This thesis is organised into seven chapters, including this introductory chapter. Chapter 2 presents the modelling framework used throughout the thesis, including vehicle, actuator, and powertrain models for analysis, simulation, and control design. Chapter 3 addresses the **RQ1** by investigating modular electric powertrain architectures for heavy commercial vehicles and by analysing how dedicated cruise and startability axles can be configured to satisfy operating requirements. Chapter 4 addresses the **RQ2** by formulating wheel torque coordination as an optimisation-based actuator coordination problem and by evaluating alternative allocation strategies with respect to power losses and energy efficiency. Chapter 5 addresses the **RQ3** by extending actuator coordination toward safe vehicle motion under tyre-force limitations through hierarchical control and predictive force generation. Chapter 6 addresses the **RQ4** by extending actuator coordination

from power-loss minimisation to operational-cost minimisation through the inclusion of tyre wear. Finally, Chapter 7 summarises the conclusions of the thesis and outlines directions for future work.

Parts of Chapter 2 on axle groups definition and Chapter 5 on - an alternative allocation strategy for increasing longitudinal braking capability during combined braking and cornering while maintaining yaw stability, are not appended in any papers. The contributions from and relation between the Papers and thesis chapters are presented in Figure 1.3.

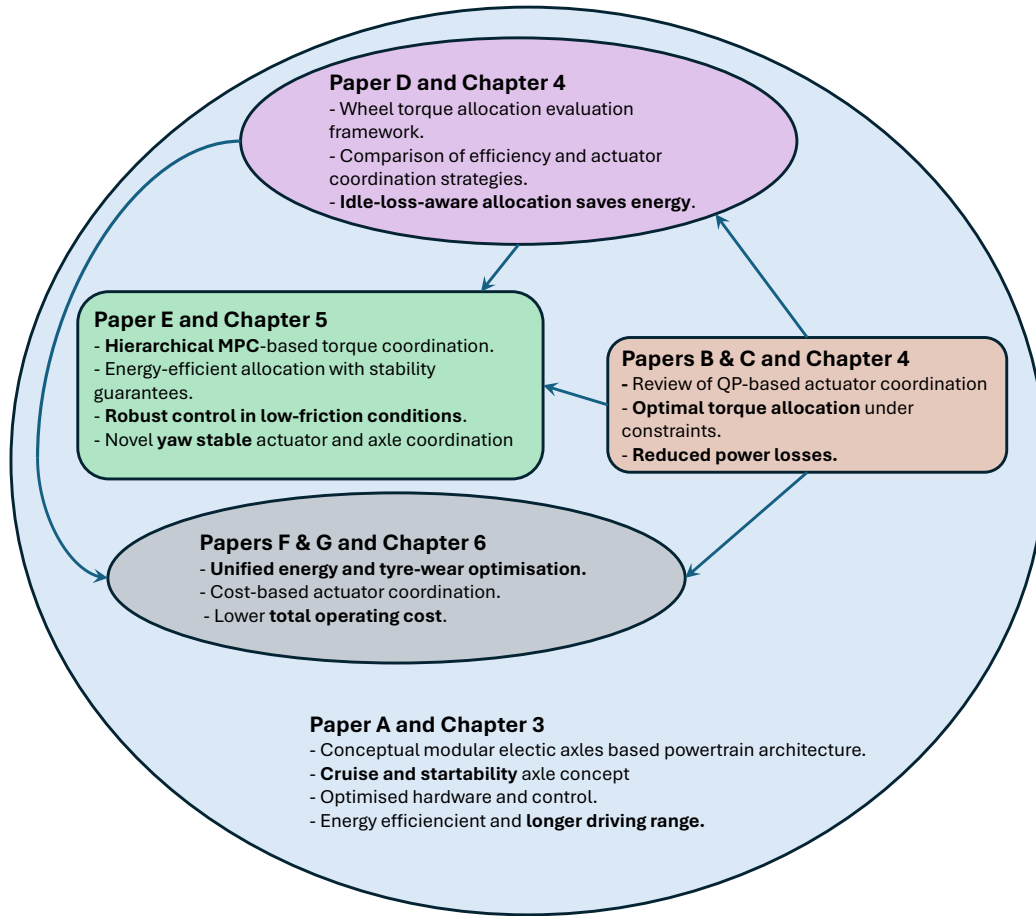


Figure 1.3: Highlights and contributions from the different papers and the chapters within the thesis. The arrows here presents the interconnections, reuse, extension of optimisation frameworks, simulation models, and methodologies across the studies.

Chapter 2

Vehicle modelling

Vehicle modelling is a fundamental step in the analysis, simulation, and control design of automotive systems. A vehicle model mathematically describes the dynamic behaviour of a vehicle by representing the relationships between forces, motions, and control signals. The required modelling fidelity depends on the intended application. Two broad types of vehicle models can be distinguished based on their intended use:

- **High-fidelity models:** These models aim to capture a large number of physical phenomena, often including detailed representations of vehicle components and their interactions. They can accurately reproduce vehicle behaviour but are typically computationally demanding, time-consuming to parameterise, and not readily invertible. They are therefore well suited as plant models for evaluating control designs under realistic operating conditions.
- **Purpose-specific models:** These models represent only the physical phenomena considered relevant to the intended analysis or design task. By deliberately neglecting phenomena that are not essential for the task, they can be computationally efficient and, where appropriate, formulated in an invertible form. They are therefore particularly suitable for design, including control design.

In the context of heavy electric vehicles equipped with distributed propulsion systems, vehicle modelling becomes particularly important for the development of motion control and power coordination strategies. To support the development and evaluation of these control strategies, appropriate vehicle modelling is required. The modelling approach adopted in this thesis includes three main components:

- Steady state ground plane vehicle dynamic models used within the control algorithms to achieve wheel torque control through actuator coordination.

- Minimalistic and high-fidelity vehicle models describing the motion of the vehicle body as the plant model.
- Models representing electric powertrain and braking systems in both control algorithms and as actuators.

This modelling structure used in this thesis captures the essential vehicle dynamics and actuator behaviour while maintaining a computational complexity suitable for simulation and control design.

2.1 Model indices and notation

All vehicle models in this thesis are formulated using the ISO 8855 vehicle-fixed coordinate system. The longitudinal, lateral, and vertical axes of the coordinate system are denoted by x , y , and z , respectively. The x -axis points in the forward direction of the vehicle, the y -axis points towards the left side of the vehicle, and the z -axis points upwards. Accordingly, positive longitudinal forces act in the forward direction, whereas positive lateral forces act towards the left side of the vehicle. The wheel-side index $l \in \{L, R\}$ refers to the left and right sides of the vehicle according to this coordinate convention.

To maintain a consistent notation throughout this thesis, a generalised indexing convention is adopted for the different vehicle models. Indices are introduced only when they distinguish physically different components and are omitted whenever they are unnecessary.

For a general multi-axle vehicle, the following indices are used:

- i : axle index,
- j : actuator index associated with axle i ,
- l : wheel-side index associated with axle i .

The axle index is defined as

$$i \in \{1, 2, \dots, n\}, \quad (2.1)$$

where n denotes the number of axles. The actuator index is defined as

$$j \in \{1, 2, \dots, m_i\}, \quad (2.2)$$

where m_i denotes the number of actuators associated with axle i . The wheel-side index is defined as

$$l \in \{L, R\}, \quad (2.3)$$

where L and R denote the left and right wheels, respectively.

Accordingly, $T_{EM,ij}$ denotes the torque produced by electric machine j associated with axle i , while $F_{x,il}$, $\omega_{w,il}$, and κ_{il} denote the longitudinal tyre force, wheel angular velocity, and longitudinal slip ratio of wheel l on axle i , respectively.

In general, the actuator index j is used only for actuator-specific variables, whereas the wheel index l is used only for wheel-specific quantities. Parameters that are common to all actuators or wheels on an axle, such as the transmission gear ratio gr_i , transmission efficiency $\eta_{trmn,i}$, effective wheel radius $r_{w,i}$, and axle normal load $F_{z,i}$, are indexed only by the axle index i . When these parameters are identical for all axles, the axle index may also be omitted for simplicity.

Additionally, the generalised notation is simplified according to the vehicle model employed. For the single-wheel vehicle model, only one equivalent wheel is considered. Consequently, all axle, actuator, and wheel indices are omitted unless required for clarity.

For the single-track model, equivalent front and rear axle groups are introduced, and only the axle index is retained. In this case,

$$i \in \{f, r\}, \quad (2.4)$$

representing the front and rear axle groups, respectively. Actuator and wheel indices are omitted because the model represents equivalent axle-level quantities.

For the distributed electric propulsion (DEP) architecture considered in this thesis, the axle index is typically defined as

$$i \in \{crs, stb\} \quad (2.5)$$

representing the cruise and startability axles. When a single electric machine is associated with each axle, the actuator index j is omitted, giving quantities such as $T_{EM,i}$ and $P_{EM,i}^{\text{loss}}$. Similarly, wheel indices are omitted whenever axle-level quantities are sufficient. When left and right wheel quantities are explicitly considered, the wheel-side index $l \in \{L, R\}$ is retained.

This notation is used consistently throughout the thesis and provides a unified description ranging from simplified single-wheel models to generalised multi-axle vehicles with multiple actuators and independently driven wheels, while avoiding unnecessary notation.

2.2 Axle groups on heavy vehicles

Heavy vehicles typically employ multiple axles to increase load-carrying capacity and comply with road load regulations. These axles appear in the form of steered, driven, and non-driven axles (tag or pusher) and are often arranged in tandem or tridem configurations, resulting in several closely spaced axles at the front and/or

rear of the vehicle. Modelling each axle individually increases model complexity and is often unnecessary for control-oriented applications. Therefore, axles are commonly grouped into front and rear axle groups as shown in Figure 2.1.

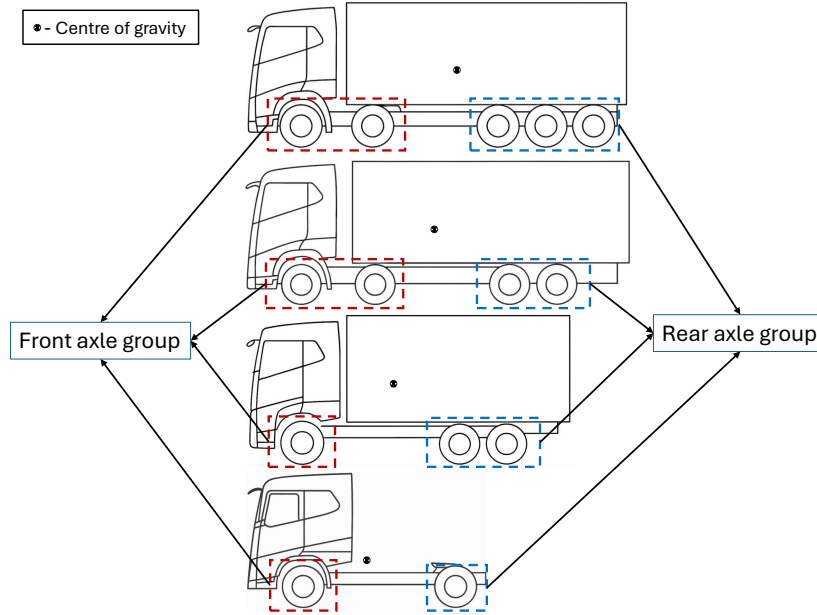


Figure 2.1: An illustration of the distribution of axles into front and rear axle groups for different vehicle configurations with respect to the centre of gravity.

The axle group formulation is particularly useful for control design, where the vehicle dynamics are described using aggregated front and rear axle group forces, and a separate control layer distributes the requested forces among individual axles or wheels for each group. This approach enables efficient modelling of multi-axle heavy vehicles while maintaining computational tractability.

Each axle group represents one or more physical axles whose tyre forces are aggregated into equivalent longitudinal and lateral forces acting at representative locations with respect to the vehicle's centre of gravity (CoG). The total force generated by a group is obtained as the sum of the forces from the individual axles (assuming they are located close to each other), while the corresponding yaw moment is determined by their longitudinal positions. This grouping preserves the structure, allowing multi-axle heavy vehicles to be represented in a compact form.

The axle index i is therefore grouped as:

$$i \in \mathcal{F} \quad (\text{front axle group}), \quad i \in \mathcal{R} \quad (\text{rear axle group}) \quad (2.6)$$

where the index sets are given by:

$$\mathcal{F} = \{1, \dots, n_f\}, \quad \mathcal{R} = \{n_f + 1, \dots, n\} \quad (2.7)$$

with n_f denoting the number of front axles, n_r the number of rear axles, and the total number of axles defined as:

$$n = n_f + n_r \quad (2.8)$$

Each axle i is located at a longitudinal distance l_i from the vehicle CoG.

As an example, consider a three-axle vehicle with one front axle and two rear axles. In this case, $n_f = 1$ and $n_r = 2$, resulting in

$$\mathcal{F} = \{1\}, \quad \mathcal{R} = \{2, 3\} \quad (2.9)$$

where axle 1 corresponds to the steered front axle, and axles 2 and 3 form the rear axle group. The corresponding longitudinal positions are given by $l_1 > 0$ and $l_2, l_3 < 0$.

2.2.1 Axle and axle group forces and their limits for multi-axle vehicles

The planar force capability of multi-axle vehicles can be approximated using a single-track representation. When adjacent axles within an axle group are closely spaced and experience similar slip conditions, they may be further combined into equivalent front and rear axle groups. as shown in Figure 2.2. For these models to be valid the axle spacing within an axle group should be close enough typically $\leq 1.5m$. This approach preserves the structure of the conventional single-track model while capturing the effect of multiple axles distributed along the vehicle body. The model assumes left-right symmetry and combines the wheels of each axle into a single equivalent wheel located at the vehicle centreline. The forces generated by all axles belonging to the front and rear groups are summed to obtain equivalent front and rear tyre forces. Within the single-track representation, each physical axle is replaced by an equivalent wheel acting at the vehicle centreline. Consequently, the quantities $F_{x,i}$, $F_{y,i}$ and $F_{z,i}$ denote model-equivalent longitudinal, lateral, and normal tyre forces associated with axle i . These are lumped variables of the reduced-order single-track model and do not represent the individual tyre forces of the physical vehicle.

Each axle i is modelled as an equivalent wheel located on the vehicle centreline with normal load $F_{z,i}$. The front axle group consists of n_f axles located ahead of the centre of gravity, while the rear axle group consists of n_r axles located behind the centre of gravity. The model-equivalent longitudinal and lateral tyre forces

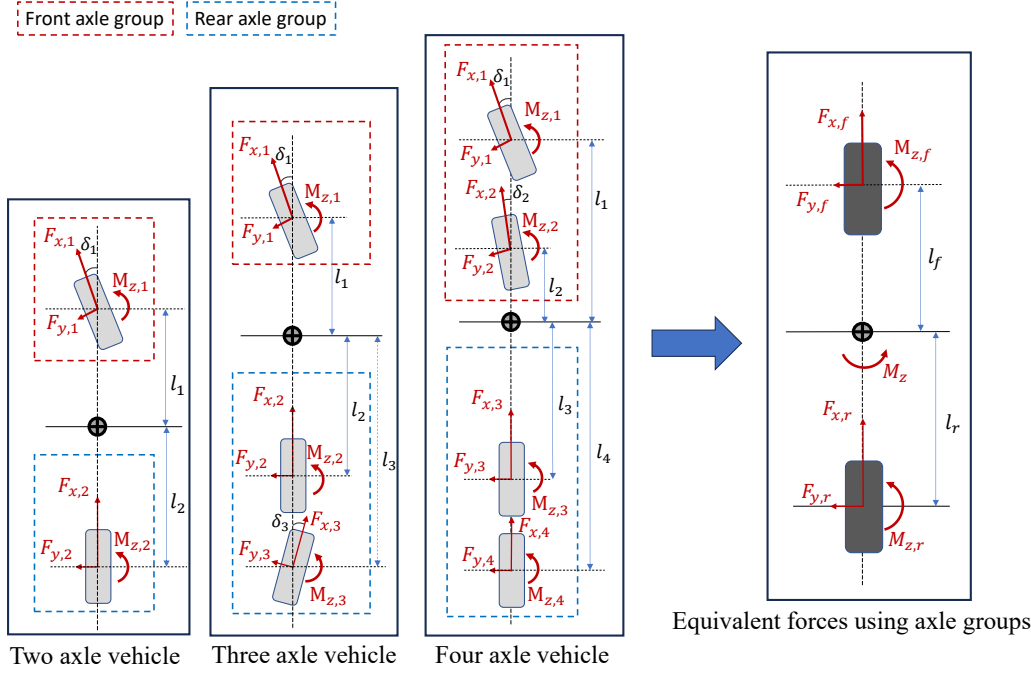


Figure 2.2: Representation of a multi-axle vehicle by a two-axle single-track model with equivalent front and rear axle group forces.

associated with axle i are denoted by $F_{x,i}$ and $F_{y,i}$, respectively. The total forces acting on the front and rear axle groups, f and r respectively, are obtained as:

$$F_{x,f} = \sum_{i \in \mathcal{F}} F_{x,i}, \quad F_{x,r} = \sum_{i \in \mathcal{R}} F_{x,i} \quad (2.10)$$

$$F_{y,f} = \sum_{i \in \mathcal{F}} F_{y,i}, \quad F_{y,r} = \sum_{i \in \mathcal{R}} F_{y,i} \quad (2.11)$$

where $\mathcal{F}$ and $\mathcal{R}$ denote the sets of front and rear axles, respectively. The resulting forces act at equivalent distances l_f and l_r from the centre of gravity.

The equivalent distances for the front and rear axle groups are defined using moment equivalence. Thus the aggregated lateral force of each axle group is assumed to act at an equivalent distance from the centre of gravity that produces the same yaw moment as the individual axle forces. The equivalent front and rear axle group distances are defined as

$$l_f = \frac{\sum_{i \in \mathcal{F}} l_i F_{y,i}}{\sum_{i \in \mathcal{F}} F_{y,i}} \quad (2.12)$$

$$l_r = \frac{\sum_{i \in \mathcal{R}} l_i F_{y,i}}{\sum_{i \in \mathcal{R}} F_{y,i}} \quad (2.13)$$

where l_i denotes the longitudinal position of axle i relative to the CoG. These distances represent the effective application points of the aggregated lateral forces for the front and rear axle groups, respectively.

The border of admissible force region on an axle (by lumping the wheel forces) are usually constrained and can be approximated using a simplified representation of a friction circle as:

$$F_{x,i}^2 + F_{y,i}^2 \leq (\mu F_{z,i})^2 \quad (2.14)$$

where μ is the tyre–road friction coefficient. For driven axles, both longitudinal and lateral forces are generated, resulting in a circular admissible force region in the road plane. In contrast, non-driven axles [9] such as tag or pusher axles do not generate longitudinal forces, except during braking and therefore satisfy

$$F_{x,i} = 0, \quad |F_{y,i}| \leq \mu F_{z,i} \quad (2.15)$$

which corresponds to a line segment in the force plane.

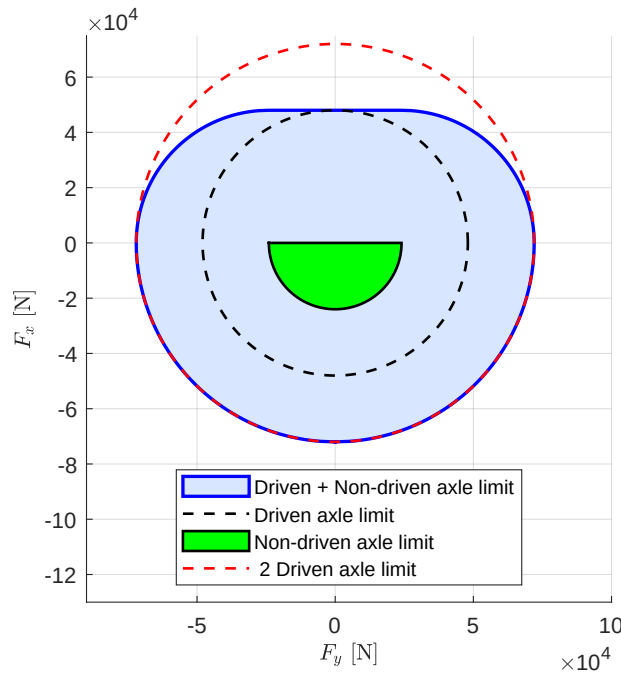


Figure 2.3: Comparison of longitudinal and lateral tyre force limits for tandem axle configurations comprising two driven axles versus one driven and one non-driven axle. Axle normal loads are set to 60 kN and 30 kN, respectively. A tyre–road friction coefficient of $\mu=0.8$ is assumed. The forces on the driven axles are friction-limited as opposed to propulsion-limited. F_x and F_y denote the maximum longitudinal and lateral forces.

When multiple axles are grouped together, the admissible force region of the axle group is obtained by combining the individual axle limits. This can be interpreted geometrically as the Minkowski sum of the individual feasible sets. As an illustrative example, consider a driven axle combined with a non-driven tag axle for a rear axle group. The driven axle, denoted by subscript '*dr*', is limited by a friction circle, while the tag axle, denoted by subscript '*tg*', contributes only to lateral force and negative longitudinal force. The resulting axle-group limits therefore correspond to a lateral expansion of the driven axle friction circle. The admissible force region becomes

$$F_{x,r} = F_{x,dr} + F_{x,tg} \quad (2.16)$$

$$F_{y,r} = F_{y,dr} + F_{y,tg} \quad (2.17)$$

$$|F_{y,tg}| \leq \sqrt{(\mu F_{z,tg})^2 - F_{x,tg}^2} \quad |F_{y,dr}| \leq \sqrt{(\mu F_{z,dr})^2 - F_{x,dr}^2}$$

which yields the combined constraint.

$$|F_{y,r}| \leq F_{y,dr} + F_{y,tg} \quad (2.18)$$

This results in a light blue shaded admissible force region. The corresponding geometry is illustrated in Figure 2.3, where the black dashed circle denotes the driven axle limits, green shaded region the tag axle limits and the light blue shaded region represents the combined driven-tag axle capability.

The same principle applies to other axle configurations. Multiple non-driven axles, such as tag and pusher axles, further expand the admissible region in the lateral direction

$$|F_{y,r}| \leq \sum_{i \in \mathcal{R}} \sqrt{(\mu F_{z,i})^2 - F_{x,i}^2} + \sum_{i \in \mathcal{N}} \sqrt{(\mu F_{z,i})^2 - F_{x,i}^2} \quad (2.19)$$

where $\mathcal{N}$ denotes the set of non-driven axles. Similarly, for multiple driven axles, the admissible region corresponds to the combination of multiple friction circles, resulting in an increased longitudinal and lateral force capability.

2.3 Vehicle dynamics

2.3.1 Single wheel vehicle model

The single-wheel vehicle model represents the vehicle as a lumped mass with a single equivalent wheel. Only longitudinal motion is considered, while lateral, vertical, and pitch dynamics are neglected. All forces and torques are interpreted as

model-equivalent quantities in a lumped single-wheel representation of the vehicle. The wheel is assumed to roll without longitudinal slip, such that the longitudinal tyre slip is zero. Consequently, wheel rotational dynamics and wheel inertia are neglected. A schematic representation of the model is shown in Figure 2.4.

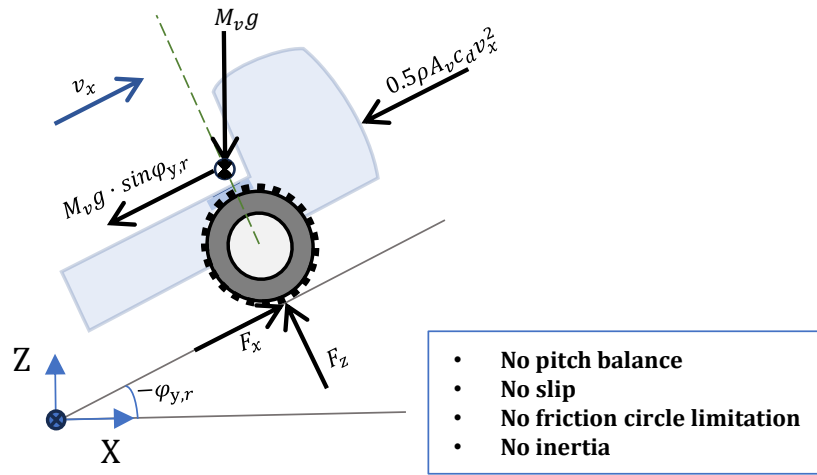


Figure 2.4: Free body diagram of the lumped single wheel vehicle model in X-Z plane with motion constrained to longitudinal direction.

Under these assumptions, the model-equivalent longitudinal tractive force is related to the effective wheel torque according to

$$F_x = \frac{T_w - T_{rr}}{r_w} \quad (2.20)$$

where T_w is the applied wheel torque, r_w is the effective loaded tyre radius, and

$$T_{rr} = r_w F_{rr} = r_w c_{rr} F_z \quad (2.21)$$

is the rolling resistance torque. Here, F_{rr} is the model-equivalent rolling resistance force, c_{rr} is the rolling resistance coefficient, and F_z is the normal force acting on the wheel. Since wheel rotational dynamics are neglected, the wheel torque is assumed to be instantaneously balanced by the rolling resistance torque and the torque required to generate the longitudinal tractive force.

The normal load is assumed constant and equal to the static vehicle weight projected onto the road surface,

$$F_z = M_v g \cos(\varphi_{y,r}) \quad (2.22)$$

which reduces to $F_z \approx M_v g$ for the small road grades considered in this work. The generated tractive force accelerates the equivalent vehicle mass while overcoming aerodynamic drag, rolling resistance, and the component of gravity acting along the road grade. Under these assumptions, the longitudinal force balance is given by

$$F_x = M_v a_x + \frac{1}{2} \rho c_d A_v v_x^2 + M_v g (c_{rr} - \sin(\varphi_{y,r})) \quad (2.23)$$

where M_v denotes the equivalent vehicle mass, v_x is the longitudinal vehicle velocity, a_x is the longitudinal vehicle acceleration, ρ is the air density, c_d is the aerodynamic drag coefficient, A_v is the vehicle frontal area, g is the gravitational acceleration, and $\varphi_{y,r}$ is the road pitch angle.

The longitudinal vehicle model serves two purposes throughout this thesis. First, it is employed as an inverse model to compute the tractive force required to follow a prescribed vehicle speed profile. Second, it is used as the plant model in the closed-loop simulations presented in Papers A and B.

2.3.2 Single track and two-axle vehicle model

A two-axle vehicle with a steered front axle is represented using a single-track model. The vehicle is modelled as a rigid body with three degrees of freedom, where multiple physical axles are lumped into equivalent front and rear axle groups located at distances l_f and l_r from the CoG, respectively as shown in Figure 2.5. The equations of motion are formulated in the body-fixed reference frame.

The equations of motion are given by

$$M_v(\dot{v}_x - v_y \cdot \omega_z) = F_{x,f} \cos(\delta_f) - F_{y,f} \sin(\delta_f) + F_{x,r} \quad (2.24)$$

$$M_v(\dot{v}_y + v_x \cdot \omega_z) = F_{y,f} \cos(\delta_f) + F_{x,f} \sin(\delta_f) + F_{y,r} \quad (2.25)$$

$$I_z \dot{\omega}_z = l_f (F_{y,f} \cos(\delta_f) + F_{x,f} \sin(\delta_f)) - l_r F_{y,r} + M_{z,\text{ext}} \quad (2.26)$$

The tyre slip angles for the front and rear axles are defined as:

$$\alpha_f = \delta_f - \arctan\left(\frac{v_y + l_f \omega_z}{v_x}\right) \quad (2.27)$$

$$\alpha_r = -\arctan\left(\frac{v_y - l_r \omega_z}{v_x}\right) \quad (2.28)$$

The lateral tyre forces are approximated using a linear tyre model

$$F_{y,f} = C_f \alpha_f \quad (2.29)$$

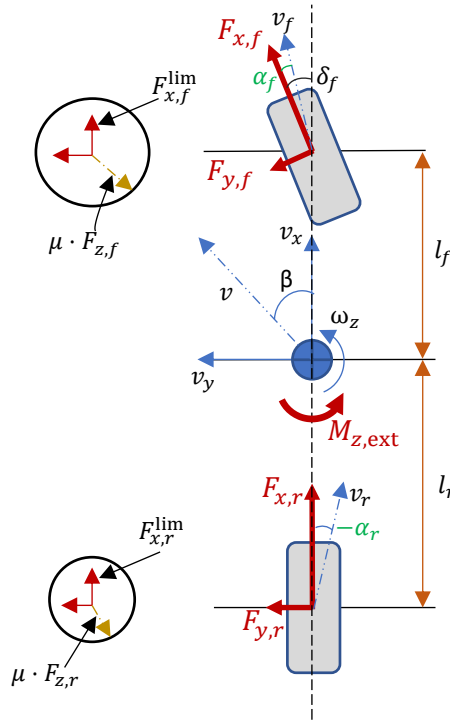


Figure 2.5: Free body diagram of the two-axle single track model with steered front axle. $M_{z,\text{ext}}$ is included here to represent the additional yaw moment generated by torque vectoring.

$$F_{y,r} = C_r \alpha_r \quad (2.30)$$

where C_f and C_r are the cornering stiffness coefficients of the front and rear axles, respectively. To ensure physically feasible tyre forces, the combined longitudinal and lateral forces are constrained by the friction circle

$$F_{x,f}^2 + F_{y,f}^2 \leq (\mu F_{z,f})^2 \quad (2.31)$$

$$F_{x,r}^2 + F_{y,r}^2 \leq (\mu F_{z,r})^2 \quad (2.32)$$

where μ is the road friction coefficient and $F_{z,f}$ and $F_{z,r}$ are the normal forces acting on the front and rear axles. These constraints represent the friction circle limits and introduce coupling between longitudinal and lateral tyre forces.

2.3.3 High fidelity simulation model

Due to the lack of access to a physical test vehicle with a distributed power-train, this thesis uses the Volvo Transport Models (VTM) library [31], developed

by Volvo Group Trucks Technology, as the primary plant model. VTM is implemented in MATLAB, Simulink, and Simscape Multibody, and is designed for analysis and control development of heavy-duty vehicles. It is used in Paper D, Paper E, and Chapter 5.

The VTM framework provides an industrially validated and scalable simulation environment for heavy-duty vehicle dynamics research. It is based on a modular multi-body formulation, where the vehicle is represented as interconnected rigid bodies such as chassis, axles, suspension, wheels, and trailers. This enables efficient reconfiguration of complex vehicle configurations, including articulated and distributed propulsion systems.

The model captures non-linear effects such as tyre–road interaction, load transfer, and dynamic coupling between tractor and trailer units. Compared to simplified models (e.g., single-track representations), it provides a more detailed description of coupled longitudinal, lateral, yaw, and roll dynamics, while remaining computationally efficient. The framework is primarily used for handling and stability analysis in critical scenarios such as lane changes, braking in curves, and low-friction conditions.

2.4 Electric powertrain

In this section the models used to represent the electric powertrain in the control system and as plant models are presented. Additionally, the efficiencies involved in electric power and mechanical power transmission are also defined.

2.4.1 Electric machines

Electric machines are modelled using simplified, control-oriented representations that capture their steady-state operational characteristics while omitting detailed electromagnetic, geometric, and construction-specific dynamics. This modelling approach ensures computational efficiency and compatibility with system-level simulations, optimisation, and real-time control frameworks.

2.4.1.1 Steady-state torque–speed representation

The electric machines are primarily represented using torque–speed maps, which discretise the operating region into a set of torque and angular speed points. These maps define key characteristics such as maximum motoring and regenerative torque limits as functions of speed, as well as efficiency and loss distributions across the operating range. The maps are implemented as lookup tables, enabling efficient interpolation of machine performance during simulation. In addition

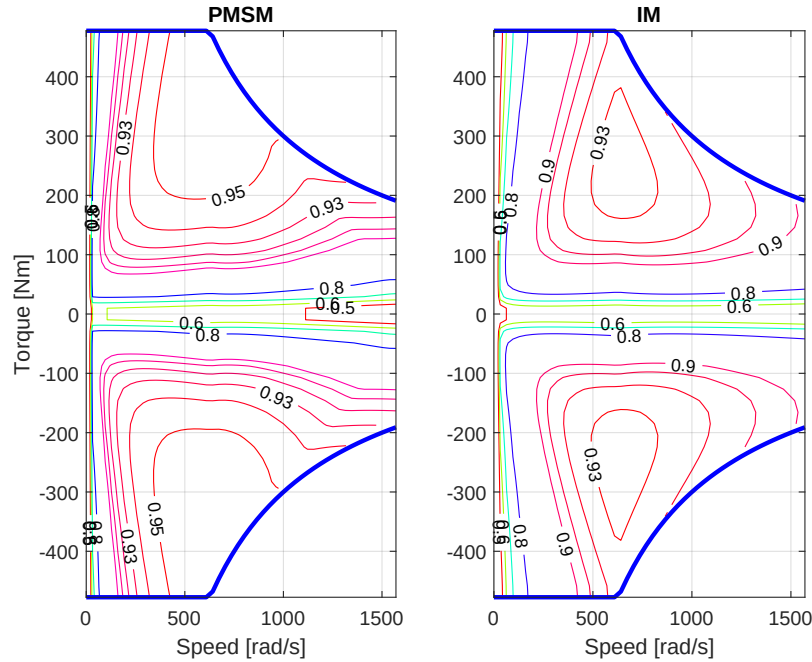


Figure 2.6: Efficiency contours, where motoring efficiency ($\eta_{EM,j}^{\text{mot}}$) is shown for positive torque and generating efficiency ($\eta_{EM,j}^{\text{gen}}$) for negative torque, together with the torque limits ($\pm T_{EM,j}^{\text{lim}}(\omega_{EM})$), represented by the thick blue lines, for 300 kW electric machines: (left) PMSM; (right) IM.

to performance characterisation, these maps are also used as plant models by combining steady-state relationships with simplified actuator dynamics.

The most common electric machine types in battery electric vehicles are induction machines (IMs) and permanent magnet synchronous machines (PMSMs), which exhibit different operational characteristics due to their construction and excitation principles [100].

2.4.1.2 Loss modelling

The total power losses in electric machines are represented through physically motivated components, including copper (winding resistive), iron (magnetic core), windage, and friction losses. In this work, loss characteristics are obtained from an electric machine model [49] and are observed to exhibit an approximately quadratic dependence on torque over the operating range.

A physically derived loss model is preferred over efficiency-based formulations, as it provides consistent representation across the entire operating region, includ-

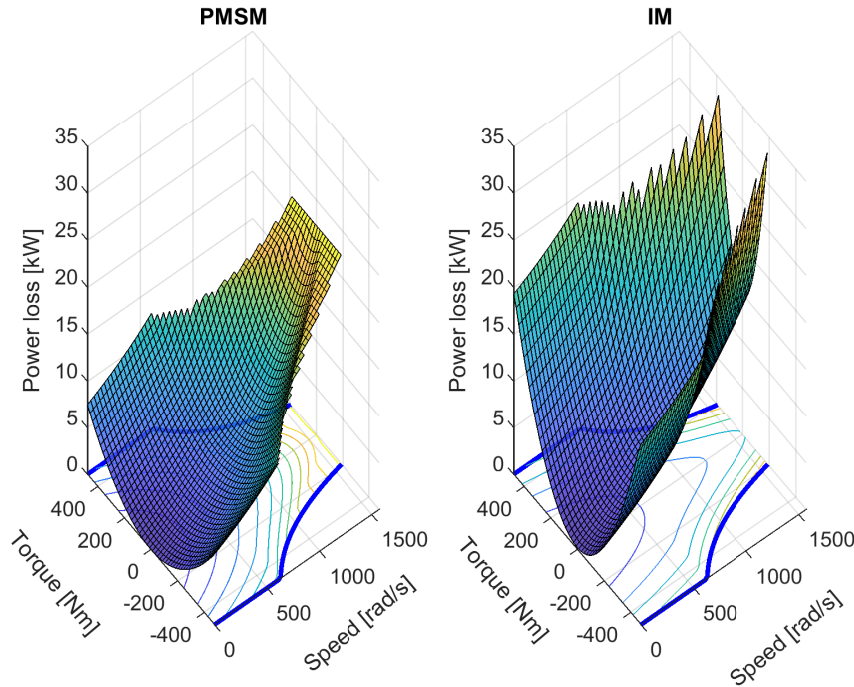


Figure 2.7: Power losses of the electric machines as functions of torque and angular speed: (left) PMSM; (right) IM. The non-zero power loss at zero torque represents the idle losses, which increase with angular speed.

ing the zero-torque condition. Efficiency-based models typically become ill-defined at zero torque and therefore require additional assumptions or corrections. In contrast, physically based models naturally capture idle losses, which are essential for accurate energy evaluation in system-level studies.

2.4.1.3 Idle and magnetisation losses

At zero or near zero torque, electric machines still incur losses, commonly referred to as idle or magnetisation losses. These losses correspond to the power required to establish and maintain the magnetic field within the machine, even in the absence of useful torque production.

In PMSMs, the rotor flux is provided by permanent magnets, and therefore no external magnetising current is required, resulting in relatively low idle losses. In contrast, induction machines require continuous reactive magnetising current to establish rotor flux, leading to significantly higher magnetisation losses, particularly at low load conditions. These losses increase with machine speed, as seen in 4.3, and are therefore important for accurately capturing overall drive-train efficiency in vehicle-level simulations. Additionally, to verify the principle

of idle losses, real vehicle measurements were performed and details are presented in Appendix A.

2.4.1.4 Actuator dynamics and torque limits

In addition to the steady-state torque–speed maps, the modelling framework is extended to include actuator dynamics in order to capture the finite torque response of the electric machines considered in Papers D, E and G. Rather than assuming instantaneous torque delivery, the actuator dynamics are represented by using a first-order lag model as:

$$\dot{T}_{EM,ij} = \frac{T_{EM,ij}^{\text{req}} - T_{EM,ij}}{\tau_{EM,ij}} \quad (2.33)$$

where $T_{EM,ij}^{\text{req}}$ is the requested electromagnetic torque, $T_{EM,ij}$ is the delivered electromagnetic torque, and $\tau_{EM,ij}$ is the torque response time constant of electric machine j on axle i .

To approximate the dynamically achievable torque without explicitly including actuator states in the controller, a first-order inverse actuator model is used:

$$T_{EM,ij}^{\text{dyn}}(t_s) = T_{EM,ij}^{\text{req}}(t_s) - \tau_{EM,ij} \frac{dT_{EM,ij}^{\text{req}}(t_s)}{dt} \quad (2.34)$$

where t_s denotes the sampling instant and $T_{EM,ij}^{\text{dyn}}$ represents the dynamically achievable torque. The delivered electric machine torque is then limited by the speed-dependent physical torque capability obtained from the torque–speed lookup tables as the following:

$$T_{EM,ij} = \max \left(T_{EM,ij}^{\min}, \min \left(T_{EM,ij}^{\max}, T_{EM,ij}^{\text{dyn}} \right) \right) \quad (2.35)$$

where the maximum and minimum torque capabilities of electric machine j on axle i are obtained from the torque–speed lookup table as given below:

$$T_{EM,ij}^{\max} = T_{EM,ij}^{\text{lim}}(\omega_{EM,ij}) \quad (2.36)$$

$$T_{EM,ij}^{\min} = -T_{EM,ij}^{\text{lim}}(\omega_{EM,ij}) \quad (2.37)$$

Alternatively, the actuator dynamics can be directly incorporated into the torque constraints. By discretising (2.33) and using a forward Euler approximation with controller sampling time T_s , the achievable torque variation within one sampling interval can be calculated as:

$$\Delta T_{EM,ij}^+ = \frac{T_s}{\tau_{EM,ij}} (T_{EM,ij}^{\max} - T_{EM,ij}^{\text{req}}) \quad (2.38)$$

$$\Delta T_{EM,ij}^- = \frac{T_s}{\tau_{EM,ij}} (T_{EM,ij}^{\min} - T_{EM,ij}^{\text{req}}) \quad (2.39)$$

The final limits combine the physical torque capability with the actuator dynamic response. Therefore, the dynamically feasible upper and lower torque limits are defined as:

$$T_{EM,ij}^{\text{max,dyn}} = \min (T_{EM,ij}^{\text{max}}, T_{EM,ij}^{\text{req}} + \Delta T_{EM,ij}^+) \quad (2.40)$$

$$T_{EM,ij}^{\text{min,dyn}} = \max (T_{EM,ij}^{\min}, T_{EM,ij}^{\text{req}} + \Delta T_{EM,ij}^-) \quad (2.41)$$

where $T_{EM,ij}^{\text{max,dyn}}$ and $T_{EM,ij}^{\text{min,dyn}}$ represent the dynamically achievable upper and lower torque limits used by the control allocator. These constraints ensure that the allocated torque commands satisfy both the speed-dependent physical torque capability and the finite torque response of the electric machines.

2.4.2 Transmissions

The purpose of a transmission system is to transfer the torque and mechanical power from the electric machine to the wheels. The transmission on an axle is represented by a fixed gear ratio selected according to the vehicle configuration and performance requirements. The ratio includes both the main transmission and differential reduction and determines the operating region of the electric machine, thereby influencing the overall powertrain efficiency.

Transmission losses depend primarily on rotational speed, transmitted load and temperature, and can be divided into load-independent and load-dependent components [30, 50]. Load-independent losses include, for example, bearing and windage losses, whereas load-dependent losses originate from rolling and sliding friction within the transmission. These losses typically result in transmission efficiencies ranging from 93-97%.

Although transmission losses can be described using detailed physical models, they are represented using a constant efficiency in this thesis. This representation is only used for power and torque transfer in the actuator model and is not included in the control problem formulations. The efficiency of the single-speed transmission associated with driven axle i is denoted by $\eta_{trmn,i}$ and is approximated as a function of the gear ratio. The transmission efficiency is given by

$$\eta_{trmn,i} = 0.99^{\frac{gr_i}{3}} \quad (2.42)$$

where gr_i represents the gear ratio of a driven axle i .

For vehicle configurations employing a single electric machine per driven axle, the axle torque is distributed equally between the left and right wheels, implicitly representing an open differential. A detailed differential model is omitted since it is not required by the considered use cases, including select-low behaviour.

2.4.3 Inverters and batteries

During motoring operation, the electric powertrain converts electrical power into mechanical power, whereas during regenerative operation, mechanical power is converted back into electrical power. These energy conversion processes introduce losses in the inverter and battery. Battery losses originate primarily from internal resistance, which dissipates stored energy as heat. Inverter losses arise from DC–AC and AC–DC conversion due to semiconductor switching and electrical conduction losses.

In this thesis, the inverter and battery losses are represented using a lumped conversion efficiency. The battery power consumption P_{Batt}^{cons} and regenerative power P_{Batt}^{gen} are calculated based on the mechanical power of the electric machines at each driven axle. The formulation includes the lumped electrical conversion efficiencies and additional electric machine losses.

The parameters η_{el}^{mot} and η_{el}^{gen} represent the motoring and regenerative lumped conversion efficiencies, respectively. The variable $P_{EM,ij}$ denotes the mechanical power of electric machine j associated with axle i , while $P_{EM,ij}^{loss}$ represents additional electric machine losses not captured by the lumped efficiency terms.

The battery power consumption in motoring operation is given by:

$$P_{Batt}^{cons} = \sum_i \sum_j \left(\frac{P_{EM,ij}}{\eta_{el}^{mot}} + P_{EM,ij}^{loss} \right) \quad (2.43)$$

Similarly, the battery power regeneration during generating operation is expressed as:

$$P_{Batt}^{gen} = \sum_i \sum_j (\eta_{el}^{gen} P_{EM,ij} - P_{EM,ij}^{loss}) \quad (2.44)$$

2.5 Brake system

The brake system is modelled using a simplified, control-oriented representation that captures its functional behaviour without detailed thermo-mechanical or pneumatic dynamics. This approach provides sufficient accuracy for system-level simulations and control design while maintaining computational efficiency. The brake system is represented by a static mapping between the brake request and the generated wheel braking torque. The achievable braking torque is limited by actuator capability constraints, enabling coordination between friction braking and regenerative braking. Since the considered applications are dominated by steady-state brake characteristics, a static representation is sufficient for the nominal brake behaviour.

In Papers D, E, and G, the brake model is extended to include actuator dynamics to capture the finite response time of the friction brakes. The brake actuator dynamics are represented using a first-order lag model:

$$\dot{T}_{Brk,ij} = \frac{T_{Brk,ij}^{\text{req}} - T_{Brk,ij}}{\tau_{Brk,ij}} \quad (2.45)$$

where $T_{Brk,ij}^{\text{req}}$ and $T_{Brk,ij}$ denote the requested and delivered brake torque, respectively, and $\tau_{Brk,ij}$ is the brake actuator time constant.

Similar to the electric machine actuator model, the dynamically achievable brake torque is estimated using a first-order inverse actuator approximation:

$$T_{Brk,ij}^{\text{dyn}}(t_s) = T_{Brk,ij}^{\text{req}}(t_s) - \tau_{Brk,ij} \frac{dT_{Brk,ij}^{\text{req}}(t_s)}{dt} \quad (2.46)$$

The delivered brake torque is obtained by limiting the dynamically achievable torque according to the physical brake capability:

$$T_{Brk,ij} = \max \left(T_{Brk,ij}^{\min}, \min \left(T_{Brk,ij}^{\max}, T_{Brk,ij}^{\text{dyn}} \right) \right) \quad (2.47)$$

where a negative torque convention is adopted for braking such that $T_{Brk,ij}^{\min} < 0$ and $T_{Brk,ij}^{\max} = 0$ denote the minimum and maximum physically achievable brake torques, respectively.

Similar to the electric machine actuator model, the actuator dynamics can also be incorporated into the torque constraints. Using a forward Euler discretisation of the first-order actuator model in (2.45) with sampling time T_s , the dynamically achievable brake torque limits are defined as:

$$T_{Brk,ij}^{\text{max,dyn}} = \min \left(T_{Brk,ij}^{\max}, T_{Brk,ij}^{\text{req}} + \Delta T_{Brk,ij}^+ \right) \quad (2.48)$$

$$T_{Brk,ij}^{\text{min,dyn}} = \max \left(T_{Brk,ij}^{\min}, T_{Brk,ij}^{\text{req}} + \Delta T_{Brk,ij}^- \right) \quad (2.49)$$

where $\Delta T_{Brk,ij}^+$ and $\Delta T_{Brk,ij}^-$ are the achievable torque variations obtained from the first-order actuator model. These dynamically feasible limits are used by the control allocator to ensure that the allocated braking torque satisfies both the physical brake capability and the actuator response limitations.

Friction brake usage is penalised in the control algorithm since the braking energy dissipated by friction brakes is entirely converted into heat. The dissipated brake power is calculated according to:

$$P_{Brk,ij}^{\text{loss}} = -T_{Brk,ij}^{\text{req}} \cdot \omega_{w,il} \quad (2.50)$$

where $\omega_{whl,il}$ is the wheel speed. Other losses associated with brake actuation, such as air compressor operation and air leakage, are neglected due to their relatively small contribution.

2.5.1 Anti-lock braking system

An Anti-lock Braking System (ABS) is incorporated as a safety mechanism to prevent wheel lock under conditions where the requested braking torque exceeds the available tyre-road friction capability. Wheel lock can reduce lateral stability and steerability; therefore, the ABS monitors wheel slip and modulates the applied friction brake torque when predefined slip limits are exceeded.

When ABS is activated, the electric machines on the affected axle are assigned zero torque request to avoid additional braking torque from regenerative braking. Although a combined regenerative and friction braking strategy is possible, this simplified implementation is selected to reduce complexity.

The ABS controller is formulated based on the longitudinal braking slip ratio of a tyre:

$$\kappa_{il} = \frac{v_x - \omega_{w,il} \cdot r_{w,i}}{\max(v_x, \epsilon)} \quad (2.51)$$

where v_x is the longitudinal vehicle velocity, $\omega_{w,il}$ is the wheel angular velocity on side l of axle i , $r_{w,i}$ is the effective wheel radius of axle i , and ϵ is a small positive constant used to avoid division by zero at low speeds. The braking torque is constrained according to the slip limit κ_i^{lim} . The friction brake torque is adjusted according to the following logic

$$T_{Brk,ij} = \begin{cases} 0, & \kappa_{il} > \kappa_i^{\text{lim}} \\ T_{Brk,ij}^{\text{req}} & \end{cases} \quad (2.52)$$

This simplified ABS formulation captures the essential wheel-lock prevention behaviour required for vehicle-level simulations.

Chapter 3

Electric vehicle powertrain architectures

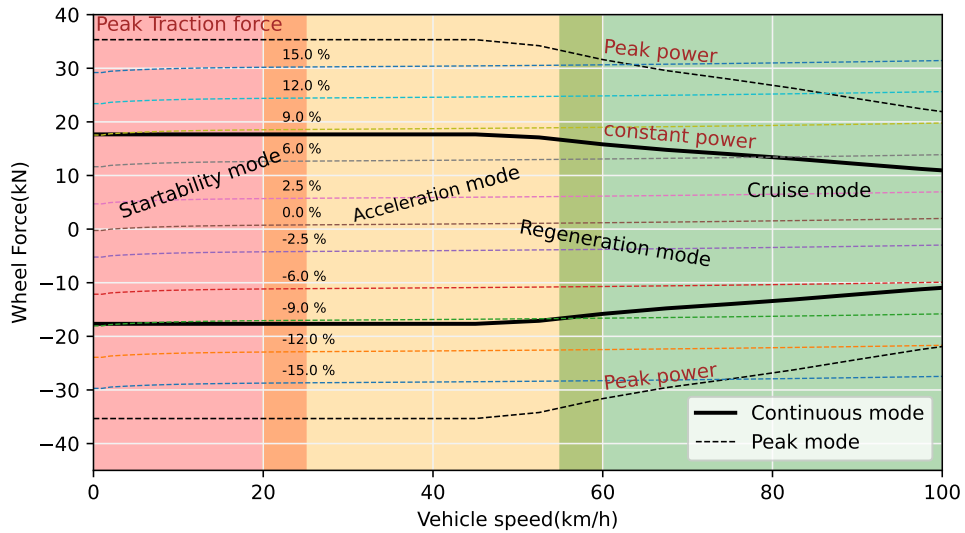


Figure 3.1: A simplified representation a wheel-force versus vehicle-speed diagram, commonly referred to as a traction diagram for 40 tonne heavy electric truck with 600 kW (continuous power) electric powertrain. The various operating modes are highlighted on the traction diagram. The thick black lines represent the maximum wheel force available in continuous mode and the dashed black lines represents the peak mode at which high wheel force is available for short durations.

The longitudinal motion performance requirements of heavy commercial vehicles are commonly defined through operating conditions such as acceleration, cruising, braking, startability, and gradeability. These operating conditions determine the tractive force and power that must be delivered by the powertrain

over the vehicle speed range. A convenient and widely used tool for representing these requirements is the traction diagram shown in Figure 3.1, which relates the available tractive effort to vehicle speed together with the total road-load resistance. This representation identifies the principal vehicle operating modes, including traction-limited launch, acceleration, constant-speed cruising, gradeability, and braking, each imposing distinct requirements on the powertrain in terms of torque capability, power rating, controllability, and energy efficiency.

From a powertrain design perspective, the traction diagram provides a systematic basis for dimensioning the propulsion system. The low-speed region, dominated by startability and traction-limited operation, primarily determines the maximum torque requirement and therefore influences motor sizing, transmission ratio selection, and the number of driven axles. The intermediate speed range, where acceleration and gradeability requirements overlap, governs the continuous torque capability and thermal loading of the electric machines. At higher vehicle speeds, aerodynamic drag becomes the dominant resistance, and the intersection between the tractive effort and road-load curves determines the continuous power requirement. Similarly, braking and downhill operation define the required regenerative braking capability, which influences the sizing of the electric machines, power electronics, and energy management strategy.

Different electric powertrain architectures provide varying capabilities in satisfying these operating requirements. Architectures based on a central electric machine with transmission, distributed e-axes, or in-wheel motors differ in their torque distribution capability, redundancy, efficiency characteristics, packaging, and scalability. For example, distributed drive architectures improve traction utilisation, startability, and gradeability by distributing driving torque among multiple driven axles, while centralised architectures typically offer lower system complexity and cost. Likewise, multi-motor configurations facilitate regenerative braking and enable wheel torque vectoring, improving vehicle controllability during low-friction and highly dynamic manoeuvres.

This chapter introduces the principal electric powertrain architectures for heavy electric vehicles and evaluates their characteristics using the operating regions identified in the traction diagram as a common assessment framework. The resulting torque and power requirements are used to compare the architectures in terms of propulsion system sizing, drivetrain layout, number of driven axles, transmission requirements, and overall vehicle performance. This provides a consistent basis for assessing the suitability of different electric powertrain configurations for heavy commercial vehicle applications.

3.1 Envisioned functionality architecture for motion control

To support different vehicle configurations and electric powertrain architectures, this thesis adopts a hierarchical functionality reference architecture for vehicle motion control based on the conceptual frameworks proposed in [83, 102]. Hierarchical motion control architectures provide a modular structure that separates decision making according to temporal horizon and functional abstraction, thereby facilitating adaptation to different vehicle and actuator configurations while reducing development complexity [72, 102, 61]. Higher layers typically operate over longer prediction horizons and lower update frequencies, whereas lower layers execute high-frequency control actions. Figure 3.2 illustrates the proposed functional architecture. Dashed blocks denote functionalities that are outside the scope of this thesis. The operation of each layer is described below, proceeding from the highest abstraction level towards the actuator level.

Higher layer functions

The Higher layer functions consolidate the transport mission management, route management, and route situation management functionalities described in [83]. These functions are included for completeness and interact with the lower layers through motion requests together with aggregated actuator capabilities and status information. Typical examples include route selection, predictive energy management strategies (e.g., Volvo I-See), and manoeuvre planning such as highway merging or exit planning.

Traffic situation management

Traffic situation management performs tactical driving functions over an intermediate prediction horizon by considering the surrounding traffic environment, obstacles, and road geometry. Representative functions include adaptive cruise control, automatic emergency braking, highway pilot, and lane-change assistance. These functions generate requests for longitudinal and lateral vehicle motion, represented in Figure 3.2, these requests are denoted by Δa_x^{req} and $\Delta \omega_z^{\text{req}}$, representing the requested changes in longitudinal acceleration and vehicle yaw rate, respectively.

Vehicle motion management

Vehicle motion management translates the requests received from the higher layers into generalised vehicle force requests while considering vehicle states, actuator

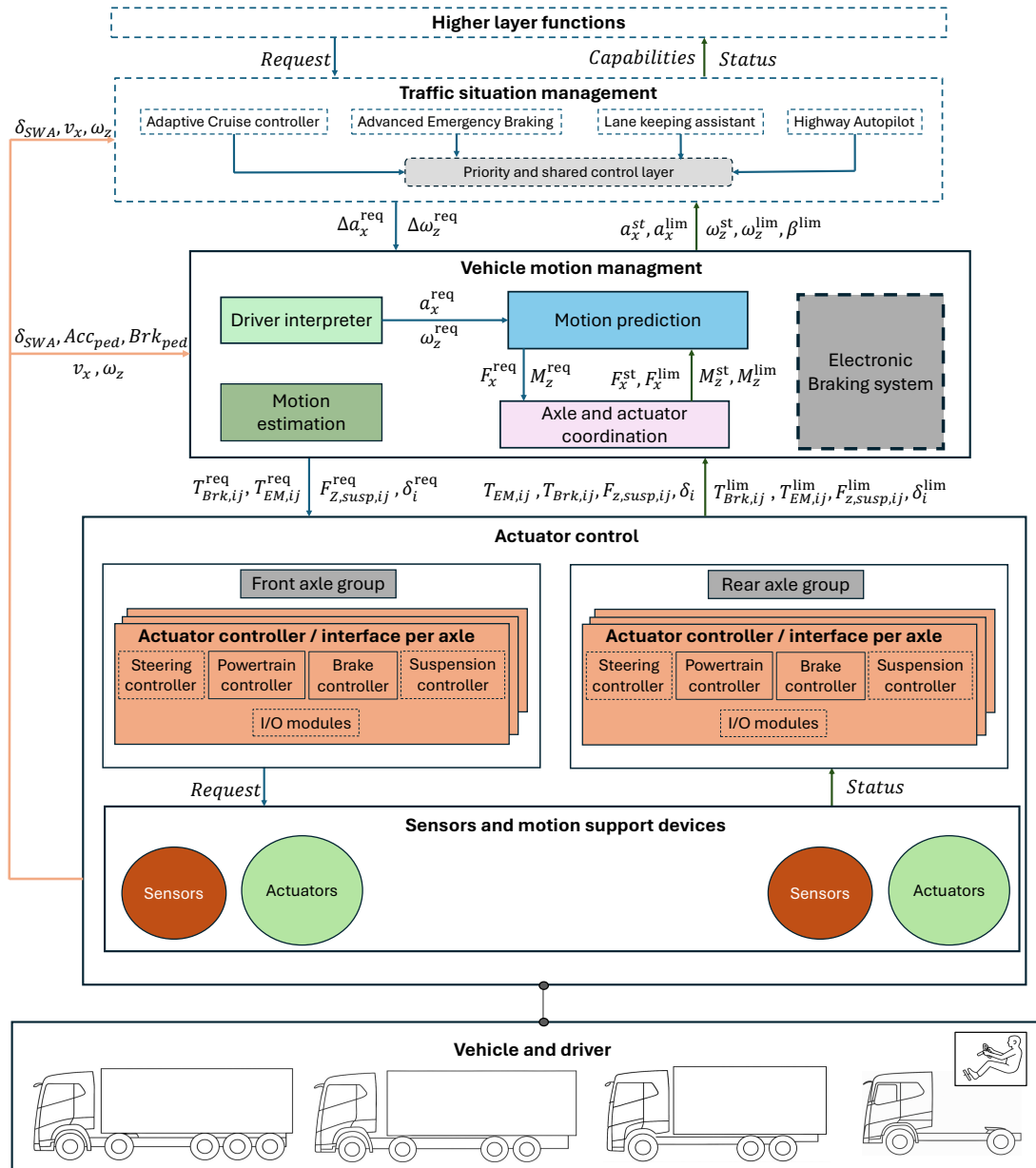


Figure 3.2: Generic layers of envisioned functionality reference architecture for motion control. The scope of functionalities within each layer may vary depending on the vehicle configuration and driver train configuration. I/O modules represent the input/output modules which act as gateway between the controllers and the actuator devices.

capabilities, and operating constraints. This layer forms the primary focus of this thesis and consists of four main functionalities: driver interpretation, motion estimation, motion prediction, and actuator coordination. The electronic braking system (EBS) is assumed to be integrated within this layer, where braking requests are coordinated together with the remaining motion actuators.

- **Driver interpreter:** The driver interpreter converts steering wheel angle and pedal inputs into driver requests, primarily the desired longitudinal acceleration, a_x^{req} , and desired yaw rate, ω_z^{req} . Additional driver intent quantities may also be generated to improve vehicle performance [102, 89], although these are outside the scope of this thesis.
- **Motion estimation** Motion estimation provides the vehicle states required by the motion controller, including longitudinal velocity, yaw rate, body side-slip angle, tyre forces, and normal loads. In practical implementations, this functionality is essential for ensuring robust operation under varying vehicle loading conditions, road profiles, and sensor availability. Since the studies presented in this thesis are simulation-based, the required vehicle states are obtained directly from the high-fidelity vehicle model rather than estimated. Consequently, state estimation algorithms are not further investigated.
- **Motion prediction** The motion prediction functionality combines the requests received from the driver interpreter and traffic situation management to generate generalised vehicle force requests consisting of the desired longitudinal force, F_x^{req} , and yaw moment, M_z^{req} . These requests are represented by the virtual control vector

$$\mathbf{v} = \begin{bmatrix} F_x^{\text{req}} & M_z^{\text{req}} \end{bmatrix}^T$$

The generated requests are continuously adjusted using feedback from the actuator coordinator to ensure that vehicle stability limits and actuator constraints are respected.

- **Axle and actuator coordination** The actuator coordinator transforms the generalised vehicle force request, $\mathbf{v}$, into actuator-level requests required by the available steering, propulsion, braking, and suspension systems. This functionality coordinates the use of redundant actuators while respecting actuator operating limits and the available tyre-road friction. Consequently, constraints arising from tyre friction cannot be handled independently by the local actuator controllers but require coordination at the vehicle level.

Actuator control

The actuator control layer consists of local controllers associated with the individual actuators or axle groups. These controllers receive actuator requests from the actuator coordinator, execute the corresponding low-level control functions, and report actuator status together with their available operating capabilities. Examples include electric machine torque controllers, friction brake controllers, steering controllers, and suspension controllers. The local controllers thereby generate the wheel forces and moments required to realise the requested vehicle motion.

The layered architecture separates vehicle-level motion control from actuator-specific implementation, allowing different vehicle configurations and powertrain architectures to be accommodated without modifying the higher-level motion control functions. For example, an axle group may consist of one or several physical axles, employ a shared differential, or use individually controlled electric machines distributed across different axles. This modularity enables the same motion control framework to be applied across different heavy vehicle platforms while requiring only limited modifications to the lower control layers.

3.2 Classification of electric powertrain architectures for HCVs

Electric machines provide several advantages for heavy vehicle traction compared with conventional diesel engines, including high torque capability at low speed, regenerative braking capability, high efficiency, and a broad operating range. Furthermore, electric machines generally achieve higher specific power than diesel engines. Reported specific power values for diesel engines are typically in the range of 0.1–0.6 kW/kg, whereas electric machines commonly achieve approximately 2–5 kW/kg, corresponding to an improvement of around 5–10 times in specific power [13, 7].

The higher power density and modularity of electric machines provide additional design freedom for heavy vehicle powertrains. Unlike conventional powertrains, where a single centralised engine typically defines the powertrain layout, electric propulsion allows the traction power to be distributed among multiple smaller machines. This enables increased scalability, improved packaging flexibility, and new powertrain configurations through different combinations of motors, transmissions, and driven axles [14, 21].

Due to their wide speed and torque operating range, electric machines can often be coupled to the powertrain using single-stage reduction gears or, in some cases, directly connected to the wheels. The reduced requirement for multi-speed transmissions and clutches further simplifies the powertrain layout. Consequently,

electric powertrains enable a wider range of architectures compared with conventional diesel-based vehicles, which can generally be classified into centralised and distributed propulsion configurations [21, 108].

3.2.1 Centralised powertrain architecture (CEP)

Due to the strong design heritage of heavy vehicles based on diesel powertrains, many battery electric vehicles (BEVs) adopt layouts that retain a centralised propulsion unit. In this thesis, such configurations are classified as centralised electric powertrain (CEP) architectures. In a CEP, one or more electric machines are mounted on the vehicle frame. Their torque is combined through a common transmission and subsequently distributed to the driven wheels through propeller and drive shafts as shown in Figure 3.3. The number of machines and the transmission configuration depends on the vehicle specification, required torque capability, power rating and operating speed range of the electric machines.

The generated torque is transmitted to the driven axles through conventional drivetrain components, including propeller shafts, differentials, and axle gear sets [21, 81]. For heavy-duty applications, multiple electric machines may be connected in parallel to achieve the required propulsion power while retaining a similar layout to conventional powertrains. Since the propulsion torque is generated upstream of the transmission, the available traction and regenerative braking forces are typically restricted to the driven axle group. A representative CEP configura-

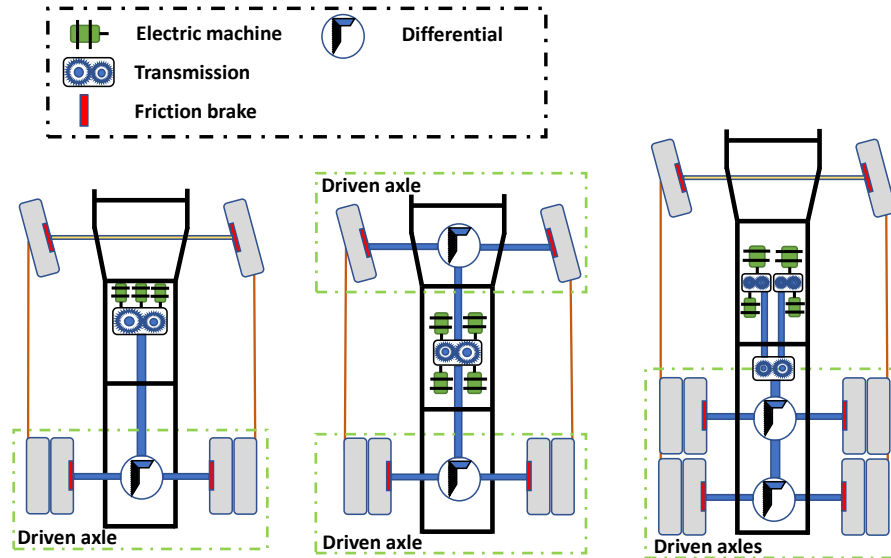


Figure 3.3: Examples of centralised powertrain architecture on different vehicle configurations; (left): 4x2; (Center): 4x4 (right): 6x4.

tion is a long-haul heavy-duty truck where a centrally mounted electric machine drives one or more rear axles through a driveshaft. The main advantages of this architecture are compatibility with existing chassis designs, manufacturing processes, and service infrastructure. However, retaining conventional drivetrain components introduces additional mechanical losses and packaging constraints. Components such as propeller shafts, differentials, and axle gears contribute both load-dependent and load-independent losses, while occupying space that could otherwise be allocated to batteries or power electronics [84, 21]. requirements.

Furthermore, because traction and regenerative braking are concentrated on the driven axle group, friction brakes remain necessary on non-driven wheels to satisfy braking performance requirements. Additional components such as mechanical disconnect clutches may be introduced to reduce electric machine idle losses, particularly for permanent magnet synchronous machines, but they increase system complexity, cost, and packaging requirements.

3.2.2 Distributed powertrain architecture (DEP)

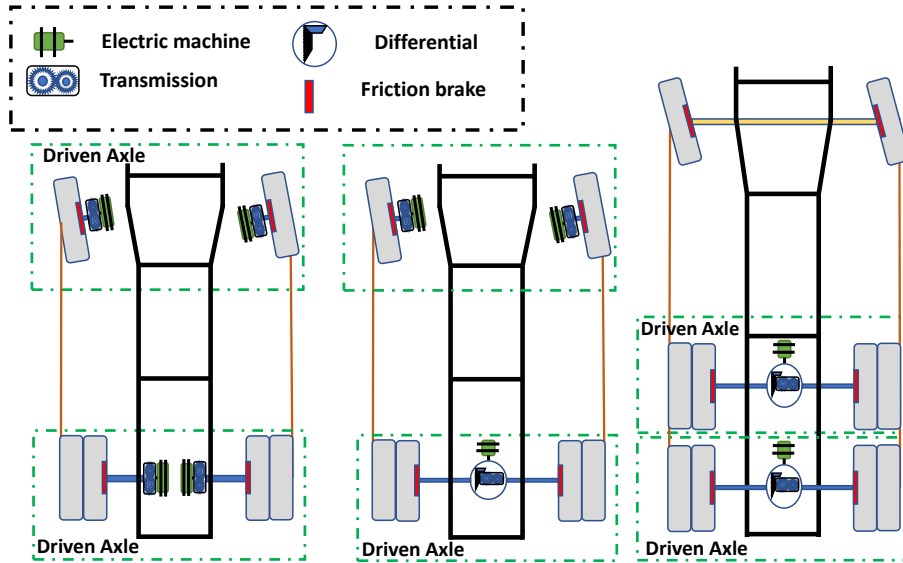


Figure 3.4: Examples of distributed powertrain architecture on different vehicle configurations; (left): 4x4 (IWM); (Center): 4x4 (IWM of front axle and e-axle on rear axle) (right): 6x4 (e-axles on rear axle group)

In contrast to CEP configurations, distributed electric powertrain architectures locate propulsion units closer to the driven axles or individual wheels. In this thesis, a DEP configuration refers to a vehicle architecture where the driven axles are propelled by independent electric power sources rather through a common mechanical driveline. Thus, the propulsion power is not mechanically combined into

a common shaft before being distributed to the driven axles. This eliminates or reduces the need for conventional components such as propeller shafts and central differentials or distribution differentials, enabling modular drivetrain layouts with independent torque control of the driven axles or wheels [114, 13].

Unlike CEP configurations, where multiple machines may be mechanically coupled through a common drivetrain, the propulsion units in a DEP are mechanically decoupled. For example, a 4x4 vehicle may employ independently driven front and rear axles, whereas a 6x4 heavy vehicle may use independently driven rear axles with a non-driven steered front axle. Each driven axle can be equipped with one or multiple electric machines connected through a common or individual reduction gear. Examples of some DEP configurations are shown in Figure 3.4.

The mechanical decoupling of propulsion units provides additional degrees of freedom for vehicle motion control. Independent torque distribution enables improved traction utilisation, propulsion redundancy, and fault tolerance. Furthermore, the ability to generate different torque levels at individual axles or wheels enables advanced control strategies, including torque vectoring and enhanced vehicle stability control [114, 3]. However, these advantages are achieved at the cost of increased system complexity, higher installed power requirements, additional control challenges, and potentially increased manufacturing cost.

A common implementation of DEP is the electric axle (e-axle), where the electric machine, power electronics, and reduction gear are integrated into a compact axle-level propulsion unit. Compared with CEP architectures, e-axes shorten the mechanical power transmission path while maintaining an axle-based layout similar to conventional heavy vehicles. Multiple e-axes can be combined to electrify several driven axles, enabling modular powertrain configurations for heavy-duty trucks and buses [115, 10].

A higher level of distribution is achieved using individual wheel motor (IWM) architectures, where each driven wheel is equipped with an independent electric machine. By eliminating mechanical differentials and further reducing drivetrain components, IWMs provide the highest degree of actuator redundancy and wheel-level torque controllability. This enables advanced traction control, torque vectoring, and regenerative braking strategies [78, 18]. However, the increased integration at the wheel level introduces additional challenges, including higher system complexity, thermal management requirements, durability constraints, and increased unsprung mass for in-wheel implementations [12, 66]. Therefore, IWM architectures represent the most flexible but also the most challenging form of distributed electric propulsion.

3.3 Studied DEP architecture using modular e-axes

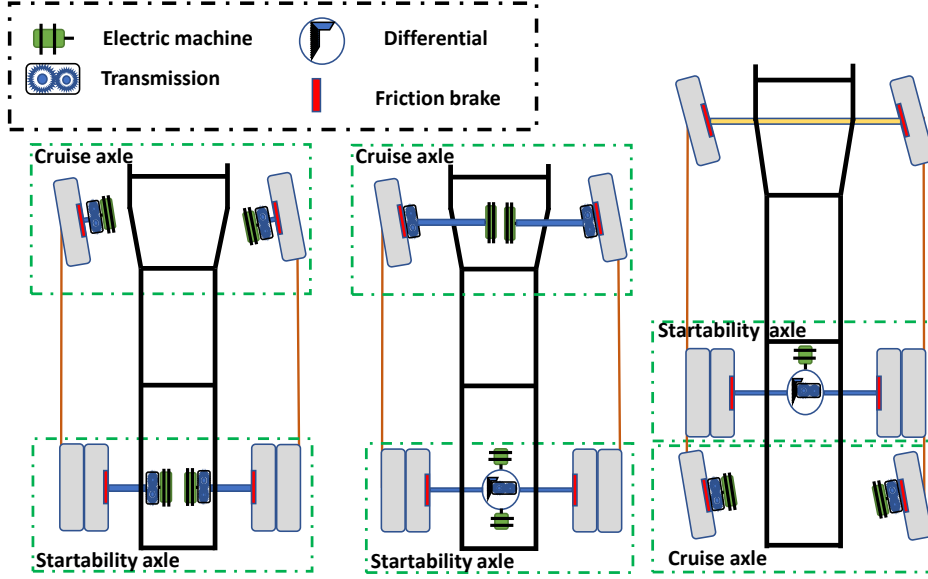


Figure 3.5: Examples of arrangement of cruise and startability axes; (left): 4x4 (IWMs on all axes); (Center): 4x4 (IWM of front axle and e-axle on rear axle); (right): 6x4 (e-axes on first rear axle and IWM on the second rear axle.)

The powertrain architecture investigated in this thesis is based on a distributed electric powertrain (DEP) concept employing modular e-axes for heavy commercial vehicles. Unlike a centralised powertrain, the propulsion capability is distributed among independently controlled axes, enabling each axle group to be designed according to its intended operating function. The proposed architecture consists of two functionally differentiated propulsion units: a cruise axle and a startability axle.

The cruise axle is optimised for continuous operation during high-speed driving, whereas the startability axle provides additional tractive capability during low-speed operation and high gradeability requirements. By assigning different operating functions to the axle groups, the architecture enables efficient utilisation of the electric machines across different vehicle operating conditions while avoiding excessive sizing of a single propulsion unit. The term modular refers to the possibility of adapting the number and placement of e-axes according to the vehicle configuration, such as different axle layouts for rigid trucks and tractor configurations.

Representative implementations for 4×4 and 6×4 heavy vehicles are illustrated

in Figure 3.5. In this study, each driven axle is equipped with one electric machine and a single-stage reduction gear. Although the proposed architecture is evaluated using this configuration, the concept can be extended to alternative implementations involving multiple machines per axle or different transmission configurations. The functional separation between propulsion units is inspired by multi-motor electric vehicle concepts, where different machines can be optimised for different operating conditions [62].

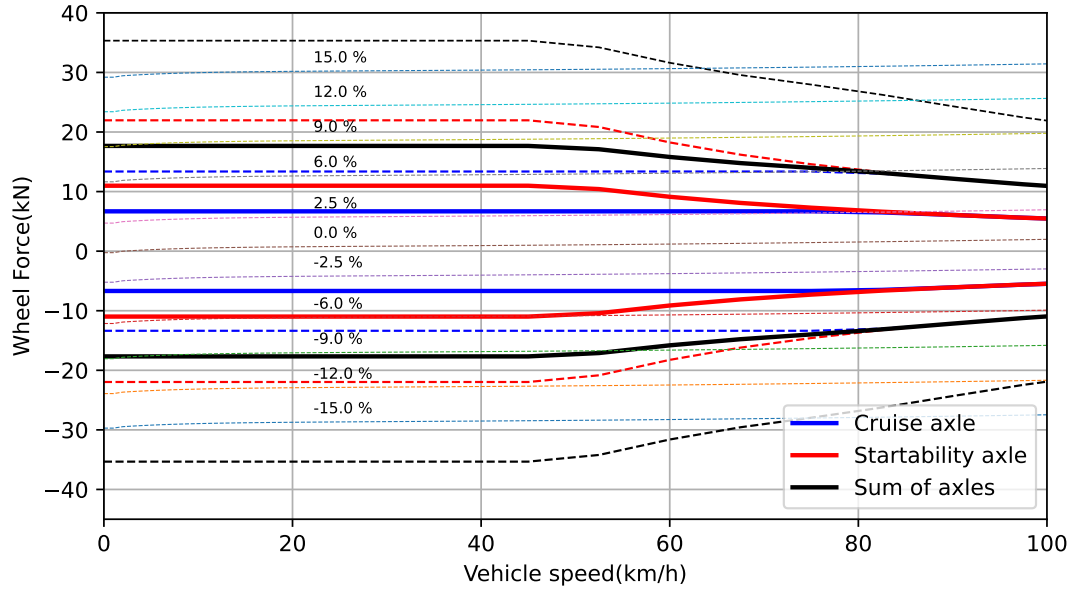


Figure 3.6: Traction diagram for a 40 tonne heavy electric vehicle using electric machines with 300 kW maximum continuous power on the cruise and startability axle. The gear ratio on the cruise axle is chosen to be 14 and that on the startability axle as 23. The electric machines are configured with a maximum rotating speed of 15000 rpm. The solid and dashed lines represents the maximum continuous and peak wheel forces available on each axle (red and blue) and sum of the axles (black).

The combination of the cruise and startability axles enables multiple propulsion modes. A typical traction diagram for the studied DEP architecture is shown in Figure 3.6. By combining the continuous and peak operating regions of the individual axles, the available tractive effort can be extended over a wider vehicle speed range. This provides a similar functional benefit to selecting different gear ratios in a conventional powertrain, while retaining the advantages of electric propulsion. However, peak operation is limited by thermal constraints and is therefore available only for a limited duration.

For powertrain dimensioning, the following operating modes are defined:

- **Cruise mode:** Continuous operation at vehicle speeds of 80–85 km/h on

road grades up to approximately $\pm 2.5\%$, representing typical long-haul highway operation.

- **Startability mode:** Vehicle take-off and low-speed operation on steep grades, typically around $\pm 8 - 10\%$, where high wheel torque capability is required.
- **Power mode:** Combined operation of both axle groups during transient high-demand conditions, such as steep grades exceeding the cruise capability or high-speed acceleration events. In this mode, the startability axle assists the cruise axle, while during severe low-speed conditions the cruise axle can provide additional traction support.

The distributed architecture also provides flexibility for braking and vehicle motion control. Electric machines on individual axles or wheels can be coordinated with friction brakes to enable regenerative braking optimisation and brake blending. Furthermore, independently controlled wheel torques enable torque-vectoring functionality, which is exploited in the motion control strategies developed in later chapters.

The cruise axle is configured using a permanent magnet synchronous machine (PMSM), selected due to its high efficiency and torque density, which are advantageous for continuous long-haul operation. The startability axle employs an induction machine (IM), selected for its robustness and ability to reduce idle losses when inactive through de-energisation [88]. This provides an alternative to mechanical disconnect mechanisms while maintaining the capability to deliver high torque when required. The combination of PMSM and IM technologies also provides a balance between efficiency, performance, and system cost, which is critical for commercial operations of heavy vehicles.

The electric machine ratings and gear ratios are selected to satisfy the required vehicle performance over the complete operating range. Other machine technologies, including electrically excited synchronous machines (EESMs), synchronous reluctance machines (SynRMs), and switched reluctance machines (SRMs), may also be considered depending on application requirements and cost constraints.

3.3.1 Optimising cruise and startability axle configurations to maximise driving range

Maximising the driving range of a heavy electric vehicle requires both an efficient powertrain configuration and coordinated control of the cruise and startability e-axes. Following the methodology proposed in Paper A, this problem is formulated as a joint hardware and control optimisation, in which the electric machine ratings, transmission gear ratios, and longitudinal force allocation between the

cruise and startability axes are determined simultaneously to minimise the total battery energy consumption over representative long-haul operating conditions. The distributed electric powertrain enables operating-mode-based propulsion, allowing each axle to be utilised according to its efficiency characteristics and the instantaneous traction demand.

The methodology combines a quasi-static longitudinal vehicle model, electric machine efficiency maps, and an optimal force coordination strategy. The vehicle model is used to determine the required wheel force and wheel speed for representative long-haul operating conditions, including steady-state cruising, gradeability, and transient acceleration events. The efficiency maps are then used to evaluate the corresponding electrical power consumption.

The joint optimisation problem is formulated by defining the objective function as the net electrical energy drawn from the battery over the drive cycle:

$$\min_{\Phi, F_{x,crs}(\cdot), F_{x,stb}(\cdot)} J = \int_0^{t_f} (P_{Batt}^{cons}(t) - P_{Batt}^{gen}(t)) dt, \quad (3.1)$$

where P_{Batt}^{cons} and P_{Batt}^{gen} denote the battery power consumption during motoring and the regenerative power during braking, respectively, and Φ denotes the set of design variables comprising the electric machine ratings and transmission gear ratios. The optimisation simultaneously determines the longitudinal force contributions of the cruise axle, $F_{x,crs}$ and the startability axle, $F_{x,stb}$, subject to the total traction demand,

$$F_x^{req} = F_{x,crs} + F_{x,stb} \quad (3.2)$$

and the axle force limits:

$$0 \leq F_{x,crs} \leq F_{x,crs}^{max}(\omega_{w,crs}; \Phi), \quad (3.3)$$

$$0 \leq F_{x,stb} \leq F_{x,stb}^{max}(\omega_{w,stb}; \Phi), \quad (3.4)$$

where the maximum axle forces are determined from the electric machine ratings and transmission gear ratios. Additional constraints enforce the longitudinal vehicle dynamics together with the specified maximum speed and gradeability requirements. The force contributions $F_{x,crs}$ and $F_{x,stb}$ are determined at each operating point using a control allocation strategy based on a weighted least-squares formulation, also referred to as global force error minimisation. The problem formulation of the control allocation problem is presented in Section (4.1.2.1). The resulting force allocation minimises unnecessary actuator utilisation while satisfying the requested traction demand. Consequently, the cruise axle is primarily utilised for propulsion whenever it can satisfy the traction demand, whereas the startability axle contributes only when the required force exceeds the capability of the cruise axle.

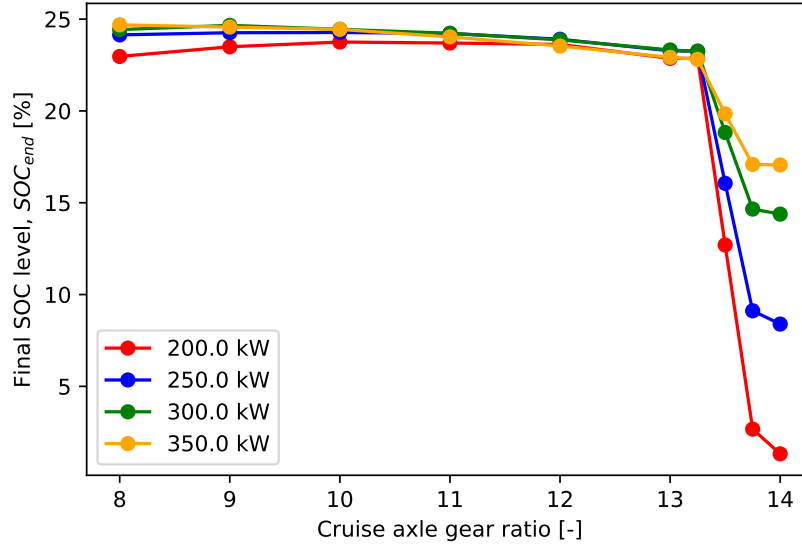


Figure 3.7: Battery state of charge level at the end of route for different cruise axle gear ratios and cruise axle machine powers using a constant startability axle gear ratio of 23.

Results

The results presented in Paper A demonstrate the influence of the cruise axle gear ratio and electric machine rating on the achievable driving range. Fig. 3.7 shows the final battery state of charge, (SOC_{end}), for different hardware configurations. For completeness, several configurations that do not satisfy the specified cruise-mode and startability requirements are also included. The results indicate that configuring both axles with electric machine ratings of approximately 300 kW and selecting a cruise axle gear ratio between 12 and 13 maximises the driving range for the considered operating conditions. The wheel force–vehicle speed characteristics of the selected powertrain configuration are approximately similar to those shown in Figure 3.6.

In addition to the results presented in Paper A, additional simulations were performed in this thesis to further evaluate the joint hardware and control design framework. Figure 3.8 shows the results for a configuration using identical PMSMs and equal transmission gear ratios on both axles. The same use case as in Paper A is considered. The control strategy is modified to distribute the traction demand equally between the two electric machines, resulting in simultaneous operation of both axles throughout the drive cycle.

Since the same single-wheel quasi-static vehicle model is used to represent the longitudinal dynamics, the vehicle motion remains identical to the reference case,

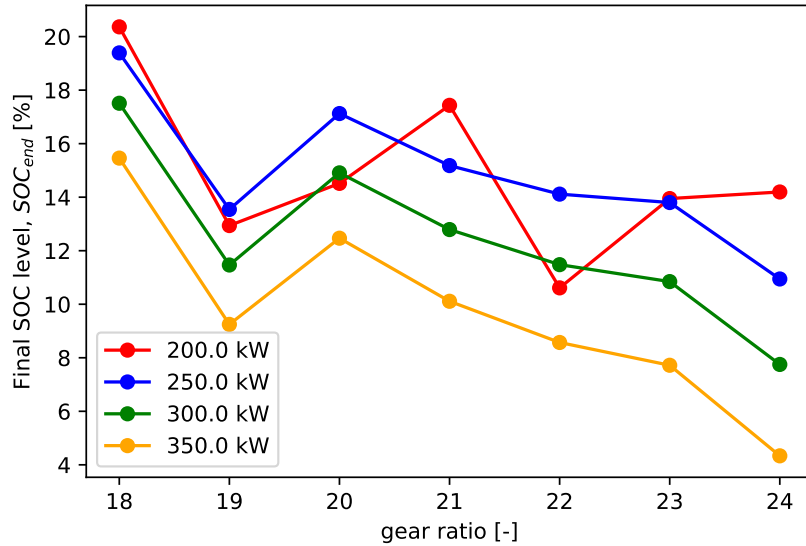


Figure 3.8: Battery state of charge level at the end of route using identical PMSM machines, gear ratios of both the axles and equal torque request. The results here presented for machine and gear ratio on one e-axis.

with differences arising only from actuator utilisation and powertrain efficiency. In this configuration, both axles operate simultaneously with equal force requests; therefore, the distinction between cruise and startability modes is no longer applicable. Instead, this configuration can be interpreted as a CEP, where two identical electric machines contribute to the vehicle propulsion.

Compared with the configuration presented in Paper A, the achievable driving range is reduced by approximately 5–6%, and larger gear ratios are required to achieve the optimal performance. The best-performing configurations correspond to electric machine ratings of 200 kW, 250 kW, and 300 kW combined with gear ratios of 21, 20, and 18, on each axle respectively.

Finally, the influence of increasing the contribution of the cruise axle during startability operation was investigated by configuring the startability axle with a smaller gear ratio of 14 while increasing the cruise axle gear ratio. The corresponding results are shown in Figure 3.9. Compared with the baseline configuration, this approach further reduces the achievable driving range. Moreover, a pronounced sensitivity to the transmission gear ratio is observed around values of 21–22 for all electric machine ratings, which is attributed to the characteristics of the electric machine efficiency maps (i.e. numerical issues in defining the maps).

Overall, the investigated use case indicates that lower cruise axle gear ratios combined with equal electric machine ratings on both axles provide the highest

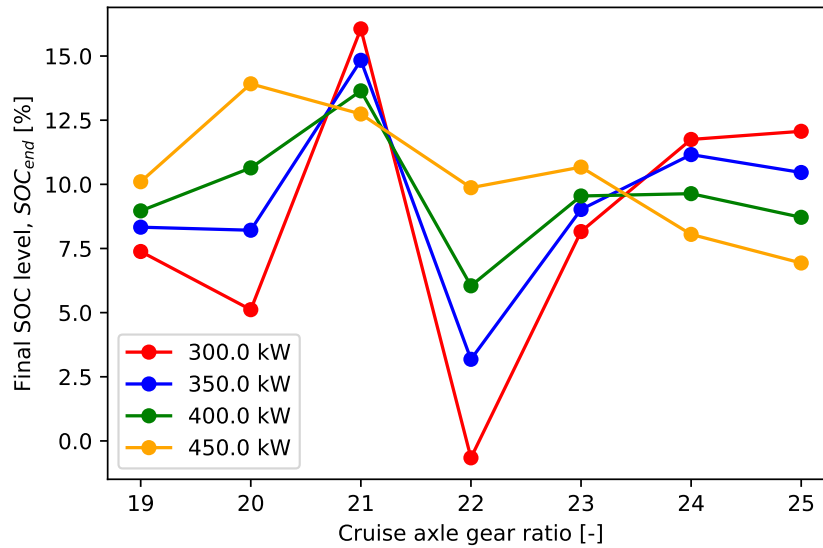


Figure 3.9: Battery state of charge level at the end of route for different cruise axle gear ratios and cruise axle machine powers using a constant startability axle gear ratio of 14.

driving range when used together with the proposed control strategy. These conclusions are, however, specific to the considered drive cycle and electric machine characteristics. Different operating conditions or electric machine designs may lead to different optimal configurations and therefore require a similar analysis. Furthermore, the study employs a quasi-static single-wheel vehicle model that does not account for vehicle dynamics, axle load limits, tyre force constraints, or packaging restrictions, all of which may influence the optimal hardware configuration and force allocation strategy.

Chapter 4

Motion control by actuator and axle coordination

Road vehicles are typically equipped with multiple actuator systems, each designed to fulfil specific functional requirement. For example, the braking system must provide sufficient retardation over the complete operating range and maintain vehicle standstill on inclines, whereas the propulsion system must provide high startability torque at low speeds and high continuous power at high speeds. With the introduction of electric propulsion systems, these functional boundaries become less distinct, as electric machines can contribute not only to propulsion but also to braking and, depending on the vehicle configuration, vehicle motion control. Although these additional functions are limited by actuator bandwidth and system configuration, they increase the available actuation capability and enable the distribution of vehicle-level motion requests among multiple actuators.

Consequently, the vehicle becomes an over-actuated system, where the number of available actuator inputs exceeds the number of controlled degrees of freedom. Therefore, a given vehicle-level motion request can be achieved through multiple actuator combinations. This redundancy enables the use of control allocation strategies to optimally distribute actuator effort according to objectives such as minimising actuator utilisation, tyre wear, energy consumption, or drivetrain power losses.

4.1 Wheel torque control through actuator coordination

4.1.1 Over-Actuation in the Longitudinal Direction

For a CEP architecture, where multiple electric machines are coordinated as a single propulsion unit, longitudinal actuator coordination is relatively straightforward when inter-axle force distribution is not considered. A suitable strategy is to utilise the electric machines for propulsion and regenerative braking within their operational limits, while supplementing with friction brakes when additional braking force is required.

However, the coordination problem becomes more complex for DEP architectures. In such configurations, the longitudinal vehicle motion can become over-actuated even without considering lateral or yaw control objectives. By lumping all electric machines on the same axle into a single equivalent electric machine, a vehicle equipped with a cruise axle, a startability axle, and friction brakes can be represented as generating the required longitudinal force through three actuator groups:

$$F_x^{\text{req}} = F_{x,crs}^{\text{req}} + F_{x,stb}^{\text{req}} + F_{x,Brk}^{\text{req}} \quad (4.1)$$

where $F_{x,crs}^{\text{req}}$, $F_{x,stb}^{\text{req}}$, and $F_{x,Brk}^{\text{req}}$ denote the requested longitudinal force contributions from the cruise axle, startability axle, and friction brakes, to be transmitted through the wheel–road contacts, respectively. The individual wheel forces generated by the electric machines are lumped into equivalent axle forces, and the total friction brake forces are represented as equivalent longitudinal forces for compact formulation.

Although the physical actuator inputs are torque requests, the actuator coordination problem in this thesis is formulated in the force domain except in MPC formulation in Paper E. A low-level conversion from requested force to actuator torque is assumed but is not explicitly included in the actuator coordination formulation. The relationship between the requested axle forces and actuator torques can be expressed as

$$F_x^{\text{req}} = \frac{T_{EM,crs}^{\text{req}}}{r_w} + \frac{T_{EM,stb}^{\text{req}}}{r_w} + \frac{T_{Brk}^{\text{req}}}{r_w} \quad (4.2)$$

where the sign convention for braking torque is defined such that it generates a negative longitudinal force.

The longitudinal actuator coordination problem can therefore be formulated as a control allocation problem, where the requested vehicle-level longitudinal force is distributed among redundant actuators while satisfying actuator constraints

and selected optimisation objectives. Different optimisation formulations for this problem are discussed in the following sections.

4.1.2 Actuator coordination strategies

In this section, two optimisation-based formulations for the actuator coordination problem are presented. The first approach formulates the problem as a weighted multi-objective regularised least-squares optimisation, where different performance objectives are balanced through appropriate weighting factors. The second approach is based on physical modelling of actuator power losses and aims to achieve an energy-efficient distribution of actuator effort. These formulations represent different strategies for distributing the required longitudinal force among the available actuators while satisfying system constraints and selected objectives.

To implement these formulations, the vehicle representation used for actuator coordination is simplified using the single-wheel vehicle model described in section 2.3.1. Consequently, only actuator constraints are considered in the optimisation formulation, while axle-wise tyre force limitations are neglected. The simplified representation of the actuator coordination process is illustrated in Figure 4.1. The actuator coordination problem is solved at each control instance to determine the actuator force requests required to achieve the desired longitudinal force.

Considering over-actuation only in the longitudinal direction, the actuator request vector is defined as

$$\mathbf{u}^* = \begin{bmatrix} F_{x,crs}^{\text{req}*} & F_{x,stb}^{\text{req}*} & F_{x,Brk}^{\text{req}*} \end{bmatrix}^T$$

The actuator force requests are constrained by the available actuator capabilities. The lower and upper bounds of the actuator force vector are defined as:

$$\mathbf{u}_l = \begin{bmatrix} F_{x,crs}^{\min} & F_{x,stb}^{\min} & F_{x,Brk}^{\min} \end{bmatrix}^T$$

$$\mathbf{u}_u = \begin{bmatrix} F_{x,crs}^{\max} & F_{x,stb}^{\max} & F_{x,Brk}^{\max} \end{bmatrix}^T$$

The individual actuator force limits are calculated from the corresponding torque capabilities as follows:

$$F_{x,i}^{\min} = \frac{T_{EM,i}^{\min}(\omega_{EM,i})}{r_w}, \quad F_{x,Brk}^{\min} = \frac{T_{Brk}^{\min}}{r_w}, \quad i \in \{crs, stb\}$$

$$F_{x,i}^{\max} = \frac{T_{EM,i}^{\max}(\omega_{EM,i})}{r_w}, \quad F_{x,Brk}^{\min} = 0$$

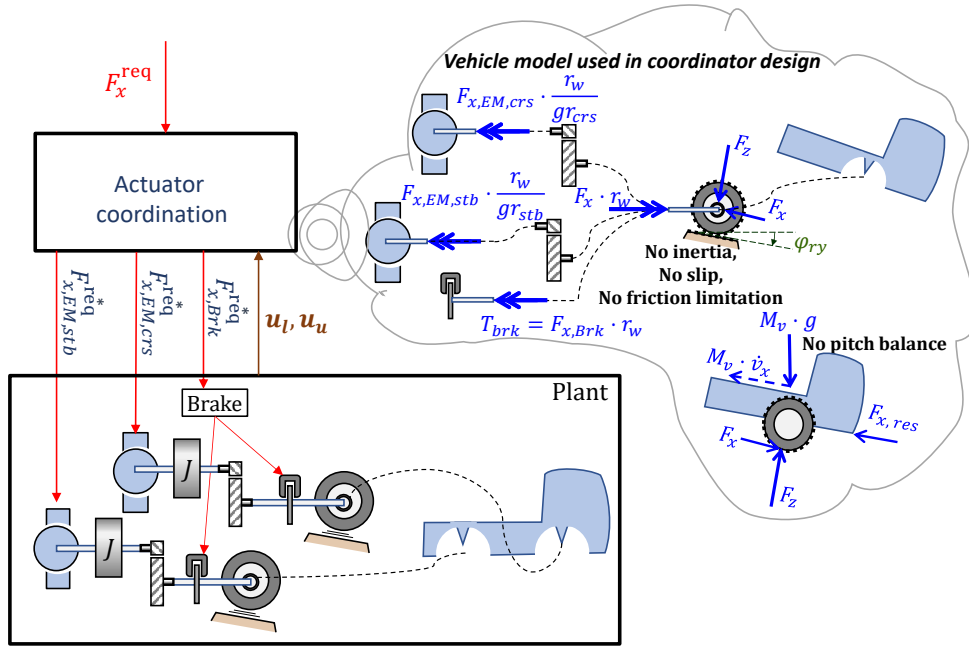


Figure 4.1: A simplified depiction of the actuator coordination process, where the vehicle is represented using a single wheel vehicle model in the coordination framework.

where $T_{EM,i}^{\min}$ and $T_{EM,i}^{\max}$ denote the minimum and maximum physical torque capabilities of the electric machine, respectively, and $T_{Brk}^{\min}$ represents the minimum friction brake torque capability of lumped friction brake actuator.

4.1.2.1 Weighted multi-objective optimisation

The actuator coordination problem is formulated as a constrained weighted least-squares optimisation problem, where the objective is to determine the optimal actuator force request vector $\mathbf{u}^*$ while balancing actuator utilisation and virtual force tracking performance:

$$\begin{aligned} \mathbf{u}^* &= \arg \min_{\mathbf{u}} \left(\|W_u(\mathbf{u}_{des} - \mathbf{u})\|_2^2 + \gamma \|W_v(B\mathbf{u} - \mathbf{v})\|_2^2 \right) \\ \text{s.t. } \mathbf{u}_l &\leq \mathbf{u} \leq \mathbf{u}_u, \quad \mathbf{u}_{des} = [0, 0, 0]^T \end{aligned} \quad (4.3)$$

Weighted least-squares optimisation is a commonly used approach for control allocation in the literature [38, 61, 102]. The objective function consists of two quadratic terms with different interpretations. The first term penalises deviation from the desired actuator usage vector $\mathbf{u}_{des}$, thereby enabling actuator prioritisation. The second term penalises the error between the achieved generalised forces

$B\mathbf{u}$ and the requested virtual control vector $\mathbf{v}$, where B represents the control effectiveness matrix.

The desired actuator usage vector $\mathbf{u}_{des}$ is not an optimisation variable but is provided externally to the control allocation problem. It can be generated using heuristic rules, actuator prioritisation strategies, or previous actuator states. The weighting matrices W_u and W_v determine the relative importance and numerical scaling of the actuator usage and virtual force tracking objectives, respectively. The scalar parameter γ adjusts the trade-off between actuator utilisation and tracking performance. Since both objectives are formulated in the force domain, the corresponding quadratic terms have the same physical units, while the weighting matrices are used to tune their relative contribution.

The resulting formulation is a convex quadratic programming (QP) problem, which is suitable for real-time implementation in over-actuated systems where multiple actuator combinations can satisfy the same vehicle-level force request. This formulation is referred to as a mixed optimisation problem in Paper C and as global force error minimisation in Paper B, since the virtual force tracking term is typically given a higher weighting and therefore dominates the optimisation.

The calculation of $\mathbf{u}_{des}$ and W_u is not the main focus of this thesis. In Papers A–C, a prioritisation strategy is applied where the PMSM on the cruise axle is assumed to operate preferentially. Consequently, the cruise axle is utilised up to its available capability, and any remaining force demand is allocated to the startability axle, followed by the friction brakes. This prioritisation is achieved through the tuning of W_u . The influence of this tuning on the force distribution between the cruise and startability axles is illustrated in Figure 4.2.

4.1.2.2 Power loss minimisation

Power loss represents the instantaneous energy dissipation in the electric machines and friction brakes. Since this energy is dissipated as heat and cannot be recovered, power loss provides a physically meaningful and readily quantifiable objective for actuator coordination. When combined with appropriate cost factors (for example $\frac{\epsilon}{s}$ or equivalently $\frac{\epsilon}{m}$), the power loss can later be expressed as an economic cost rate; however, in this formulation it is used directly as the minimisation objective.

The considered power losses represent the dominant powertrain-related losses. Other loss sources, such as tyre slip losses and rolling resistance, are neglected in this formulation since the focus is on the coordination of propulsion and braking actuators.

The actuator coordination problem is expressed as an optimisation formulation

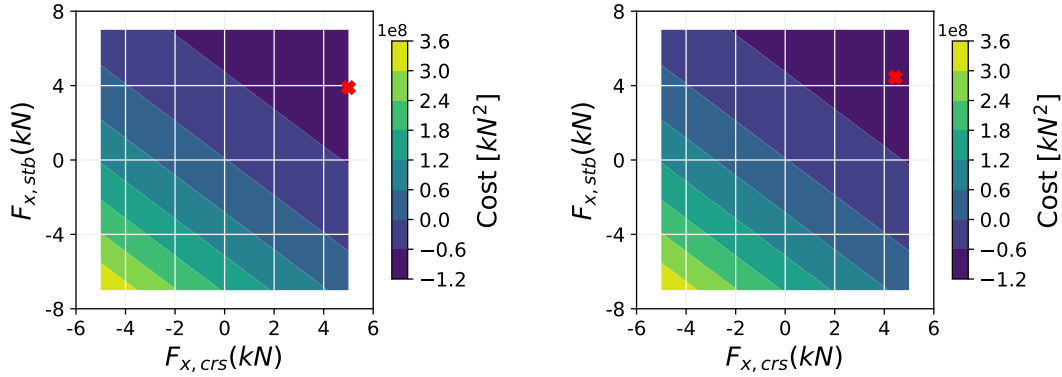


Figure 4.2: Influence of W_u values on force allocation with $u_{des} = [0; 0]$, $\mathbf{v} = 8895N$, $\mathbf{u}_l = [-5000, -7000]$, $\mathbf{u}_u = [5000, 7000]$ and with $B = [1, 1]$ and $W_v = 1$. The X and Y axis limits in the plots represent the range of force allocation possible for the cruise and startability axle respectively. The cost of the entire solution space, represented by the contours, is achieved by solving the cost function in Eq.(4.3) for each point on the X and Y axis. Optimal solutions are represented by the red cross marker for each of the following configurations of W_u and γ used: (left): $W_u = \text{diag}(10^{-5}, 10^{-2})$, $\gamma = 10^2$; (right): $W_u = \text{diag}(1, 1)$, $\gamma = 10^2$

solving for optimal actuator requests, $\mathbf{u}^*$:

$$\begin{aligned} \mathbf{u}^* &= \arg \min_{\mathbf{u}} \left(\sum_{i \in \{crs, stb\}} P_{EM,i}^{\text{loss}} + P_{Brk}^{\text{loss}} \right) \\ \text{s.t. } B\mathbf{u} &= \mathbf{v} \\ \mathbf{u}_l &\leq \mathbf{u} \leq \mathbf{u}_u \end{aligned} \quad (4.4)$$

Since both $\mathbf{u}$ and $\mathbf{v}$ are defined in the force domain, B is dimensionless and represents the contribution of each actuator to the total longitudinal force. For the considered longitudinal allocation problem,

$$B = \begin{bmatrix} 1 & 1 & 1 \end{bmatrix}$$

The term $P_{EM,i}^{\text{loss}}$ represents the electric machine power loss associated with axle i , while P_{Brk}^{loss} represents the power loss of the lumped friction brake system. The vectors $\mathbf{u}_l$ and $\mathbf{u}_u$ define the lower and upper actuator force limits, respectively.

The power loss characteristics of each electric machine are obtained from the torque–speed maps described in Section 2.4.1.2. Figure 4.3, illustrates the variation of electric machine losses with torque for different vehicle speeds. The torque limits of each machine are represented by the end points of the corresponding curves.

For the optimisation formulation, the electric machine losses are approximated using a quadratic regression model:

$$P_{EM,i}^{\text{loss}} \approx a_i \cdot T_{EM,i}^2 + b_i \cdot T_{EM,i} + c_{EM,i} \quad (4.5)$$

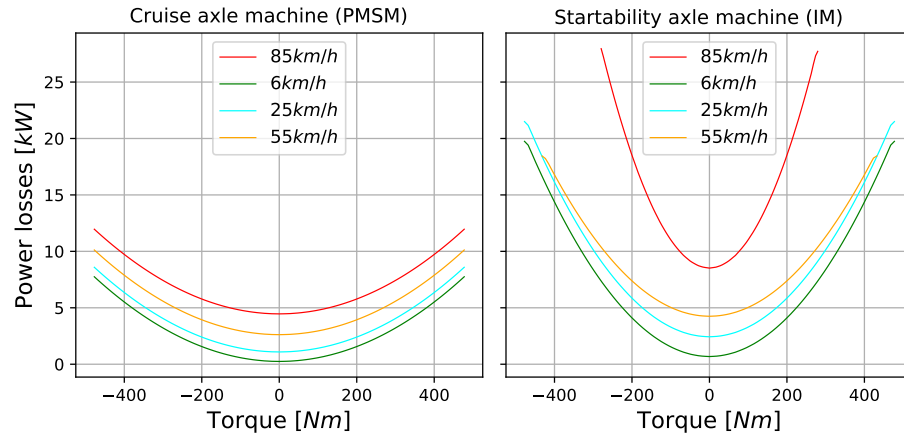


Figure 4.3: Power losses of electric machines as a function of torque for different vehicle speeds generated using the loss models in Section 2.4.1.2. The offset of power losses at zero torque is referred to as idle losses which is higher at higher speeds.

where a_i , b_i , and $c_{EM,i}$ are regression coefficients obtained from the electric machine loss maps. The constant term $c_{EM,i}$ represents the zero-torque losses associated with the electric machine, including idle or magnetisation losses.

The mechanical power absorbed by the friction brakes is dissipated as heat and is therefore considered as a complete loss. The corresponding lumped brake power loss is expressed as:

$$P_{Brk}^{loss} = -T_{Brk} \cdot \omega_w \quad (4.6)$$

where ω_w is the wheel angular velocity of the wheel in the single-wheel model. The negative sign follows the convention that braking torque is negative during braking operation.

Figure 4.4 illustrates the power loss minimisation problem by showing the power loss characteristics of each actuator as a function of wheel torque at a given vehicle speed. The electric machine torque and loss characteristics are transformed to the wheel domain using the transmission ratio. The objective of the actuator coordinator is to select the operating point of each actuator such that the total requested wheel force is achieved while minimising the total power loss.

The power loss minimisation problem can be reformulated as a quadratic programming problem by substituting the quadratic electric machine loss models into the objective function defined in Eq. (4.4).

$$\begin{aligned} \mathbf{u}^* &= \arg \min_{\mathbf{u}} \quad \frac{1}{2} \mathbf{u}^T H \mathbf{u} + g^T \mathbf{u} \\ \text{s.t.} \quad & B \cdot \mathbf{u} = \mathbf{v} \end{aligned} \quad (4.7)$$

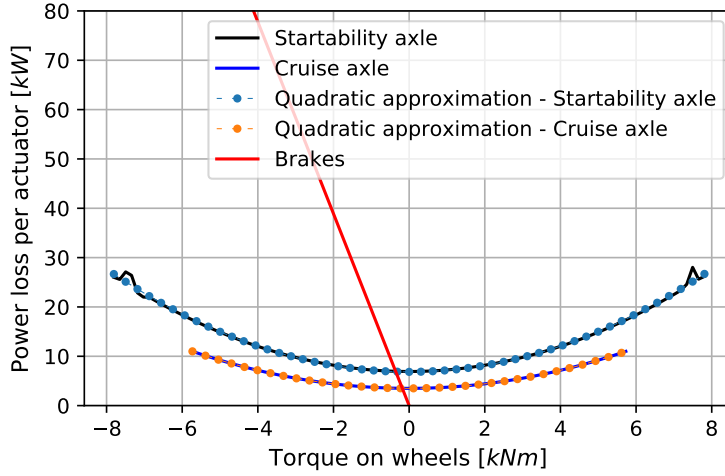


Figure 4.4: Power losses of electric machines with their quadratic approximations and friction brakes, as a function of wheel torque, for a vehicle speed of 70 km/h, as shown in Paper B. The curve fit parameters for the regressed quadratic approximations are: **a)** cruise axle electric machine - $a_{crs} = 0.033$, $b_{crs} = -0.0002$, $c_{crs} = 3498.44$ giving an R^2 fit of 0.99 **b)** Startability axle electric machine - $a_{stb} = 0.17$, $b_{stb} = 0.038$, $c_{stb} = 6838.84$ giving an R^2 fit of 0.99.

$$\mathbf{u}_l \leq \mathbf{u} \leq \mathbf{u}_u$$

The Hessian matrix and linear coefficient vector are obtained from the quadratic loss approximation:

$$H = 2 \cdot \begin{bmatrix} \frac{r_w^2}{gr_{crs}^2} \cdot a_{crs} & 0 & 0 \\ 0 & \frac{r_w^2}{gr_{stb}^2} \cdot a_{stb} & 0 \\ 0 & 0 & a_{Brk} \end{bmatrix}, \quad g^T = \left[\frac{r_w}{gr_{crs}} \cdot b_{crs}, \frac{r_w}{gr_{stb}} \cdot b_{stb}, -\omega_w \cdot r_w \right]$$

If the requested force exceeds the combined actuator capability, the equality constraint $B\mathbf{u} = \mathbf{v}$ cannot be satisfied. Throughout this thesis, the requested force is saturated before the optimisation, ensuring that a feasible solution exists. A small quadratic coefficient, $a_{Brk} = 10^{-5}$, is introduced for the friction brake term to ensure a positive definite Hessian matrix. Only positive brake force requests are considered in the quadratic programming formulation. Extending the formulation to near-standstill and reversing conditions requires additional modifications and is outside the scope of this thesis.

4.1.2.3 Comparative analysis of actuator coordination methods

Both formulations achieve the desired longitudinal force through actuator coordination; however, they differ fundamentally in their objective interpretation,

constraint handling, and tuning requirements. The differences between the two approaches are discussed in terms of problem formulation and solution performance.

Problem formulations

The power loss minimisation formulation in Eq.(4.4) can be interpreted as a physical cost-rate optimisation problem. The objective consists of actuator power losses, $P_{\text{loss},ij}$, expressed in watts (J/s), which directly quantify the energy dissipated by the electric machines and friction brakes. These quantities can subsequently be converted into economic cost rates (like $\frac{\text{€}}{\text{s}}$) through appropriate scaling factors. Therefore, the optimisation objective has a direct physical interpretation, and the contribution of each actuator is explicitly represented without requiring additional weighting parameters.

In contrast, the weighted multi-objective formulation in Eq.(4.3) is a penalty-based optimisation approach, where actuator utilisation and force tracking errors are combined through weighting matrices. The objective terms are formulated in the force domain and do not directly represent physical energy losses or economic costs. Instead, the matrices W_u , W_v , and the scalar parameter γ determine the relative importance of actuator usage and force tracking objectives. Consequently, the solution depends on appropriate tuning of these parameters.

A fundamental difference between the two approaches is therefore the interpretation of the optimisation objective. The power loss formulation provides a physically grounded framework that can be directly extended to include additional cost components, such as tyre wear, battery degradation, or brake pad wear. In contrast, the weighted formulation provides greater flexibility in defining objectives but requires additional tuning to achieve the desired actuator prioritisation. The weighted formulation lacks an inherent physical basis; even if the weights are assigned units (e.g., $(\frac{\text{€}}{N})$), they do not originate from explicit cost models but rather from heuristic design choices. Consequently, the optimisation outcome is highly dependent on parameter tuning and may vary across operating conditions.

Furthermore, the power loss formulation enforces the global force request as a hard constraint, ensuring exact tracking (subject to actuator limits), whereas the weighted formulation treats it as a soft constraint. This introduces a trade-off between tracking accuracy and actuator usage in the weighted formulation as shown in Eq.(4.3), which must be carefully balanced through weight selection.

In summary, the power loss minimisation approach provides a physically grounded and interpretable cost-rate optimisation framework, while the weighted multi-objective formulation constitutes a flexible but tuning-dependent penalty-based approach. This distinction has important implications for robustness, scalability, and the ability to incorporate additional cost components in a consistent manner.

Solution performance

Both approaches result in convex quadratic programming problems and are therefore suitable for real-time actuator coordination. However, they differ in how the global force request is handled. In the power loss minimisation formulation, the requested virtual control vector is enforced as a hard constraint:

$$Bu = v \quad (4.8)$$

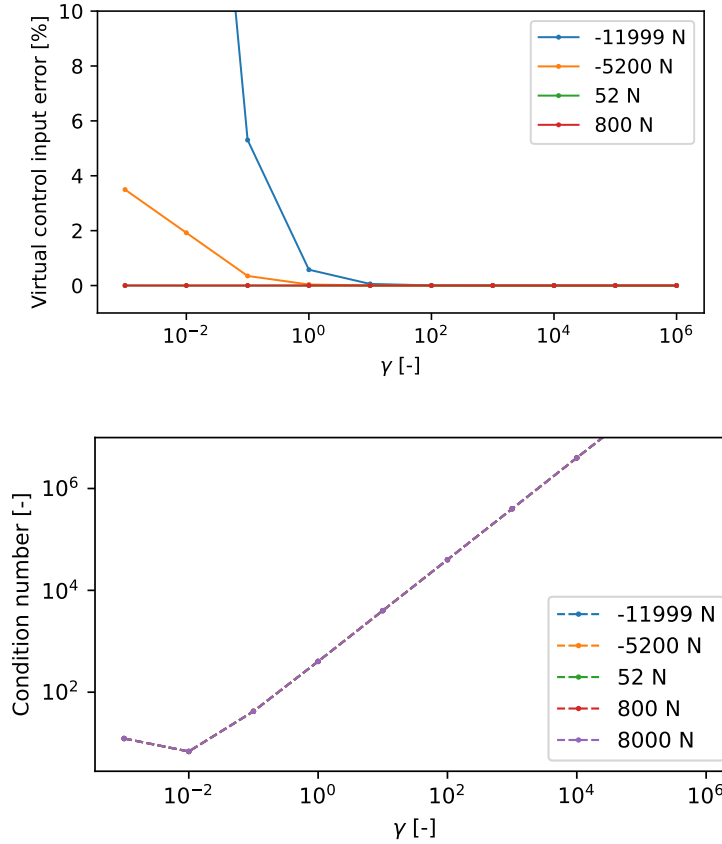


Figure 4.5: Influence of weighting factor γ on virtual control input error and condition number. (a) The variation of virtual control input error $Bu - v$ is calculated for different virtual control inputs with a fixed $W_u = \text{diag}(10^{-5}, 10^{-2})$. Log scale is used for the x-axis. (b) Condition number variation with γ for different virtual control inputs with a fixed $W_u = \text{diag}(10^{-5}, 10^{-2})$. Log scale is used for both x and y axes.

Consequently, the optimisation focuses solely on minimising actuator power losses while satisfying the required longitudinal force. When a feasible solution exists, the formulation provides a unique optimum with predictable Karush-Kuhn-Tucker (KKT) conditions and does not require objective-weight tuning. The

resulting performance is primarily determined by the accuracy of the actuator models, constraints, and numerical properties of the QP formulation.

The main limitation of the equality-constrained formulation is feasibility. If the requested force exceeds the combined actuator capability, the optimisation problem becomes infeasible. In practice, this can be addressed by saturating the incoming force request or introducing additional relaxation variables.

In the weighted multi-objective formulation, the force tracking requirement is instead incorporated into the objective function:

$$J = \|W_u(\mathbf{u}_{\text{des}} - \mathbf{u})\|_2^2 + \gamma \|W_v(B\mathbf{u} - \mathbf{v})\|_2^2$$

This formulation remains feasible even when the requested force cannot be fully achieved, since tracking errors are allowed. Therefore, it provides a smooth degradation of performance under actuator saturation conditions. However, this flexibility introduces sensitivity to the selection of W_u , W_v , and γ . The weighting parameters directly influence actuator prioritisation, force tracking accuracy, and numerical conditioning of the QP.

Increasing γ increases the importance of force tracking and makes the solution approach the equality-constrained formulation, but may increase the Hessian condition number. Conversely, reducing γ improves numerical conditioning at the expense of larger force tracking errors. The influence of γ on numerical conditioning and optimisation error is illustrated in Figure 4.5.

The comparison presented in Paper C shows that the power loss minimisation formulation generally provides better numerical conditioning, with a lower Hessian condition number compared with the weighted multi-objective formulation. Therefore, the equality-constrained formulation is less sensitive to parameter scaling and solver perturbations.

Considering the objective interpretation, reduced tuning requirements, and improved numerical robustness, the power loss minimisation approach is selected for further analysis in this thesis.

4.2 Wheel torque control through actuator and axle coordination

4.2.1 Over-actuation in the planar vehicle motion

In addition to longitudinal motion, a road vehicle operating in the horizontal plane exhibits coupled longitudinal, lateral, and yaw dynamics. Under the assumption that vertical, pitch, and roll motions are negligible, the planar vehicle motion can be described by three degrees of freedom. The corresponding vehicle dynamics

are illustrated in Figure 4.6 and expressed as:

$$\begin{aligned}
 M_v(\dot{v}_x - v_y \cdot \omega_z) &= F_{x,fL} + F_{x,fR} + F_{x,rL} + F_{x,rR} = F_x^{\text{req}} \\
 M_v(\dot{v}_y + v_x \cdot \omega_z) &= F_{y,fL} + F_{y,fR} + F_{y,rL} + F_{y,rR} = F_y^{\text{req}} \\
 I_z \dot{\omega}_z &= -F_{x,fL} \cdot t/2 + F_{x,fR} \cdot t/2 - F_{x,rL} \cdot t/2 + F_{x,rR} \cdot t/2 \\
 &\quad + F_{y,fL} \cdot l_f + F_{y,fR} \cdot l_f - F_{y,rL} \cdot l_r - F_{y,rR} \cdot l_r = M_z^{\text{req}}
 \end{aligned} \tag{4.9}$$

The variables $F_{x,il}$ and $F_{y,il}$ denote the longitudinal and lateral tyre forces at wheel l on axle i , while F_x^{req} , F_y^{req} , and M_z^{req} are the global force and yaw moment requests generated by the vehicle motion controller.

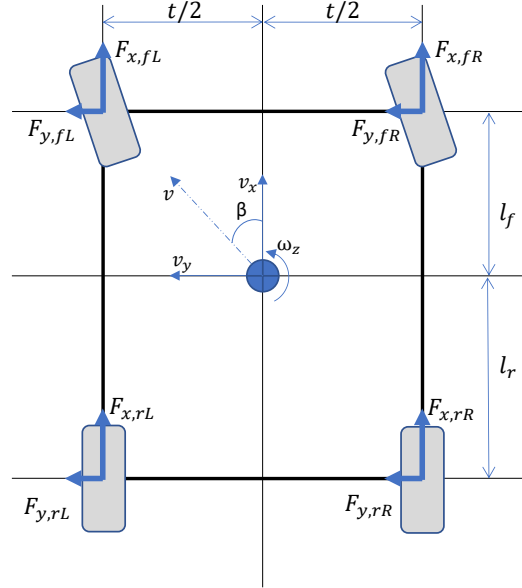


Figure 4.6: Two track vehicle model in the road plane having longitudinal, lateral and yaw rotational degree of freedom.

The corresponding global motion request is defined as:

$$\mathbf{v} = \begin{bmatrix} F_x^{\text{req}} \\ F_y^{\text{req}} \\ M_z^{\text{req}} \end{bmatrix} \tag{4.10}$$

Unlike the purely longitudinal case, the planar actuator coordination problem must simultaneously satisfy multiple coupled motion objectives. The available tyre force at each wheel is limited by the friction envelope; therefore, the achievable longitudinal force depends on the lateral force, and vice versa. These constraints directly affect the feasible combinations of F_x , F_y , and M_z .

One approach is to incorporate the non-linear tyre force coupling directly into the vehicle motion controller to generate feasible and safe motion requests. However, this increases controller complexity and makes the formulation more sensitive to modelling uncertainties. In this thesis, tyre force limitations are instead handled within the actuator coordination layer by including them as constraints in the optimisation problem. This maintains a modular control structure, where the motion controller generates desired vehicle-level forces and moments, while the actuator coordinator determines feasible actuator requests.

Consequently, the actuator coordination problem extends from longitudinal force distribution to a constrained allocation problem that maps the global motion request $\mathbf{v}$ to feasible actuator requests while considering actuator limits and tyre force constraints. This formulation forms the basis for the integrated over-actuated vehicle motion control framework developed in the following chapters.

4.2.2 Adding axle constraints to fulfill axle level coordination

To account for the coupling between longitudinal and lateral tyre forces within the actuator coordination framework, axle-level tyre force constraints are introduced into the optimisation problem. These constraints ensure that the allocated longitudinal forces remain feasible within the available tyre-road friction envelope while preserving the modular structure of the control architecture. The friction circle formulation is adopted as a computationally efficient representation of the coupled tyre force limitations. To obtain a model suitable for optimisation, the following assumptions are adopted:

- No lateral load transfer is considered, allowing the left and right wheel forces to be combined into equivalent axle forces, corresponding to a single-track vehicle representation.
- Only propulsion and braking actuators (electric machines and friction brakes) are considered in the actuator coordination problem, while steering actuation and its dynamics are excluded (known input to, but not requested by, the optimisation).

Under the additional assumption of quasi-steady-state cornering, the planar vehicle dynamics in Eq.(4.9) reduces to

$$\begin{aligned} M_v \dot{v}_x &= F_{x,f} + F_{x,r} \approx F_x^{\text{req}} \\ M_v v_x \omega_z &= F_{y,f} + F_{y,r} \approx F_y^{\text{pred}} \end{aligned} \quad (4.11)$$

The required lateral force corresponding to the desired vehicle motion is distributed between the axles based on their available normal loads. This provides

an estimate of the axle-level lateral force demand required for the friction constraints in the actuator coordination problem. Therefore the predicted lateral force demand at each axle is therefore calculated as:

$$F_{y,f}^{\text{pred}} = F_{z,f} \frac{v_x \omega_z}{g} \quad (4.12)$$

$$F_{y,r}^{\text{pred}} = F_{z,r} \frac{v_x \omega_z}{g} \quad (4.13)$$

where $F_{z,f}$ and $F_{z,r}$ denote the normal loads on the front and rear axles, respectively. These loads can be estimated dynamically depending on the available vehicle model considering both lateral and longitudinal load transfer.

Depending on the vehicle configuration and powertrain architecture, the number of actuators associated with each axle may differ. Therefore, the longitudinal force equilibrium is expressed in a generalised form as the following:

$$\sum_{i=1}^n \left(\sum_{j=1}^{M_{EM,i}} F_{x,EM,ij}^{\text{req}} + \sum_{j=1}^{M_{Brk,i}} F_{x,Brk,ij}^{\text{req}} \right) = F_x^{\text{req}} \quad (4.14)$$

where n denotes the number of axles, $M_{EM,i}$ and $M_{Brk,i}$ denote the number of electric machines and friction brake actuators connected to axle i , respectively. The total number of actuators associated with axle i is given by

$$m_i = M_{EM,i} + M_{Brk,i} \quad (4.15)$$

The longitudinal force allocated to the axle is constrained by the available tyre-road friction according to:

$$-F_{x,i}^{\text{lim}} \leq \left(\sum_{j=1}^{M_{EM,i}} F_{x,EM,ij}^{\text{req}} + \sum_{j=1}^{M_{Brk,i}} F_{x,Brk,ij}^{\text{req}} \right) \leq F_{x,i}^{\text{lim}} \quad (4.16)$$

The available longitudinal force capacity is obtained from the friction circle as:

$$F_{x,i}^{\text{lim}} = \sqrt{(\mu F_{z,i})^2 - (F_{y,i}^{\text{pred}})^2} \quad (4.17)$$

Here, μ is the tyre-road friction coefficient, $F_{z,i}$ is the normal load on axle i , and $F_{y,i}^{\text{pred}}$ is the predicted lateral force demand at that axle. Consequently, $F_{x,i}^{\text{lim}}$ represents the remaining longitudinal force capability at tyre-road contact (friction limited) available for actuator coordination after accounting for the predicted lateral tyre force requirement.

The axle-level tyre force constraints are subsequently included in the actuator coordination optimisation to ensure that the allocated actuator forces remain within the feasible tyre force envelope.

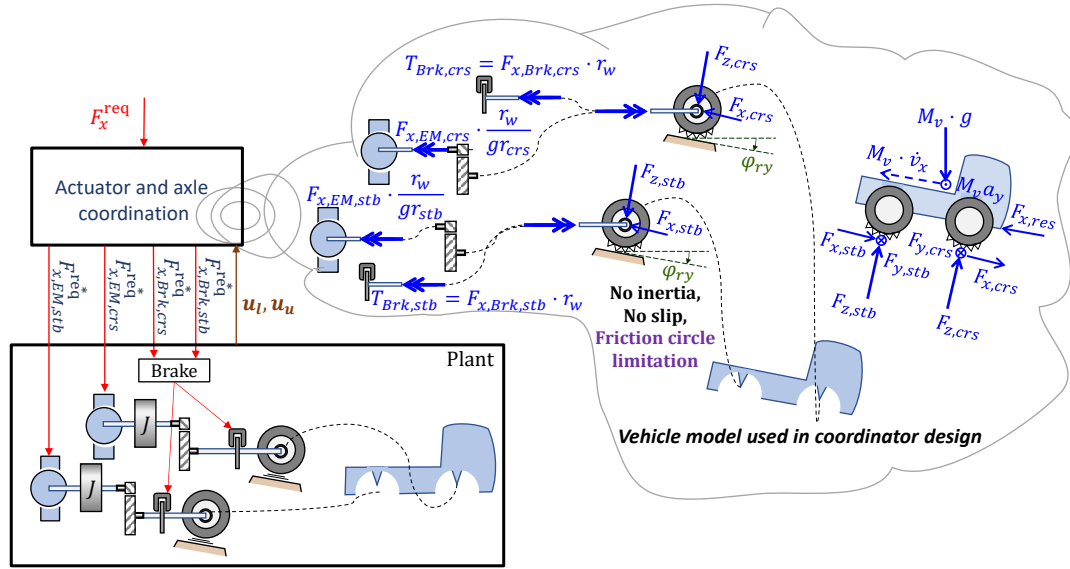


Figure 4.7: Pictorial abstract of the actuator coordination by adding axle constraints for a 4x4 vehicle configuration with cruise and startability axles. The vehicle model is represented as a two axle model in the actuator coordinator design.

4.2.3 Actuator and axle coordination strategies

This section presents a comparative evaluation of two actuator and axle coordination strategies for battery electric heavy vehicles. The investigated approaches aim to improve energy efficiency while maintaining feasible vehicle operation under nominal driving conditions. The underlying actuator and axle coordination framework is based on the methodology presented in Paper D.

The investigated strategies are formulated on the principle of instantaneous actuator power loss minimisation, using simplified power loss models of the electric powertrains and friction brakes to determine an energy-efficient wheel torque distribution. In both the strategies, the wheel force limitations are explicitly included as constraints to ensure feasible actuator operation and prevent violation of the available tyre force envelope.

4.2.3.1 Power loss minimisation with axle force constraints (PLM)

The power loss minimisation formulation introduced previously can be extended to include axle-level tyre force constraints. In this formulation, actuator forces are distributed to minimise the instantaneous power losses of the electric machines and friction brakes while satisfying actuator capability limits and axle-level tyre force constraints. The vehicle representation adopted for the actuator and axle coordination framework is illustrated in Figure 4.7.

The actuator and axle coordination problem is formulated as:

$$\begin{aligned} \mathbf{u}^* = \arg \min_{\mathbf{u}} & \left(\sum_{i=1}^n \left(\sum_{j=1}^{M_{EM,i}} F_{EM,ij}^{\text{loss}} + \sum_{j=1}^{M_{Brk,i}} F_{Brk,ij}^{\text{loss}} \right) \right) \\ \text{s.t. } & \mathbf{u}_l \leq \mathbf{u} \leq \mathbf{u}_u \\ & B\mathbf{u} = \mathbf{v} \\ & -\mathbf{h} \leq G\mathbf{u} \leq \mathbf{h} \end{aligned} \quad (4.18)$$

where $\mathbf{u}$ is the actuator force request vector, $\mathbf{v} = F_x^{\text{req}}$ is the requested global longitudinal force, and B represents the mapping between actuator forces and the vehicle-level force request. Since both actuator requests and global requests expressed in the force domain, B becomes a row vector containing ones:

$$B = [1 \quad 1 \quad \dots \quad 1] \in \mathbb{R}^{1 \times q} \quad (4.19)$$

The total number of actuators connected to axle i is defined as:

$$m_i = M_{EM,i} + M_{Brk,i} \quad (4.20)$$

where $M_{EM,i}$ and $M_{Brk,i}$ denote the number of electric machines and friction brake actuators connected to axle i , respectively. The total number of actuators in the vehicle is therefore

$$q = \sum_{i=1}^n m_i = \sum_{i=1}^n (M_{EM,i} + M_{Brk,i}) \quad (4.21)$$

where n denotes the number of axles.

The actuator force request vector is separated into electric machine and friction brake components:

$$\mathbf{u} = \begin{bmatrix} \mathbf{u}_{EM} \\ \mathbf{u}_{Brk} \end{bmatrix} \quad (4.22)$$

where the actuator vectors are constructed by collecting actuator requests over all axles:

$$\mathbf{u}_{EM} = \text{col} \left(F_{x,EM,ij}^{\text{req}} \right), \quad i = 1, \dots, n; j = 1, \dots, M_{EM,i} \quad (4.23)$$

$$\mathbf{u}_{Brk} = \text{col} \left(F_{x,Brk,ij}^{\text{req}} \right), \quad i = 1, \dots, n; j = 1, \dots, M_{Brk,i} \quad (4.24)$$

The matrix G represents the actuator-to-axle mapping and depends on the powertrain architecture and actuator distribution across the axles. It maps the actuator forces to the corresponding axle forces and is defined as:

$$G \in \mathbb{R}^{2n \times q}. \quad (4.25)$$

The first n rows of G formulate the lower limit of axle force constraints, whereas the remaining n rows formulate the upper limit of axle force constraints. The elements of G are defined as:

$$G_{r,j} \in \{-1, 0, 1\} \quad (4.26)$$

A value of one indicates that actuator j contributes to the corresponding axle force constraint, while zero indicates no contribution. The negative rows are introduced to represent the lower force limits in the form $-\mathbf{h} \leq G\mathbf{u}$.

The axle force constraint vector is defined as:

$$\mathbf{h} = [F_{x,1}^{\text{lim}} \quad \dots \quad F_{x,n}^{\text{lim}} \quad F_{x,1}^{\text{lim}} \quad \dots \quad F_{x,n}^{\text{lim}}]^T \quad (4.27)$$

The lower and upper actuator force limits are obtained by collecting the corresponding actuator limits:

$$\mathbf{u}_l = \begin{bmatrix} \mathbf{u}_{l,EM} \\ \mathbf{u}_{l,Brk} \end{bmatrix}, \quad \mathbf{u}_u = \begin{bmatrix} \mathbf{u}_{u,EM} \\ \mathbf{u}_{u,Brk} \end{bmatrix} \quad (4.28)$$

where

$$\mathbf{u}_{l,EM} = \text{col}(F_{x,EM,ij}^{\text{min}}), \quad i = 1, \dots, n; j = 1, \dots, M_{EM,i} \quad (4.29)$$

$$\mathbf{u}_{u,EM} = \text{col}(F_{x,EM,ij}^{\text{max}}), \quad i = 1, \dots, n; j = 1, \dots, M_{EM,i} \quad (4.30)$$

$$\mathbf{u}_{l,Brk} = \text{col}(F_{x,Brk,ij}^{\text{min}}), \quad i = 1, \dots, n; j = 1, \dots, M_{Brk,i} \quad (4.31)$$

$$\mathbf{u}_{u,Brk} = \text{col}(F_{x,Brk,ij}^{\text{max}}), \quad i = 1, \dots, n; j = 1, \dots, M_{Brk,i} \quad (4.32)$$

The available longitudinal force capacity of each axle is obtained from the friction circle:

$$F_{x,i}^{\text{lim}} = \sqrt{(\mu F_{z,i})^2 - (F_{y,i}^{\text{pred}})^2} \quad (4.33)$$

Therefore, the axle constraints ensure that the allocated longitudinal force remains feasible considering the simultaneous lateral tyre force demand.

Remark: The global longitudinal force request is limited before optimisation to ensure that the equality constraint remains feasible. Consequently, the optimisation always satisfies the vehicle-level force request within the available actuator and tyre force capabilities.

Example: CEP and DEP powertrain configurations

The modularity of the proposed actuator and axle coordination framework is demonstrated using **CEP** and **DEP** powertrain architectures. In both cases, the same general optimisation formulation in Eq.(4.18) is applied. The difference between the configurations is represented through the actuator-to-axle mapping matrix G and the corresponding actuator force vector $\mathbf{u}$, actuator limits $\mathbf{u}_l$ and $\mathbf{u}_u$, and power loss terms.

Centralised electric propulsion: The **CEP** configuration represents a two-axle vehicle where the electric machines are connected to a common driveline on one axle, resulting in a 4×2 configuration. In this example, the electric machine actuators are connected to the rear axle, while friction brake actuators are available on both axles. The actuator force vector is therefore defined as:

$$\mathbf{u}_{CEP} = [F_{x,EM,21}^{\text{req}} \quad F_{x,EM,22}^{\text{req}} \quad F_{x,Brk,11}^{\text{req}} \quad F_{x,Brk,21}^{\text{req}}]^T \quad (4.34)$$

The corresponding actuator-to-axle mapping matrix is

$$G_{CEP} = \begin{bmatrix} 0 & 0 & -1 & 0 \\ -1 & -1 & 0 & -1 \\ 0 & 0 & 1 & 0 \\ 1 & 1 & 0 & 1 \end{bmatrix} \quad (4.35)$$

The first two rows of G_{CEP} impose the lower axle force limits, while the last two rows impose the corresponding upper axle force limits. The mapping reflects that the electric machines contribute only to axle 2, whereas the friction brakes contribute independently to both axles.

Distributed electric propulsion: The **DEP** configuration represents a two-axle vehicle with independent electric machines on both axles, corresponding to a 4×4 configuration. Each actuator contributes only to the longitudinal force of the axle to which it is connected. The actuator force vector becomes:

$$\mathbf{u}_{DEP} = [F_{x,EM,11}^{\text{req}} \quad F_{x,EM,21}^{\text{req}} \quad F_{x,Brk,11}^{\text{req}} \quad F_{x,Brk,21}^{\text{req}}]^T \quad (4.36)$$

The corresponding actuator-to-axle mapping matrix is

$$G_{DEP} = \begin{bmatrix} -1 & 0 & -1 & 0 \\ 0 & -1 & 0 & -1 \\ 1 & 0 & 1 & 0 \\ 0 & 1 & 0 & 1 \end{bmatrix} \quad (4.37)$$

In contrast to the **CEP** configuration, each actuator contributes only to its corresponding axle force. Therefore, the mapping matrix directly represents the distributed nature of the electric propulsion system.

The two examples demonstrate that different powertrain architectures can be handled within the same optimisation framework. The actuator characteristics, constraints, and objective function remain unchanged, while the actuator distribution and vehicle topology are captured through the actuator indexing and the mapping matrix G .

4.2.3.2 Idle-loss-aware power loss minimisation with axle force constraints (ILPLM)

The PLM strategy presented in the previous section determines the actuator force distribution assuming that all electric machines remain continuously energised. While this assumption enables the actuator coordination problem to be formulated as a convex quadratic programming (QP) problem, it restricts the optimisation to operating points where all electric machines remain active and therefore does not necessarily minimise the total powertrain losses. When additional degrees of freedom, such as a mechanical disconnect clutch ¹ or electrical de-energisation of induction machines, are available, further reductions in power loss can be achieved by selectively deactivating individual actuators. Consequently, the idle losses associated with inactive electric machines should be considered explicitly during actuator coordination.

As shown in Figure 4.3 the induction machine exhibits considerably higher idle losses than the PMSM over the operating range considered in this thesis. Consequently, for the cruise and startability axle architecture, it is advantageous to minimise the utilisation of the IM whenever its contribution to the requested longitudinal force is small. In the quadratic power loss approximation introduced in Section (4.5), the idle losses represented by the constant term $c_{EM,i}$. Since this constant does not influence the optimisation variables, it has no effect on the QP solution and is therefore unable to influence actuator selection. As a result, the conventional PLM formulation cannot determine whether energising an electric machine is beneficial from a system-level energy perspective.

To address this limitation, the idle-loss-aware power loss minimisation (ILPLM) strategy, proposed in Paper D, augments the PLM formulation with a supervisory decision layer that explicitly accounts for idle losses while retaining the computational efficiency of the underlying QP formulation. Rather than modifying the optimisation problem itself, the supervisory layer evaluates alternative actuator operating modes and selects the one that minimises the total instantaneous power loss.

- The conventional PLM optimisation is first solved to obtain the optimal actuator force distribution and the corresponding actuator power losses.
- The resulting power loss is then compared with alternative operating modes in which the induction machine remains de-energised. In these modes, the

¹Introducing clutches into the system leads to additional cost. Moreover, for PMSMs clutches are necessary to exploit the idles losses. Even at zero torque and zero stator current, the permanent magnets continue to produce air-gap flux. When the machine remains mechanically connected and rotating, this flux induces back-EMF in the stator, leading to iron losses and electromagnetic drag torque.

requested longitudinal force is generated either solely by the PMSM or, during braking, by the PMSM together with the friction brakes.

- If the requested longitudinal force can be supplied by the PMSM alone and the corresponding total power loss is lower than that obtained from the PLM solution, the induction machine remains de-energised.
- If the PMSM alone cannot satisfy the requested force, the supervisory layer compares the power losses associated with activating the induction machine against those obtained by supplementing the PMSM with friction braking, and selects the operating mode with the lowest total power loss.

An overview of the operating modes and decision logic for the two-axle vehicle configuration is illustrated in Figure 4.8 and can be summarised as follows:

In this manner, the ILPLM strategy extends conventional power loss minimisation from actuator force allocation to actuator operating mode selection. By explicitly accounting for idle losses, the algorithm determines not only the optimal distribution of actuator forces but also whether energising the induction machine is energetically justified under the prevailing operating conditions.

The proposed supervisory strategy is well suited to the two-axle cruise and startability axle architecture considered in Paper D and can be extended to three-axle vehicles with two driven axles [52]. However, as the number of independently controllable electric machines increases, the number of possible operating modes grows rapidly, making rule-based supervisory logic increasingly difficult to maintain. For more complex powertrain architectures, mixed-integer optimisation provides a systematic framework for simultaneously determining actuator activation and force allocation, although at the expense of substantially higher computational complexity.

4.2.3.3 Comparison of the actuator and axle coordination strategies

Although both the PLM and ILPLM strategies satisfy the requested longitudinal force while respecting the actuator and axle-level tyre force constraints, they exhibit fundamentally different actuator coordination behaviour due to their treatment of electric machine idle losses. This section compares the two strategies with emphasis on their operating characteristics, energy efficiency, and system-level behaviour. The comparison considers the two-axle **DEP** configuration presented in Paper D.

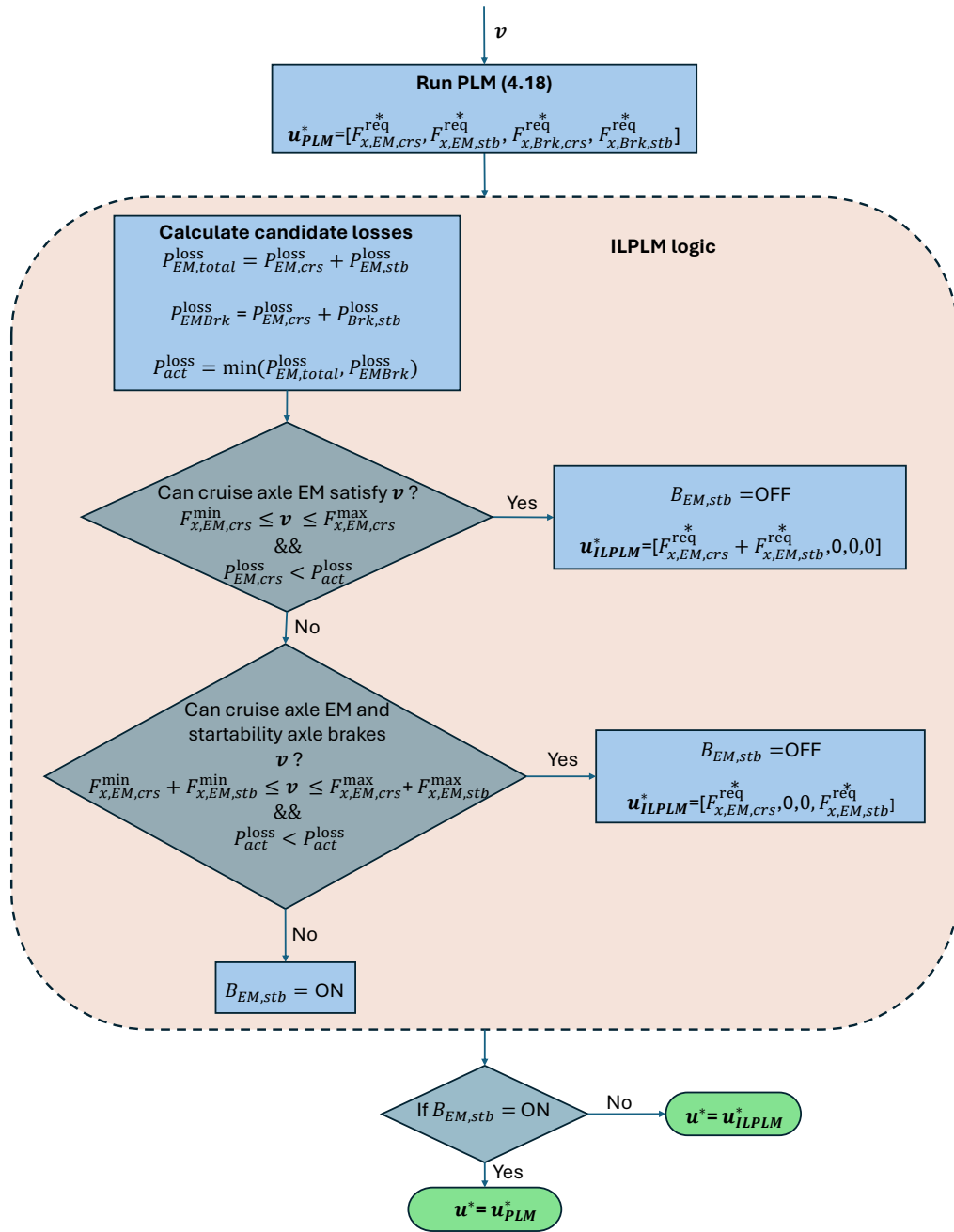


Figure 4.8: Simplified representation of ILPLM decision logic for the two-axle vehicle **DEP** architecture described in Paper D.

Power loss characteristics of PLM and ILPLM

The PLM strategy minimises the instantaneous power losses of the electric machines and friction brakes while assuming that all electric machines remain energised. Consequently, the requested longitudinal force is distributed among the available actuators to minimise load-dependent losses, regardless of the idle losses associated with maintaining the induction machine in an active state.

In contrast, the ILPLM strategy additionally accounts for induction machine idle losses during actuator coordination. Consequently, the optimisation balances the reduction in load-dependent losses achieved through force sharing against the idle losses incurred by energising the induction machine. As a result, ILPLM determines both the optimal actuator force distribution and whether activating the induction machine is energetically justified under the prevailing operating conditions.

Figure 4.9 compares the operating characteristics of PLM and ILPLM in the force–velocity plane using contours of total power loss. For PLM, the contour levels represent the total power loss assuming that all electric machines remain energised. For ILPLM, the contours additionally include operating modes in which the induction machine is de-energised and, during braking, operating modes combining regenerative and friction braking. The thick blue boundaries indicate the maximum achievable longitudinal force using only the electric drivetrains, assuming that the tyre and friction brake capabilities exceed those of the electric machines.

In the upper subplot, the area enclosed by the red boundary identifies operating points where it is more energy-efficient to de-energise the induction machine and satisfy the requested longitudinal force solely using the PMSM on the cruise axle. Within this region, the reduction in load-dependent losses obtained by distributing the force demand between both electric machines is smaller than the idle loss associated with keeping the induction machine energised.

Similarly, in the lower subplot, the area enclosed by the red and cyan boundaries identifies operating points where supplementing regenerative braking with friction braking on the startability axle results in lower total power loss than activating the induction machine. This operating region occurs only at very low vehicle speeds, where the induction machine operates inefficiently and the additional loss associated with friction braking is comparatively small. Consequently, its practical significance is limited.

Outside these operating regions, PLM and ILPLM converge to the same actuator allocation and therefore yield identical power losses. Under these conditions, the requested longitudinal force is sufficiently high that the reduction in load-dependent electrical losses achieved through force sharing outweighs the idle loss penalty associated with energising the induction machine.

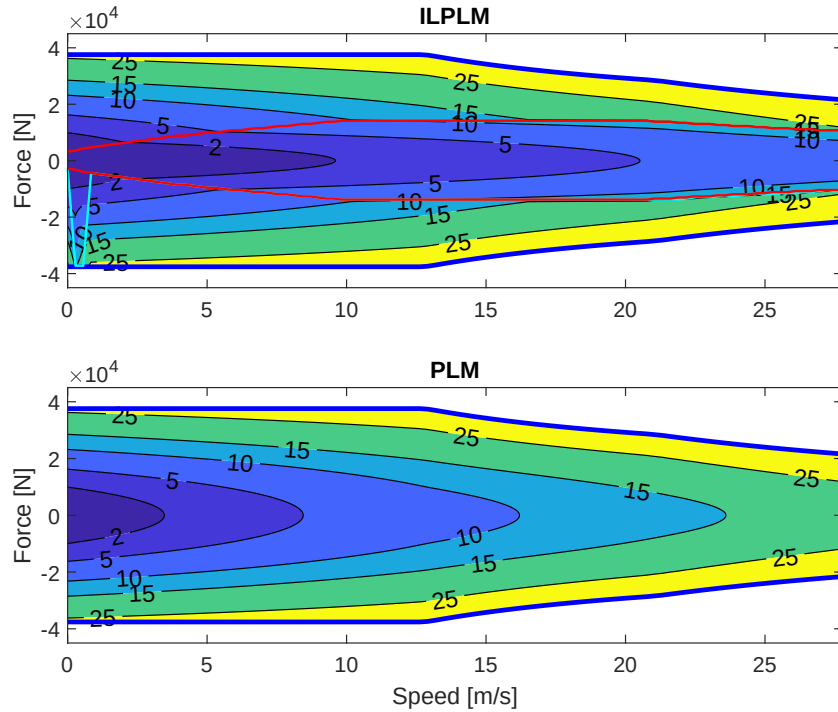


Figure 4.9: Contour of power losses obtained using the ILPLM and PLM algorithms, represented using the traction diagram. The contour levels represent the power losses associated with the operating electric machines and friction brakes in the case of the ILPLM and without friction brakes for the PLM. The blue thick lines represent the limits of the total maximum force achieved using only the drivetrains. The contour level values are expressed in kW . In the upper subplot, the area between the red curves represents where it is optimal to demagnetise the startability axle powertrain. Furthermore, a similar region enclosed between the lower red and cyan curves, shows where it is optimal to use friction brakes instead of electric machines on the startability axle.

Overall, the comparison demonstrates that ILPLM extends the conventional PLM formulation by incorporating actuator activation decisions based on idle losses. Consequently, lower system-level power losses can be achieved under low-load operating conditions, while both strategies provide identical actuator coordination when operation of both electric machines is energetically favourable.

Energy consumption of PLM and ILPLM for a use-case

The energy consumption of the PLM and ILPLM strategies is evaluated for a 4×4 tractor with a three-axle semi-trailer operating over a representative long-haul transport duty cycle presented in Paper D. The comparison is performed based on the accumulated battery energy consumption over the considered driving cycle.

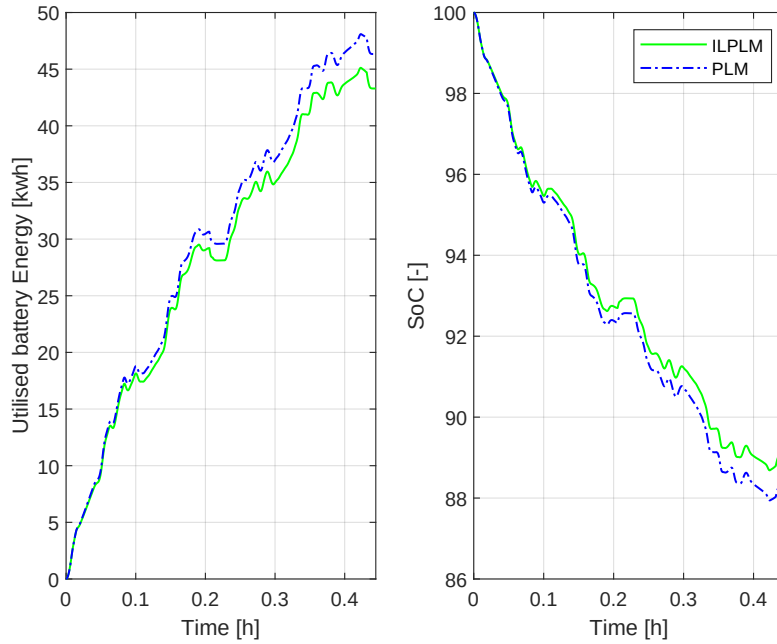


Figure 4.10: Energy consumption and battery SoC levels in the PLM and ILPLM allocation strategies for use case defined in Paper D.

The total battery energy consumption is calculated by integrating the traction energy demand, electric drivetrain losses, and friction brake losses over time.

As shown in Figure 4.10, achieves the lowest overall energy consumption, reducing the battery energy consumption by approximately 7% compared with PLM for the investigated application. The energy benefit is primarily obtained during low-load operating conditions, where the ability to deactivate the startability axle induction machine reduces unnecessary idle losses.

During the initial acceleration phase, both strategies exhibit similar energy consumption since the demanded traction force requires utilisation of both electric axles. The energy trajectories begin to diverge when the vehicle enters cruising operation and the required longitudinal force decreases. Under these conditions, ILPLM deactivates the induction machine and supplies the required force primarily through the cruise axle PMSM, whereas PLM continues to utilise both electric machines. This demonstrates the benefit of including actuator activation as an additional degree of freedom in actuator coordination.

Figure 4.10, further highlights the contributions from different loss mechanisms over the driving cycle. The calculation method for these losses is described in Paper D. The reduction in total energy consumption achieved by ILPLM is mainly attributed to reduced electric drivetrain losses resulting from reduced in-

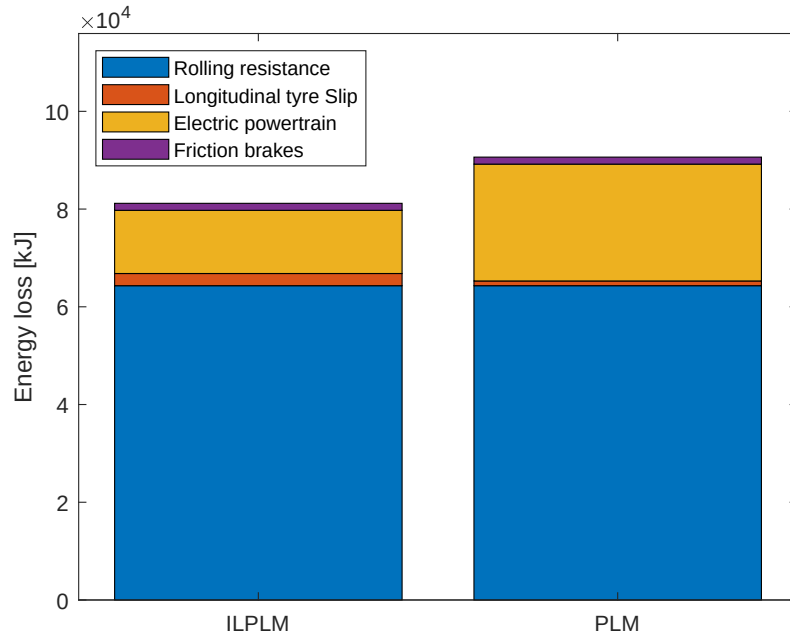


Figure 4.11: Energy consumed by different sources, presented in Paper D, for the PLM and ILPLM strategies.

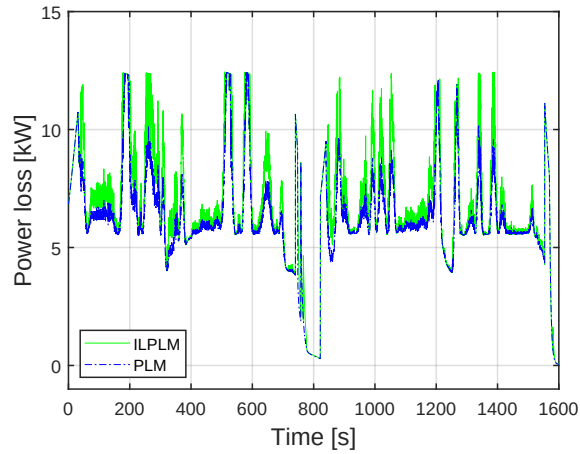
duction machine utilisation. In comparison, friction brake losses remain similar between the two strategies, while rolling resistance contributes only marginally to the difference in total energy consumption.

The improved energy efficiency achieved by ILPLM introduces a trade-off in tyre utilisation. Since the startability axle is used less frequently, a larger portion of the requested longitudinal force is generated by the cruise axle. This increases the longitudinal slip demand on the utilised axle and highlights the importance of considering tyre-related effects and stability constraints when extending actuator coordination towards energy optimisation.

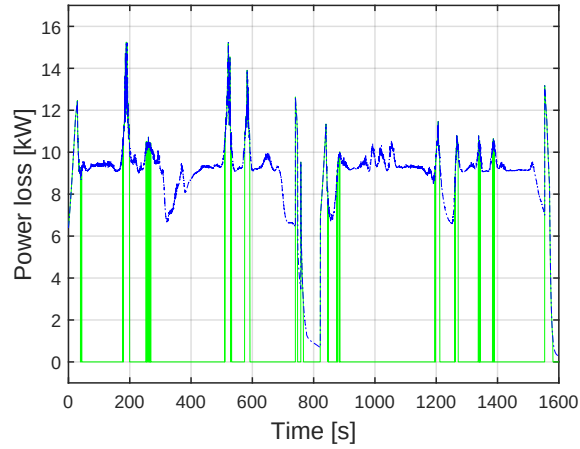
To further verify the origin of the energy savings, the axle-wise power losses and induction machine operating state are analysed in Figure 4.12. The results show that, under ILPLM operation, the startability axle drivetrain remains inactive for extended periods, reducing the idle losses associated with maintaining the induction machine energised. This behaviour agrees with the force-velocity operating regions identified in Figure 4.9, where single-axle propulsion provides lower total power losses under low-load conditions.

However, frequent transitions between activated and deactivated operating states are observed in certain parts of the driving cycle. Such transitions may introduce additional driveline disturbances and increase actuator wear. Previ-

ous studies [57, 46] have shown that this trade-off between energy efficiency and switching frequency can be addressed using switching penalties, hysteresis constraints, or minimum dwell-time requirements. These aspects are not considered in the present thesis and are therefore identified as future improvements to the actuator coordination strategy. Consequently, the reported energy reduction should be interpreted as an upper bound achievable without additional switching constraints.



(a) Cruise axle



(b) Startability axle

Figure 4.12: Power loss on cruise and startability axle versus time for PLM and ILPLM strategies over the given driving cycle.

Overall, the results demonstrate that the energy benefit of ILPLM originates

from extending actuator coordination beyond force distribution and incorporating actuator activation as an additional optimisation variable. By selectively deactivating the induction machine during low-load operation, ILPLM reduces electric drivetrain losses and improves the overall energy efficiency of the heavy electric vehicle. More generally, the results highlight that actuator operating state is an important degree of freedom in multi-machine electric powertrains, particularly for heavy-duty applications where prolonged low-load operation can make idle losses a significant contributor to total energy consumption.

Chapter 5

Ensuring safe motion control using actuator and axle coordination

The actuator and axle coordinator presented in this thesis operates within the lower control layer of the hierarchical motion control architecture shown in Figure 3.2. Within this framework, a control allocation algorithm instantaneously distributes requested global forces and moments among individual actuators, thereby generating wheel torque requests. Although effective for instantaneous optimisation, the coordinator is inherently myopic: it considers only the current operating condition and does not account for the future evolution of the vehicle states.

These limitations are particularly pronounced in electric HCVs. Their large mass and yaw inertia, coupled longitudinal–lateral dynamics arising from load transfer, high centre of gravity, and varying payload conditions result in longer dynamic time constants and reduced responsiveness to corrective actions. Consequently, instability events can develop more rapidly relative to the controller’s ability to react, increasing the consequences of suboptimal torque allocation.

Furthermore, the coordinator optimises a predefined objective function and therefore cannot adapt its optimisation priorities according to the vehicle operating condition. Near the handling limits, where longitudinal, lateral and yaw dynamics become strongly coupled, it neither anticipates future vehicle behaviour nor explicitly balances energy efficiency against future stability margins. Consequently, closed-loop stability cannot be guaranteed over a prediction horizon.

To address the challenges discussed above, this chapter presents two approaches for incorporating vehicle-level information into actuator and axle coordination. The main motivation is to extend the operating range of the control allocation framework such that vehicle-level motion constraints are handled proactively, delaying or even avoiding the need for intervention by the vehicle stability controller.

Reducing stability controller interventions not only improves driving comfort by avoiding abrupt corrective actions, but can also improve energy efficiency by enabling more optimal use of the available actuators.

The first approach proposes a model Predictive Control Allocation (MPCA) framework to generate vehicle-level force and moment requests over a prediction horizon of 1 s. The MPC formulation minimises a cost function with state deviations from a reference model and utilization of actuator requests $\mathbf{u}$, see Eq.(5.30). By explicitly considering future vehicle dynamics, the approach enables the coordination system to account for stability constraints while optimising actuator usage. The methodology and results are presented in Paper E and 5.1.

The second approach focuses on the instantaneous determination of front and rear axle wheel-torque distributions while maintaining yaw stability. Building upon the friction-circle-based constraints proposed in [49], the method generates yaw-stable wheel torque distributions for a given force request. In this context, the problem can be interpreted as a predictive control problem with an infinitesimal prediction horizon. The methodology and results are presented in Section 5.2.

5.1 Optimising wheel torque distribution using a hierarchical MPCA framework

The actuator and axle coordinator presented in Eq.(4.18), considers actuator and tyre-force constraints at the individual axle level while optimising the proposed control objectives. However, vehicle-level motion constraints such as yaw stability, rollover stability, and side slip limits are not explicitly incorporated into the optimisation problem.

Furthermore, the coordinator assumes that the requested global forces and moments generated by the hierarchical controllers are feasible. Near the handling limits this assumption may be conservative because tyre force constraints can prevent the requested vehicle forces from being realised. For example, when a vehicle is cornering near the limits of tyre adhesion, friction-circle constraints may prevent additional braking force requests from being realised by the wheel actuators, even when such braking could maintain vehicle stability. Consequently, an additional supervisory control layer is required to overcome these limitations and regulate the global force requests according to the prevailing vehicle operating conditions.

To address these limitations, a hierarchical wheel torque coordination framework is proposed for heavy electric vehicles with distributed electric powertrains. The framework, presented in Paper E, addresses the challenge of coordinating multiple actuators to simultaneously achieve vehicle stability, energy efficiency, and predictable handling characteristics. By incorporating vehicle-level dynamic

constraints, the framework can adapt force and moment requests according to the current operating state of the vehicle.

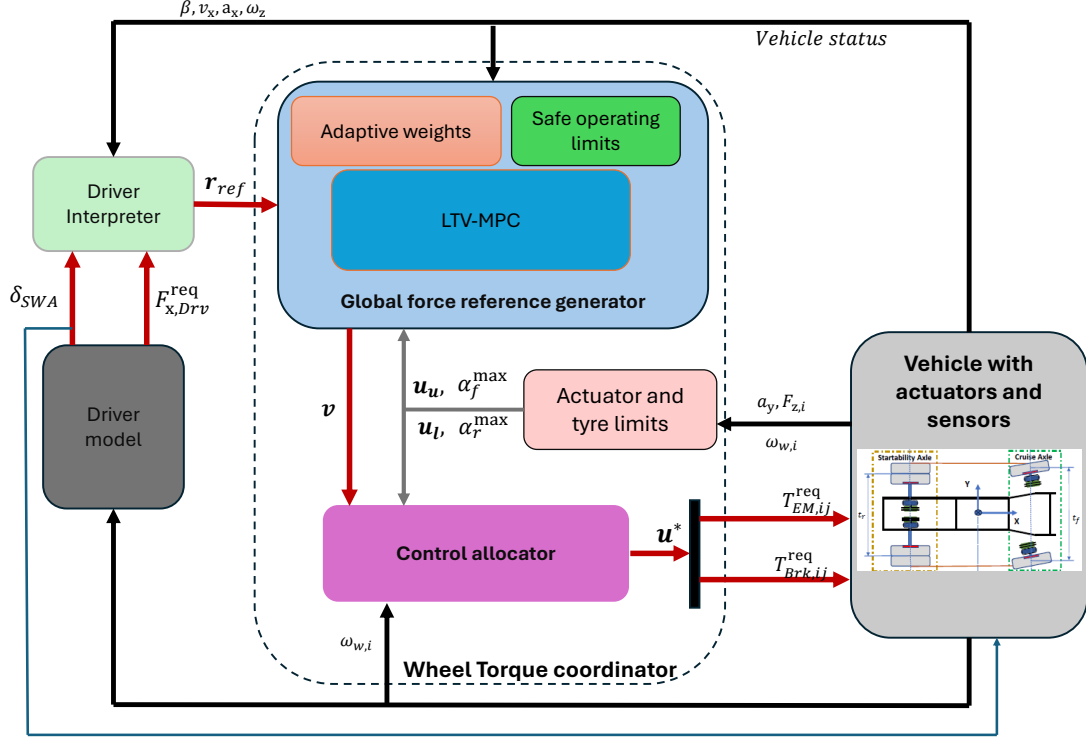


Figure 5.1: System model representing the wheel torque coordination framework used in Paper E.

As illustrated in Figure 5.1, the proposed framework consists of two layers:

- **Global Force Reference Generator:** A supervisory layer that determines optimal vehicle-level force and moment requests while enforcing stability and safety constraints.
- **Control Allocator:** A lower-level layer that distributes the requested forces and moments among the available actuators while minimising power losses and satisfying actuator constraints.

The hierarchical structure provides a modular architecture that reduces controller complexity and computational burden while maintaining flexibility in the choice of optimisation objectives. The framework is evaluated using a 4×4 electric tractor equipped with individual electric machines and friction brakes at each wheel

as presented in Paper E. Although demonstrated on this configuration, the framework can be readily extended to other vehicle and powertrain architectures. The two components within the hierarchical wheel torque coordination framework is presented in the following sections.

5.1.1 MPC based Global force reference generator

The global force reference generator receives interpreted driver requests from the driver interpreter while simultaneously obtaining feedback regarding the vehicle states, actuator limitations, and tyre force constraints. The driver interpreter converts the steering wheel angle input, δ_{SWA} and pedal requests into vehicle-level reference signals according to:

$$\mathbf{r}_{ref} = [a_x^{\text{ref}} \quad v_x^{\text{ref}} \quad \omega_z^{\text{ref}} \quad \beta^{\text{ref}}]^T \quad (5.1)$$

The longitudinal acceleration reference is generated from the single-track vehicle model as:

$$a_x^{\text{ref}} = \frac{F_{x,Drv}^{\text{req}}}{M_v} \quad (5.2)$$

where $F_{x,Drv}^{\text{req}}$ denotes the longitudinal force requested by the driver.

Since longitudinal acceleration and velocity references represent the same driver-intended motion demand, only one of these variables is required depending on the selected MPC formulation.

The reference yaw rate is obtained from the steady-state response of the single-track vehicle model presented in Section 2.3.2. Assuming small steering angles, the steady-state yaw rate is expressed as:

$$\omega_z^{\text{ref}} = \frac{v_x \cdot \delta_{SWA}}{k_{st} (l_f + l_r + M_v K_u v_x^2)} \quad (5.3)$$

where K_u is the understeer gradient. For a neutral steering vehicle, $K_u = 0$.

The reference sideslip angle is obtained from the vehicle lateral velocity as

$$\beta^{\text{ref}} = \tan^{-1} \left(\frac{v_y}{v_x} \right) \approx \frac{v_y}{v_x} \quad (5.4)$$

where the approximation is valid for small sideslip angles.

In torque-vectoring applications, independent tracking of yaw rate and sideslip angle is generally infeasible near the vehicle handling limits, because both quantities are coupled through the available tyre forces. Controller designs therefore either adapt the yaw-rate reference based on the side slip response or prioritise one objective while respecting tyre force constraints [64, 80]. Therefore, the MPC

formulation adapts the generated force and moment requests according to the current vehicle state and tyre constraints.

For the torque-vectoring control problem considered in this chapter, the vehicle-level request vector is defined as the following:

$$\mathbf{v}_r = \begin{bmatrix} F_x^{\text{req}} \\ M_z^{\text{req}} \end{bmatrix} \quad (5.5)$$

where F_x^{req} and M_z^{req} denote the requested longitudinal force and yaw moment, respectively.

These requests are generated using a MPC formulation. MPC is adopted because it enables prediction of future vehicle behaviour while explicitly considering state and input constraints. This allows the supervisory controller to generate feasible vehicle-level force and yaw-moment requests while accounting for vehicle stability requirements. Following the receding-horizon principle, the MPC optimisation predicts the future vehicle behaviour over the prediction horizon, applies only the first optimal control action, and repeats the optimisation at the next sampling instant.

5.1.1.1 Formulation of the vehicle model used in MPC

The prediction model employed by the MPC is obtained by linearising the single-track vehicle model introduced in Section (2.3.2). In contrast to the model used in actuator coordination framework presented in Chapter 4, where only longitudinal forces are coordinated at the axle level, the MPC operates directly with individual wheel torque requests. This enables active torque vectoring, where differences in left and right wheel torques generate an external yaw moment $M_{z,\text{ext}}$.

The continuous-time vehicle dynamics are expressed as

$$\dot{\mathbf{x}}(t) = f(\mathbf{x}(t), \mathbf{u}_{\text{mpc}}(t)), \quad (5.6)$$

with the state and input vectors as:

$$\mathbf{x} = [\dot{v}_x \quad v_x \quad \omega_z \quad \beta]^T, \quad \mathbf{u}_{\text{mpc}} = [T_{w,fL} \quad T_{w,fR} \quad T_{w,rL} \quad T_{w,rR}]^T$$

The longitudinal acceleration state $\dot{v}_x$ is introduced as an augmented state to enable acceleration-based reference tracking within the discrete-time MPC formulation. It does not represent an additional independent vehicle degree of freedom, since the longitudinal acceleration is obtained from the longitudinal force equilibrium.

The non-linear coupling terms in the vehicle dynamics are represented using time-varying parameters updated at each MPC sampling instant t_s . In particular,

relation between quantities such as the longitudinal and lateral velocities¹ and yaw rate are evaluated using the current vehicle state estimate and assumed constant over the prediction horizon.

The steering angle δ_f is also not explicitly included in the prediction model. Instead, it is treated as an operating-point parameter obtained from the vehicle state estimation. This approximation is justified by the relatively slow steering dynamics of heavy commercial vehicles and the short prediction horizon considered in this work.

The control-oriented model is further simplified by two assumptions: (i) longitudinal and lateral tyre forces are decoupled, and (ii) wheel rotational dynamics, are neglected.

The individual lateral tyre forces are described using the non-linear tyre model:

$$F_{y,il} = \mu \cdot F_{z,il} \cdot \tanh \left(\frac{C_{il} \cdot \alpha_{il}}{\mu \cdot F_{z,il}} \right) \quad (5.7)$$

To preserve the convexity of the MPC optimisation problem, the non-linear tyre model is linearised at every sampling instant. The effective cornering stiffness is obtained from the local derivative of the non-linear tyre characteristic:

$$\tilde{C}_{il} = \left. \frac{\partial F_{y,il}}{\partial \alpha_{il}} \right|_{\alpha_{il}(t_s)} \quad (5.8)$$

The resulting linear time-varying tyre representation is updated at each MPC iteration thereby maintaining a linear prediction model over the optimisation horizon.

The interaction between longitudinal and lateral tyre forces is represented through friction constraints rather than a combined-slip tyre model. Thus the lateral tyre force required for evaluating the available friction capacity is predicted from the vehicle lateral acceleration² as:

$$F_{y,il}^{\text{pred}}(t_s) = F_{z,il}(t_s) \frac{a_y(t_s)}{g}, \quad (5.9)$$

The predicted low-complexity approximation of the lateral tyre forces is calculated from the current vehicle operating condition and assumed constant over the prediction horizon.

The longitudinal tyre force limits at each wheel are imposed using the friction ellipse constraint, given by:

$$F_{x,il}^{\text{lim}} = n_{\text{conf}} \cdot \sqrt{(\mu \cdot F_{z,il}(t_s))^2 - \left(F_{y,il}^{\text{pred}}(t_s) \right)^2}$$

¹In Paper E the state v_y of vehicle plant model feedback is used as estimate.

² a_y is used as a proxy for lateral tyre force demand since it is directly available from vehicle measurements or state estimation, avoiding additional tyre force observers. The approach assumes quasi-steady single-track behaviour, negligible tyre relaxation dynamics, quasi-static load transfer, and proportional distribution of lateral forces.

$$\text{where, } n_{\text{conf}} = 0.9 \quad (5.10)$$

where $n_{\text{conf}} = 0.9$ is a confidence factor introduced to account for model uncertainties, parameter variations, and modelling simplifications. The resulting formulation maintains a convex constraint structure suitable for real-time MPC implementation.

Assuming negligible wheel inertia and wheel slip dynamics, the available longitudinal tyre force is converted into equivalent wheel torque as:

$$T_{w,il}^{\text{lim}} = F_{x,il}^{\text{lim}} \cdot r_{w,i} \quad (5.11)$$

where r_w denotes the effective wheel radius.

To reduce computational complexity, the tyre force limits are evaluated at the beginning of each optimisation interval and assumed constant over the prediction horizon. This approximation is suitable for the short prediction horizon considered in this thesis and provides a balance between model accuracy and computational efficiency.

5.1.1.2 Defining vehicle motion constraints

To ensure safe vehicle operation, constraints are imposed not only on the actuator forces and tyre forces, but also on the predicted vehicle states. The yaw-rate and side-slip constraints define a safe operating envelope in the ω_z - β phase plane. The formulation follows the approach presented in [8], where the limits are derived from the available tyre-road friction and maximum allowable tyre slip angles. The yaw-rate and side-slip constraints are evaluated at each MPC sampling instant t_s using the current vehicle states, estimated tyre-road friction coefficient, and axle loads. The resulting limits are provided to the MPC optimisation problem and are assumed constant over the prediction horizon. Therefore, the dependence on the sampling instant is omitted in the following expressions for compactness.

The yaw-rate constraint is obtained from the lateral acceleration limit:

$$|a_y| \leq \mu g \quad (5.12)$$

where the lateral acceleration is approximated by

$$a_y = v_x \omega_z \quad (5.13)$$

Consequently, the yaw-rate limits are defined as:

$$\omega_z^{\text{max}} = \frac{\mu g}{v_x} \quad \omega_z^{\text{min}} = -\frac{\mu g}{v_x} \quad (5.14)$$

where the longitudinal velocity v_x corresponds to the current measured vehicle velocity used for updating the constraint limits. The above formulation is valid

within the intended operating region where $v_x > 0$. No explicit low-speed regularisation is introduced, since the controller is designed for forward driving operation.

The side-slip angle limits are obtained by constraining the front and rear tyre slip angles within their allowable operating range. The maximum and minimum side-slip limits are therefore defined as:

$$\beta^{\max} = \min \left(\tan(\alpha_f^{\max} + \delta_f) - l_f \frac{\omega_z}{v_x}, \tan(\alpha_r^{\max}) + l_r \frac{\omega_z}{v_x} \right) \quad (5.15)$$

$$\beta^{\min} = \max \left(\tan(-\alpha_f^{\max} + \delta_f) - l_f \frac{\omega_z}{v_x}, \tan(-\alpha_r^{\max}) + l_r \frac{\omega_z}{v_x} \right) \quad (5.16)$$

The maximum allowable tyre slip angles, $\alpha_f^{\max}$ and $\alpha_r^{\max}$, impose the boundaries of the safe operating region for the front and rear axles, respectively. These limits are derived from the linear tyre force relationship and are used only for state constraint definition³. They do not modify the non-linear tyre force model used in the MPC prediction model, where the cornering stiffness is updated on-line through local linearisation. In particular, the peak lateral force region and corresponding slip angles depend on vertical load, friction coefficient, and tyre stiffness properties, as described in classical tyre modelling literature [86, 27]. Alternatively, in control-oriented applications, these limits may be selected to define a conservative operational envelope that ensures model validity and numerical robustness within optimisation-based controllers such as MPC [8, 29].

For the considered vehicle model, the slip angle limits are selected as:

$$\alpha_f^{\max} = \arctan \left(\frac{4\mu F_{z,f}}{C_f} \right) \quad (5.17)$$

$$\alpha_r^{\max} = \arctan \left(\frac{4\mu F_{z,r}}{C_r} \right) \quad (5.18)$$

where $F_{z,f}$ and $F_{z,r}$ denote the front and rear axle normal loads, respectively, and C_f and C_r represent the corresponding axle cornering stiffness values.

The longitudinal velocity and acceleration limits are selected according to the operational requirements of the heavy commercial vehicle:

$$v_x^{\min} \leq v_x \leq v_x^{\max} \quad (5.19)$$

³The slip angles of the left and right wheels are lumped into a single equivalent slip angle limit for each axle.

where

$$v_x^{\max} = 25 \text{ m/s} \quad v_x^{\min} = -10 \text{ m/s} \quad (5.20)$$

and the acceleration limits are defined from the available tyre-road friction:

$$a_x^{\min} = -\mu g \quad a_x^{\max} = \mu g \quad (5.21)$$

5.1.1.3 Prioritising the state cost objectives according to vehicle motion

The vehicle motion constraints introduced in the previous section are used to adapt the relative importance of the MPC state tracking objectives according to the current operating condition. The motivation is to reduce the emphasis on longitudinal performance objectives when the vehicle approaches its lateral handling limits, thereby prioritising yaw-rate and side-slip regulation.

The adaptive weighting method modifies the normalised state cost weights based on the current vehicle state, driver input, vehicle parameters, and the externally calculated state limits. The state limits are updated at every MPC sampling instant and are treated as constant parameters over the prediction horizon.

The normalised weights are defined within the range $[0, 1]$. The formulation assumes symmetric state constraints, i.e., $\omega_z^{\min} < 0$, $\omega_z^{\max} > 0$, $\beta^{\min} < 0$, and $\beta^{\max} > 0$.

The normalised yaw-rate tracking weight is defined as:

$$W_{\omega_z, sc} = \begin{cases} 1 - \left(\frac{\omega_z^{\max} - \omega_z^{\text{ref}}}{\omega_z^{\max}} \right), & \omega_z^{\text{ref}} \geq 0 \\ 1 - \left(\frac{\omega_z^{\min} - \omega_z^{\text{ref}}}{\omega_z^{\min}} \right), & \omega_z^{\text{ref}} < 0 \end{cases} \quad (5.22)$$

A visualisation of calculating the normalised weight according to Eq.(5.22) is shown in Figure 5.3.

Similarly, the normalised side-slip tracking weight is defined as

$$W_{\beta, sc} = \begin{cases} 1 - \left(\frac{\beta^{\max} - \beta^{\text{ref}}}{\beta^{\max}} \right), & \beta^{\text{ref}} \geq 0 \\ 1 - \left(\frac{\beta^{\min} - \beta^{\text{ref}}}{\beta^{\min}} \right), & \beta^{\text{ref}} < 0 \end{cases} \quad (5.23)$$

The longitudinal acceleration and velocity tracking objectives are reduced as the vehicle approaches lateral stability limits. Therefore, the corresponding weights are defined as

$$W_{a_x, sc} = W_{v_x, sc} = 1 - \max(W_{\beta, sc}, W_{\omega_z, sc}) \quad (5.24)$$

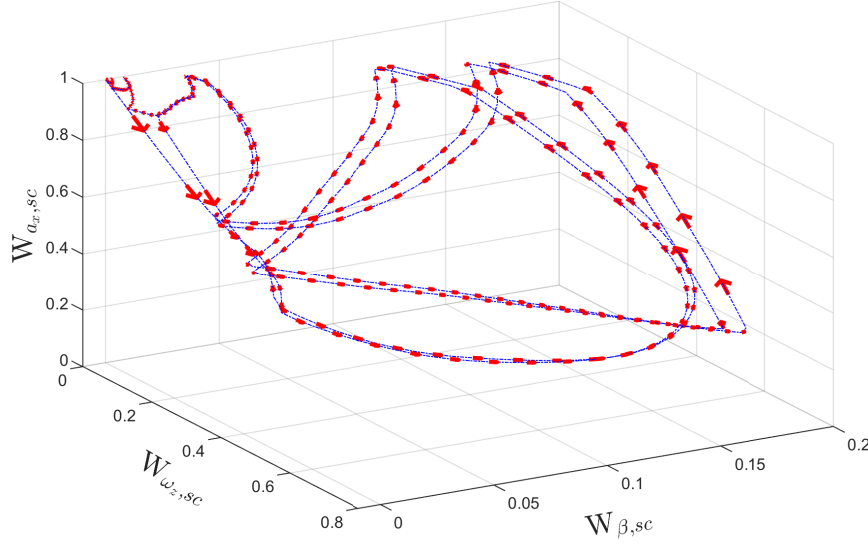


Figure 5.2: Visualisation of trajectory followed by the normalised weights during the double lane change test described in 5.1.3.

Figure 5.2, shows the implementation of normalised weights for a use case in Paper E.

The adaptive weighting formulation therefore prioritises longitudinal tracking during normal driving conditions, while increasing the relative importance of yaw-rate and side-slip regulation when the vehicle approaches the stability boundary.

The normalised weights are subsequently used to scale the MPC state weighting matrix. The weighting matrix is expressed as:

$$\mathbf{Q} = \left(\frac{M_v g}{2} \right)^2 \text{diag} \left[W_{a_x,sc}, W_{v_x,sc}, \frac{4I_z^2}{M_v^2} W_{\omega_z,sc}, \frac{4}{g^2} W_{\beta,sc} \right] \quad (5.25)$$

Remark: In this thesis, the normalised adaptive weights are defined using linear relationships. Although computationally efficient, these relationships may not provide the optimal weighting for all operating conditions or vehicle configurations. The proposed weighting framework can also be extended to additional vehicle states, such as roll angle, if required by the application.

5.1.1.4 MPC problem formulation

The objective of the MPC is to determine the optimal sequence of wheel torque requests that minimises the predicted vehicle state tracking error while satisfying actuator, tyre, and vehicle motion constraints over a finite prediction horizon.

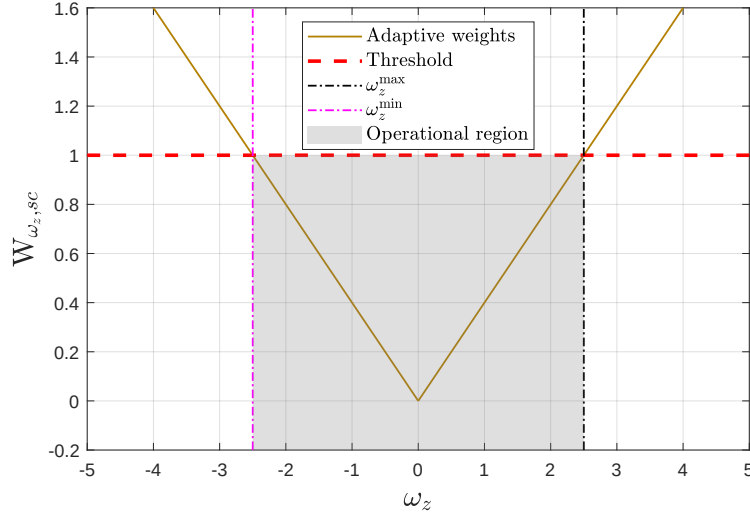


Figure 5.3: Example of evaluating $W_{\omega_z, sc}$ for ω_z using the state limits.

Following the receding-horizon principle, only the first control input of the optimal sequence is applied at each MPC update, after which the optimisation is repeated using updated vehicle measurements.

The state tracking error at the current sampling instant is defined as:

$$\mathbf{e}_x(t_s) = \mathbf{x}(t_s) - \mathbf{r}_{ref}(t_s) \quad (5.26)$$

where $\mathbf{x}$ and $\mathbf{r}_{ref}$ denote the vehicle state and reference vectors introduced in Sections 5.1.1.1 and 5.1.1 respectively. At each MPC update instant, the prediction model is initialised using the measured vehicle state

$$\mathbf{x}_{init} = \mathbf{x}(t_s) \quad (5.27)$$

The reference trajectories for the longitudinal acceleration, longitudinal velocity and body side-slip angle are assumed constant over the prediction horizon, such that

$$a_x^{\text{ref}}(k) = a_x^{\text{ref}}(t_s), \quad v_x^{\text{ref}}(k) = v_x^{\text{ref}}(t_s), \quad \beta^{\text{ref}}(k) = \beta^{\text{ref}}(t_s), \quad (5.28)$$

for $k = 0, \dots, N/T_{s,mpc}$.

Unlike the remaining reference states, the yaw-rate reference depends on the vehicle operating point through the steady-state bicycle model and therefore varies with vehicle velocity and steering input. Assuming a constant yaw-rate reference over the prediction horizon may therefore reduce tracking performance during transient manoeuvres. To improve the prediction accuracy, the yaw-rate reference

is propagated over the prediction horizon using a first-order Taylor approximation as:

$$\omega_z^{\text{ref}}(t_s + kT_{s,mpc}) = \omega_z^{\text{ref}}(t_s) + \dot{\omega}_{z,\text{ref}}(t_s)kT_{s,mpc} \quad (5.29)$$

where $\dot{\omega}_z^{\text{ref}}$ is estimated online using a Savitzky–Golay filter [93] applied to previous yaw-rate reference samples. The prediction relies only on current and past measurements and does not require preview information such as future steering inputs or road curvature. The filter order and window length are selected based on the trade-off between noise attenuation and responsiveness, as discussed in [89]. Although the same Taylor-based prediction approach may be extended to additional reference states, only the yaw rate reference is updated along the horizon in Paper E.

The MPC optimisation problem is formulated as

$$\begin{aligned} \mathbf{u}_{mpc}^* = \arg \min_{\mathbf{u}_{mpc}} & \left(\sum_{k=1}^{N/T_{s,mpc}} \mathbf{e}_x^T(k) Q(k) \mathbf{e}_x(k) \right. \\ & \left. + \sum_{k=1}^{N/T_{s,mpc}-1} \mathbf{u}_{mpc}^T(k) R(k) \mathbf{u}_{mpc}(k) \right) \end{aligned} \quad (5.30)$$

subject to

$$\begin{aligned} \mathbf{x}(k+1) &= f_d(\mathbf{x}(k), \mathbf{u}_{mpc}(k)), & \mathbf{x}(0) &= \mathbf{x}_{init} \\ \mathbf{u}_{l,mpc}(k) &\leq \mathbf{u}_{mpc}(k) \leq \mathbf{u}_{u,mpc}(k) \\ \mathbf{r}_{min}(k) &\leq \mathbf{x}(k) \leq \mathbf{r}_{max}(k) \end{aligned}$$

for $k = 1, \dots, N/T_{s,mpc}$, where $f_d(\cdot)$ denotes the discretised prediction model and $Q(k)$ and $R(k)$ are the state and control weighting matrices, respectively.

Recalling that the MPC decision vector is defined in the wheel torque domain as, the torque limits applied are also formulated in the wheel torque domain. The available actuator torque limits are first transformed according to the physical location of each actuator. Electric machine torque limits are initially defined at the motor shaft side and are converted to the wheel side using the drivetrain ratio g_i . However, the friction brake torque limits are already directly defined at the wheel side. Since friction brakes generate braking torque only, the combined actuator torque limits are obtained as:

$$T_{Act,ij}^{\min} = gr_i T_{EM,ij}^{\min} - T_{Brk,ij}^{\lim} \quad (5.31)$$

$$T_{Act,ij}^{\max} = gr_i T_{EM,ij}^{\max} \quad (5.32)$$

Then the admissible wheel torque range is obtained by additionally considering the available tyre force capability:

$$T_{w,il}^{\max} = \min(T_{w,il}^{\lim}, T_{Act,ij}^{\max}) \quad (5.33)$$

$$T_{w,il}^{\min} = \max \left(-T_{w,il}^{\lim}, T_{Act,ij}^{\min} \right) \quad (5.34)$$

The resulting MPC input constraint vectors are therefore defined as:

$$\mathbf{u}_{l,mpc}(k) \leq \mathbf{u}_{mpc} \leq \mathbf{u}_{u,mpc}(k) \quad (5.35)$$

with

$$\mathbf{u}_{u,mpc} = \text{col} \left(T_{w,fL}^{\max}, T_{w,fR}^{\max}, T_{w,rL}^{\max}, T_{w,rR}^{\max} \right) \quad (5.36)$$

$$\mathbf{u}_{l,mpc} = \text{col} \left(T_{w,fL}^{\min}, T_{w,fR}^{\min}, T_{w,rL}^{\min}, T_{w,rR}^{\min} \right) \quad (5.37)$$

The state limits defining vehicle motion constraints are then accumulated to be used as constraints in the optimisation problem using $\mathbf{r}_{min}$ and $\mathbf{r}_{max}$ as:

$$\mathbf{r}_{min} = [a_x^{\min}, v_x^{\min}, \omega_z^{\min}, \beta^{\min}] \quad (5.38)$$

$$\mathbf{r}_{max} = [a_x^{\max}, v_x^{\max}, \omega_z^{\max}, \beta^{\max}] \quad (5.39)$$

The solution of the optimisation problem provides the optimal wheel torque request vector:

$$\mathbf{u}_{mpc}^* = [T_{w,fL}^* \quad T_{w,fR}^* \quad T_{w,rL}^* \quad T_{w,rR}^*]^T$$

The corresponding vehicle-level longitudinal force and yaw moment requests are obtained through,

$$\mathbf{v}_r = \begin{bmatrix} F_x^{\text{req}} \\ M_z^{\text{req}} \end{bmatrix} = \begin{bmatrix} \frac{1}{r_w} & \frac{1}{r_w} & \frac{1}{r_w} & \frac{1}{r_w} \\ -\frac{t_{w,f}}{2r_w} & \frac{t_{w,f}}{2r_w} & -\frac{t_{w,r}}{2r_w} & \frac{t_{w,r}}{2r_w} \end{bmatrix} \mathbf{u}_{mpc}^* \quad (5.40)$$

where $t_{w,f}$ and $t_{w,r}$ denote the front and rear axle track widths, respectively.

The resulting generalised vehicle force request $\mathbf{v}_r$ is supplied to the actuator and axle coordinator introduced in Section 4.2.3, which distributes the requested longitudinal force and yaw moment among the available electric machines and friction brakes while minimising the selected control objective and satisfying the actuator, vehicle, and tyre constraints. The corresponding optimisation problem is formulated as a quadratic programme and solved online, as detailed in Paper E.

5.1.2 Control allocator

The actuator coordinator receives the generalised vehicle force request generated by the global force reference generator. Unlike the actuator and axle coordination formulation introduced in Eq.(4.18), which considers only longitudinal force allocation, the present formulation additionally includes a requested yaw moment. This extension enables the distributed electric powertrain to exploit torque-vectoring capabilities by coordinating individual wheel torques.

The generalised vehicle request is defined as:

$$\mathbf{v}_r = \begin{bmatrix} F_x^{\text{req}} \\ M_z^{\text{req}} \end{bmatrix} \quad (5.41)$$

The actuator coordinator determines the individual actuator torque requests such that the resulting generalised forces satisfy the requested vehicle motion objectives while minimising the selected actuator power losses.

The actuator torque allocation problem is formulated as:

$$\mathbf{u}^* = \arg \min_{\mathbf{u}} \left(\sum_i \sum_j P_{EM,ij}^{\text{loss}} + \sum_i \sum_j P_{Brk,ij}^{\text{loss}} \right) \quad (5.42)$$

$$\text{s.t.} \quad \mathbf{B}\mathbf{u} = \mathbf{v}_r$$

$$\mathbf{u}_l \leq \mathbf{u} \leq \mathbf{u}_u$$

where $\mathbf{u}$ represents the actuator torque request vector. For the vehicle configuration considered in this chapter, the general actuator index j is specialised to the left/right wheel index l . Consequently, the actuator torques are indexed by (i, l) , where i denotes the axle and l denotes the wheel side.:

$$\mathbf{u} = \begin{bmatrix} T_{EM,fL}^{\text{req}} \\ T_{EM,fR}^{\text{req}} \\ T_{EM,rL}^{\text{req}} \\ T_{EM,rR}^{\text{req}} \\ T_{Brk,fL}^{\text{req}} \\ T_{Brk,fR}^{\text{req}} \\ T_{Brk,rL}^{\text{req}} \\ T_{Brk,rR}^{\text{req}} \end{bmatrix} \quad (5.43)$$

The mapping matrix B relates the individual actuator torques to the generalised longitudinal force and yaw moment:

$$B = \frac{1}{r_w} \begin{bmatrix} gr_f & gr_f & gr_r & gr_r & 1 & 1 & 1 & 1 \\ -\frac{gr_f \cdot t_{w,f}}{2} & \frac{gr_f \cdot t_{w,f}}{2} & -\frac{gr_r \cdot t_{w,r}}{2} & \frac{gr_r \cdot t_{w,r}}{2} & -\frac{t_{w,f}}{2} & \frac{t_{w,f}}{2} & -\frac{t_{w,r}}{2} & \frac{t_{w,r}}{2} \end{bmatrix} \quad (5.44)$$

The first row of B represents the contribution of individual wheel torques to the longitudinal vehicle force, whereas the second row represents the generated yaw moment through left–right torque differences.

During control allocation, the requested vehicle-level longitudinal force and yaw moment are distributed among the available electric machines and friction

brakes to minimise the power-loss objective. The resulting actuator torques must satisfy both the actuator limitations and the available tyre-road friction represented by the wheel torque limits derived from the tyre force constraints.

The actuator constraints are constructed using the dynamically feasible torque limits introduced in Chapter 2. For the electric machines, the dynamically achievable torque limits are obtained from Eq.(2.40) and Eq.(2.41), which account for both the speed-dependent torque capability and the finite actuator response. These limits are further combined with the tyre-induced wheel torque limits to define the admissible electric machine torque range:

$$T_{EM,ij}^{CA,max} = \min \left(T_{EM,ij}^{max,dyn}, \frac{T_{w,il}^{lim}}{gr_i} \right) \quad (5.45)$$

$$T_{EM,ij}^{CA,min} = \max \left(T_{EM,ij}^{min,dyn}, -\frac{T_{w,il}^{lim}}{gr_i} \right) \quad (5.46)$$

Similarly, the friction brake torque limits are obtained by combining the dynamically feasible brake actuator limits introduced in Section 2.5 with the physical brake torque capability. The superscript CA denotes the torque limits used within the control allocation problem.

$$T_{Brk,ij}^{CA,max} = \min \left(T_{w,il}^{lim}, T_{Brk,ij}^{max,dyn} \right) \quad (5.47)$$

$$T_{Brk,ij}^{CA,min} = \max \left(-T_{w,il}^{lim}, T_{Brk,ij}^{min,dyn} \right) \quad (5.48)$$

The resulting upper and lower actuator bounds are assembled into the vectors $\mathbf{u}_u$ and $\mathbf{u}_l$, respectively, and enforced through:

$$\mathbf{u}_l \leq \mathbf{u} \leq \mathbf{u}_u$$

The optimisation problem is solved instantaneously at each control update. The resulting quadratic programming formulation and implementation details are provided in Paper E.

5.1.3 Use case - Double lane change test

The performance of the proposed hierarchical wheel torque coordination framework, combining adaptive MPC-based global force generation with a power-loss-aware wheel torque allocator, is evaluated using a double lane change manoeuvre. The vehicle is required to follow a reference path defined by cone positions (red markers) at a constant velocity of 15.28 m/s (55 km/h) under a tyre-road friction coefficient of $\mu = 0.3$. The proposed adaptive MPC configuration is compared with alternative controller configurations using fixed MPC weighting parameters and a PI-based reference tracking controller, as summarised in Table 5.1. The results presented here correspond to the evaluation reported in Paper E.

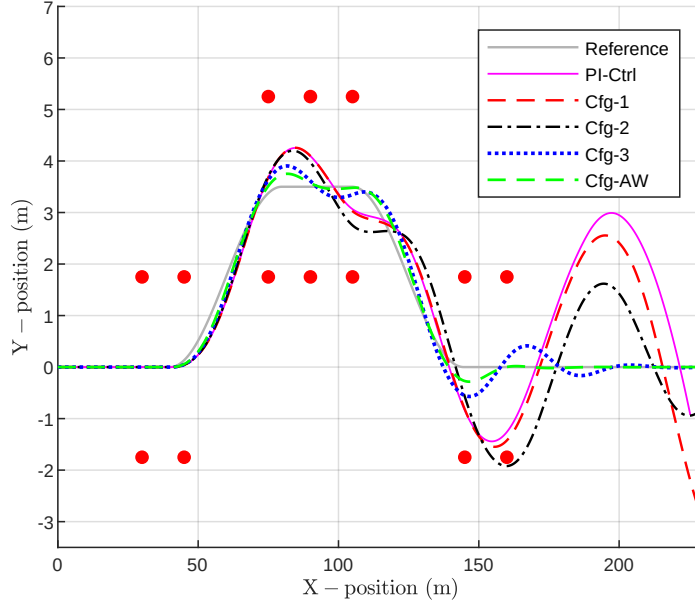


Figure 5.4: Vehicle CoG path profile on the road plane under the DLC test for the different MPC configurations.

Table 5.1: Parameters used for the different LTV-MPC configurations and PI controller

Configuration	Parameters
Cfg-1- Longitudinal motion biased	$Q = \text{diag}[M_v^2 \ 10^{-7} \ 10^{-7} \ 10^{-7}]$ $\mathbf{r}_{max} = [a_x^{\max} \ v_x^{\max} \ 0.7 \ 0.35]$ $\mathbf{r}_{min} = [a_x^{\min} \ v_x^{\min} \ -0.7 \ -0.35]$
Cfg-2 Equal weighting	$Q = \text{diag}[M_v^2 \ 10^{-7} \ M_v^2 \ 10^{-7}]$ $\mathbf{r}_{max} = [a_x^{\max} \ v_x^{\max} \ 0.5 \ 0.25]$ $\mathbf{r}_{min} = [a_x^{\min} \ v_x^{\min} \ -0.5 \ -0.25]$
Cfg-3- Stability oriented	$Q = \text{diag} \left[\frac{M_v^2}{2} \ 10^{-7} \ I_z^2 \ 10^{-7} \right]$ $\mathbf{r}_{max} = [a_x^{\max} \ v_x^{\max} \ 0.3 \ 0.2]$ $\mathbf{r}_{min} = [a_x^{\min} \ v_x^{\min} \ -0.3 \ -0.2]$
Cfg-AW-Adaptive weighting	$Q = \text{diag} (M_v g)^2 \left[\left(\frac{W_{a_x,sc}}{2} \right)^2 \left(\frac{W_{v_x,sc}}{2} \right)^2 \right.$ $\left. \left(\frac{I_z}{M_v g} W_{\omega_z,sc} \right)^2 \left(\frac{W_{\beta,sc}}{g} \right)^2 \right]$ $\mathbf{r}_{max} = [a_x^{\max} \ v_x^{\max} \ \omega_z^{\max} \ \beta^{\max}]$ $\mathbf{r}_{min} = [a_x^{\min} \ v_x^{\min} \ -\omega_z^{\min} \ \beta^{\min}]$
PI-Ctrl-Longitudinal control	$K_P = 4 \cdot M_v, K_I = 150$

The vehicle trajectory results in Figure 5.4 demonstrate the improved path tracking capability of the adaptive MPC configuration. Compared with the fixed-weight configurations, the proposed approach maintains closer tracking of the reference path, particularly during the steering reversal phases of the manoeuvre. This improvement results from the adaptive adjustment of the MPC objectives according to the vehicle operating condition, enabling a more appropriate trade-off between longitudinal tracking and lateral stability requirements.

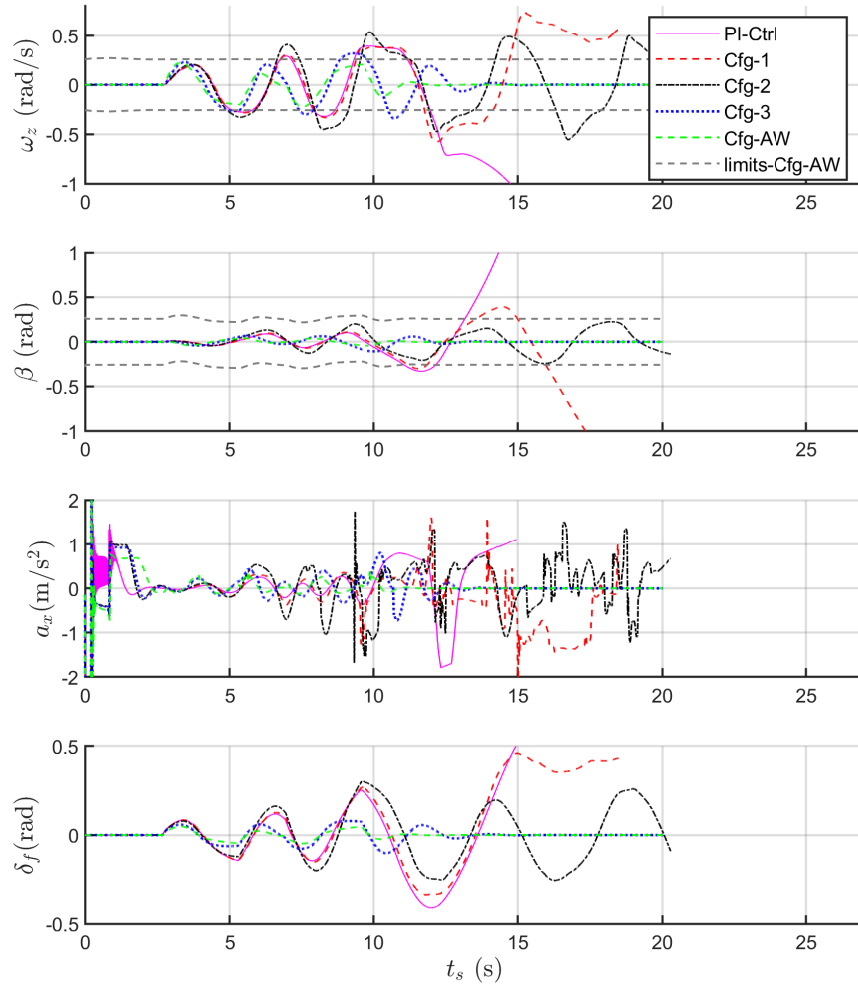


Figure 5.5: Vehicle state responses and front wheel steering inputs obtained from the vehicle plant for the DLC test. The grey dashed lines represent the limits from the Cfg-AW configuration. In the PI-Ctrl and Cfg-1 cases, the vehicle becomes unstable.

The corresponding vehicle state responses in Figure 5.5 further illustrate the effect of the adaptive weighting strategy. The proposed controller achieves reduced

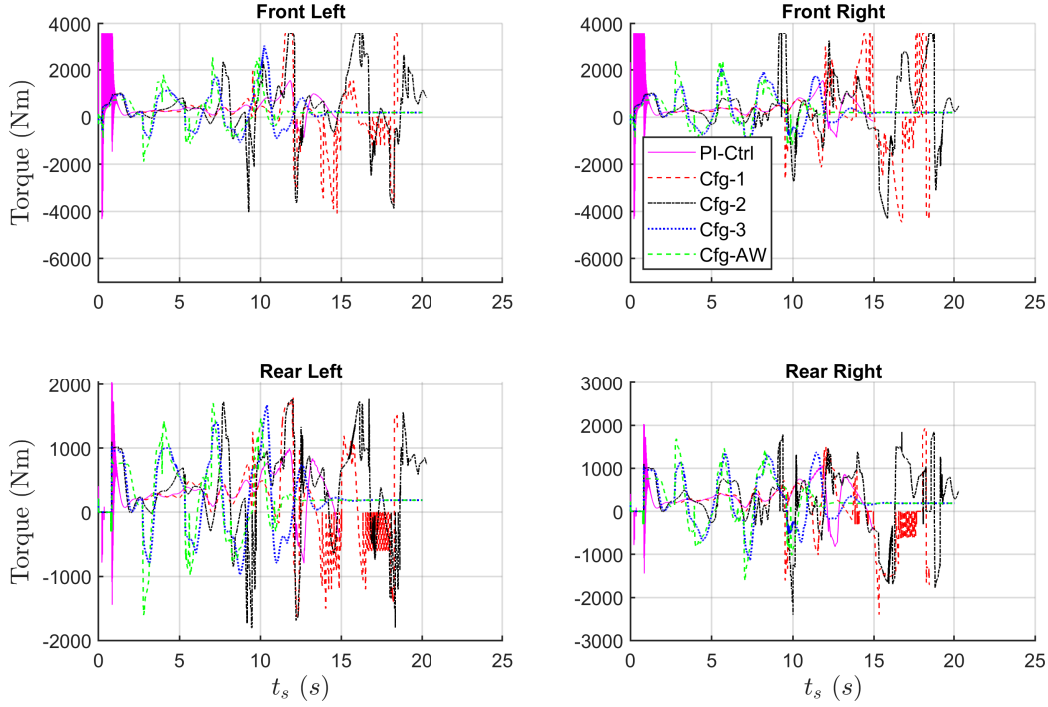


Figure 5.6: Wheel torque requests across all configurations for the DLC use case. In the Cfg-1 case, shown by dotted red lines, ABS activations are observed between 16–18 s on the rear wheels, evident from the dense clustering of torque values.

yaw-rate and side-slip overshoot, together with faster attenuation after transient steering inputs. Consequently, the vehicle operation remains further within the imposed yaw-rate and side-slip constraints, indicating improved handling performance under low-friction conditions.

The wheel torque responses in Figure 5.6 demonstrate the interaction between the global force reference generator and the control allocator. The requested yaw moment generated by the MPC is realised through asymmetric torque distribution between the left and right wheels, while the allocator simultaneously satisfies actuator limitations and tyre force constraints. In contrast to fixed allocation strategies, the adaptive configuration continuously modifies the wheel torque distribution according to the changing vehicle operating conditions, enabling effective use of the available actuators during transient manoeuvres.

Overall, the double lane change results demonstrate that incorporating vehicle-level predictive constraints into the torque coordination framework improves transient handling performance while maintaining compatibility with energy-aware actuator allocation. The hierarchical structure allows the MPC to regulate vehicle motion through future-oriented force requests, while the lower-level allocator

translates these requests into feasible wheel torques considering actuator capabilities and efficiency objectives. This coordinated approach is particularly beneficial for heavy electric vehicles, where large inertia, high centre of gravity, and limited tyre adhesion increase the sensitivity of vehicle behaviour to actuator coordination.

5.2 Yaw-stable actuator and axle coordination under friction circle constraints

The actuator and axle coordination formulation introduced in Section 4.2.3 is extended in this section by modifying the tyre-force constraints used during control allocation. The objective is to increase the achievable longitudinal force under combined braking and cornering conditions while maintaining lateral vehicle stability.

The coordinator in Section 4.2.3, hereafter referred to as **CA-Cons**⁴, determines actuator torque requests by satisfying the predicted lateral tyre force requirements while respecting the available tyre-road friction. The proposed approach, referred to as **CA-Rel**⁵, relaxes the lateral force requirement when necessary, thereby increasing the available longitudinal tyre force capacity without explicitly modifying the vehicle-level force request.

Unlike the adaptive MPC-based approach presented in Section 5.1, **CA-Rel** does not introduce an additional hierarchical control layer. Instead, the enhancement is achieved within the coordination layer by modifying the friction constraints applied to the actuator optimisation problem.

5.2.1 Motivation and formulation of lateral force relaxation

The tyre-force constraint used in the **CA-Cons** is based on the assumption that the predicted lateral force demand must be maintained during the manoeuvre. The available longitudinal tyre force at axle i is therefore limited by:

$$F_{x,i}^{\text{lim}} = \sqrt{(\mu \cdot F_{z,i})^2 - (F_{y,i}^{\text{pred}})^2} \quad (5.49)$$

Equation (5.49) illustrates the inherent trade-off between lateral and longitudinal force generation. During high lateral acceleration manoeuvres, a significant

⁴**Cons** refers to conservative nature of the longitudinal force limit generated $F_{x,i}^{\text{lim}}$

⁵**Rel** refers to relaxation of predicted need of lateral force $F_{y,i}^{\text{pred}}$

portion of the available tyre friction is consumed by lateral force generation, reducing the remaining longitudinal force capability. Consequently, during braking while cornering, the requested braking force may exceed the achievable longitudinal force because the **CA-Cons** constraint assumes that the predicted lateral force demand must be maintained. However, the full predicted lateral force may not be necessary to maintain vehicle stability. Reducing the lateral force demand can therefore free tyre-friction capacity for longitudinal braking while retaining sufficient lateral force for yaw stability.

The proposed **CA-Rel** approach addresses this limitation by allowing a controlled reduction of the lateral force requirement used in the friction constraint. The modified longitudinal force capability is therefore defined as:

$$F_{x,i}^{\text{lim}} = \sqrt{(\mu F_{z,i})^2 - \left(\min \left(|F_{y,i}^{\text{pred}}|, \lambda_{\text{rel},i} \cdot \mu \cdot F_{z,i} \right) \right)^2} \quad (5.50)$$

where $\lambda_{\text{rel},i} \in [0, 1]$ is the lateral force relaxation factor for an axle i . A value of $\lambda_{\text{rel},i} = 1$ corresponds to the friction constraint in Eq.(5.49), whereas lower values permit a larger reduction in lateral force demand and consequently increase the available longitudinal force capacity. It means that **CA-Rel** does not automatically reduce the lateral-force demand. If the predicted lateral force is already below the relaxed limit, the original demand is retained. Relaxation only becomes active when the predicted lateral force exceeds the allowable relaxed value.

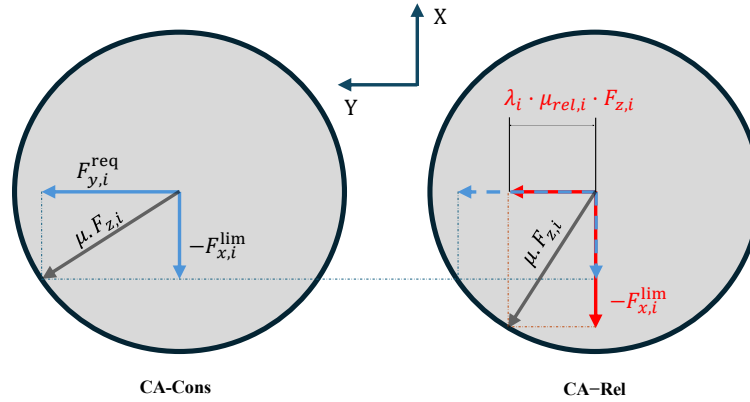


Figure 5.7: Illustration of maximising principle of the longitudinal force request in **CA-Rel** represented using friction circle

As illustrated in Figure 5.7, the relaxed constraint reduces the lateral force contribution considered in the friction-circle formulation, thereby increasing the available longitudinal force capability. The relaxation is activated only when the requested longitudinal force exceeds the maximum achievable longitudinal force

under the **CA-Cons** constraint. Otherwise, **CA-Rel** remains equivalent to **CA-Cons** and the original allocation objective is preserved. In other words, **CA-Rel** becomes active when the requested force approaches the friction limit and the longitudinal force demand can no longer be satisfied while maintaining the full predicted lateral force.

The parameter $\lambda_{rel,i}$ therefore determines the maximum lateral force considered in the relaxed friction constraint relative to the available tyre-friction capacity at axle i . Lower values permit a greater reduction in the lateral force requirement and provide additional capacity for longitudinal force generation. Excessive relaxation may, however, reduce the lateral force available for vehicle stabilisation. The relaxation factor must therefore be selected based on vehicle-dynamic constraints to ensure that sufficient lateral force is retained to satisfy the required yaw-stability conditions.

5.2.2 Yaw-stability-based lateral force relaxation factor

The **CA-Rel** coordinator extends the conservative control allocation strategy, **CA-Cons**, by allowing a controlled relaxation of the prescribed lateral tyre-force demand when the available tyre-road friction is insufficient to simultaneously satisfy the requested longitudinal and lateral forces. The relaxation is introduced only when required to accommodate an increased longitudinal force demand, while maintaining a sufficient lateral-force capability to preserve vehicle yaw stability.

The determination of the relaxation factor is based on a quasi-steady-state single-track vehicle model. The vehicle is assumed to negotiate a curve while undergoing longitudinal braking. Under these conditions, the lateral and yaw dynamics are assumed to be in equilibrium, i.e

$$\dot{v}_y = 0, \quad \dot{\omega}_z = 0, \quad (5.51)$$

while the longitudinal velocity v_x is considered a slowly varying known parameter. Aerodynamic drag and other resistance forces are neglected, and a small steering-angle approximation is applied.

The proposed method consists of two related steps. First, a critical front brake-force fraction is determined from the yaw-stability condition. This establishes the maximum longitudinal braking forces that can be allocated to the front and rear axles without causing the vehicle to cross the considered yaw-stability boundary. Second, the corresponding axle force limits are introduced into the friction-circle constraints to determine the amount of lateral-force relaxation required by **CA-Rel**.

5.2.2.1 Vehicle normal loads during braking

During braking, longitudinal load transfer shifts normal load from the rear axle to the front axle. To avoid sign ambiguity, the positive braking-deceleration magnitude is defined as:

$$a_b = |a_x| > 0 \quad (5.52)$$

The front and rear axle normal loads are then obtained as:

$$F_{z,f} = \frac{M_v g l_r + M_v a_b h}{L} \quad (5.53)$$

$$F_{z,r} = \frac{M_v g l_f - M_v a_b h}{L} \quad (5.54)$$

where h is the centre-of-gravity height and $L = l_f + l_r$ is the wheelbase. Pitch dynamics are neglected and the load transfer is represented algebraically which satisfies

$$F_{z,f} + F_{z,r} = M_v g \quad (5.55)$$

Note that the rear axle normal load must remain positive for the assumed model to be physically meaningful, i.e $F_{z,r} > 0$.

5.2.2.2 Linear tyre model and combined-slip constraint

The lateral tyre forces at the front and rear axles are approximated using a linear tyre model as:

$$F_{y,f} = C_f \alpha_f, \quad F_{y,r} = C_r \alpha_r \quad (5.56)$$

where α_f and α_r are the front and rear axle slip angles, respectively. The axle cornering stiffnesses are assumed to be proportional to the corresponding normal loads given as following:

$$C_f = C_{c,f} F_{z,f} \quad (5.57)$$

and

$$C_r = C_{c,r} F_{z,r} \quad (5.58)$$

where $C_{c,f}$ and $C_{c,r}$ are cornering stiffness coefficients of front and rear axle respectively.

The available tyre force on an axle is constrained by the friction-circle approximation:

$$F_{x,i}^2 + F_{y,i}^2 \leq (\mu F_{z,i})^2, \quad i \in f, r \quad (5.59)$$

Equation (5.59) illustrates the fundamental limitation addressed by **CA-Rel**. When a significant portion of the available friction is consumed by lateral force generation, the remaining longitudinal-force capability is reduced. Consequently,

a coordinator that strictly preserves the predicted lateral-force demand may be unable to satisfy a high braking request even though a small reduction in lateral force would permit a larger longitudinal force exploiting the combined slip properties of the tyre.

5.2.2.3 Critical brake-force distribution

The longitudinal braking force is distributed between the two axles according to the brake-balance ratio:

$$\lambda_b = \frac{|F_{x,f}|}{|F_{x,f}| + |F_{x,r}|}, \quad 0 \leq \lambda_b \leq 1 \quad (5.60)$$

For a total braking demand $M_v a_b$, the corresponding front and rear axle braking forces are:

$$F_{x,f} = \lambda_b M_v a_b \quad (5.61)$$

and

$$F_{x,r} = (1 - \lambda_b) M_v a_b \quad (5.62)$$

The brake balance is important because longitudinal force generation consumes part of the friction capacity that would otherwise be available for lateral force generation. Thus, the brake distribution affects the effective lateral capability of the two axles and consequently the yaw response of the vehicle.

For a prescribed longitudinal force, the friction circle gives the remaining lateral-force capacity as:

$$F_{y,i}^{\text{lim}} = \sqrt{(\mu F_{z,i})^2 - F_{x,i}^2} \quad (5.63)$$

It is convenient to express this effect through a combined-slip reduction factor,

$$\gamma_i = \sqrt{1 - \left(\frac{F_{x,i}}{\mu F_{z,i}} \right)^2} \quad (5.64)$$

The factor γ_i represents the fraction of the pure-lateral tyre capability that remains available after longitudinal force generation. Thus, $\gamma_i = 1$ corresponds to pure lateral force generation, whereas $\gamma_i = 0$ corresponds to complete utilisation of the friction capacity in the longitudinal direction.

Thus the combined-slip reduction factor for the front axle is obtained according to

$$\gamma_f = \sqrt{1 - \left(\frac{\lambda_b M_v a_b}{\mu F_{z,f}} \right)^2}, \quad (5.65)$$

and for the rear axle

$$\gamma_r = \sqrt{1 - \left(\frac{(1 - \lambda_b)M_v a_b}{\mu F_{z,r}} \right)^2} \quad (5.66)$$

The combined-slip effect is represented through effective cornering stiffnesses,

$$C_f^{\text{eff}} = C_f \gamma_f \quad (5.67)$$

and

$$C_r^{\text{eff}} = C_r \gamma_r \quad (5.68)$$

This representation captures the reduction in lateral response caused by longitudinal braking while retaining the linear tyre model used for the yaw-stability analysis.

5.2.2.4 Yaw-stability condition

Under the quasi-steady-state assumptions, the lateral-force and yaw-moment equilibrium equations are given as:

$$F_{y,f} + F_{y,r} = M_v \frac{v_x^2}{R} \quad (5.69)$$

$$l_f F_{y,f} - l_r F_{y,r} = 0 \quad (5.70)$$

For the linear single-track model, the transition between understeering and oversteering is determined by the relative effective cornering stiffness of the front and rear axles. The neutral-steer condition can be expressed as:

$$\frac{l_r}{C_r^{\text{eff}}} = \frac{l_f}{C_f^{\text{eff}}} \quad (5.71)$$

or equivalently,

$$l_r C_r^{\text{eff}} = l_f C_f^{\text{eff}} \quad (5.72)$$

Substituting the effective cornering stiffnesses from Eqs. (5.67)–(5.68) gives:

$$l_r C_r \sqrt{1 - \left(\frac{(1 - \lambda_b)M_v a_b}{\mu F_{z,r}} \right)^2} = l_f C_f \sqrt{1 - \left(\frac{\lambda_b M_v a_b}{\mu F_{z,f}} \right)^2} \quad (5.73)$$

Equation (5.73) defines the brake balance at which the combined-slip vehicle model reaches the neutral-steer boundary.

Squaring both sides gives

$$l_r^2 C_r^2 \left[1 - \frac{(1 - \lambda_b)^2 M_v^2 a_b^2}{\mu^2 F_{z,r}^2} \right] = l_f^2 C_f^2 \left[1 - \frac{\lambda_b^2 M_v^2 a_b^2}{\mu^2 F_{z,f}^2} \right] \quad (5.74)$$

Rearranging Eq. (5.74) gives

$$\frac{l_r^2 C_r^2 M_v^2 a_b^2}{\mu^2 F_{z,r}^2} (1 - \lambda_b)^2 - \frac{l_f^2 C_f^2 M_v^2 a_b^2}{\mu^2 F_{z,f}^2} \lambda_b^2 = l_r^2 C_r^2 - l_f^2 C_f^2 \quad (5.75)$$

For compactness, the following quantities are defined:

$$A = \frac{l_r^2 C_r^2 M_v^2 a_b^2}{\mu^2 F_{z,r}^2}, \quad B = \frac{l_f^2 C_f^2 M_v^2 a_b^2}{\mu^2 F_{z,f}^2}, \quad D_0 = l_r^2 C_r^2 - l_f^2 C_f^2$$

Equation (5.75) can consequently be written as:

$$A(1 - \lambda_b)^2 - B\lambda_b^2 = D_0 \quad (5.76)$$

Expanding the squared term gives:

$$A - 2A\lambda_b + A\lambda_b^2 - B\lambda_b^2 - D_0 = 0 \quad (5.77)$$

Collecting the terms according to the power of λ_b results in

$$(A - B)\lambda_b^2 - 2A\lambda_b + (A - D_0) = 0 \quad (5.78)$$

Defining

$$a_q = A - B, \quad b_q = -2A, \quad c_q = A - D_0 \quad (5.79)$$

the critical brake-balance ratio can be obtained from

$$\lambda_b = \frac{-b_q \pm \sqrt{b_q^2 - 4a_q c_q}}{2a_q} \quad (5.80)$$

A real-valued solution requires $b_q^2 - 4a_q c_q \geq 0$ and in addition the solution must satisfy $0 \leq \lambda_b \leq 1$. Furthermore, the corresponding front and rear braking forces must individually remain within their friction capacities,

$$\lambda_b M_v a_b \leq \mu F_{z,f} \quad (5.81)$$

$$(1 - \lambda_b) M_v a_b \leq \mu F_{z,r} \quad (5.82)$$

These additional conditions are important because Eq. (5.78) is obtained after squaring Eq. (5.73). Consequently, mathematical roots must be checked against the original physical constraints before being considered feasible.

The critical brake-balance ratio is therefore selected from the physically feasible roots of Eq. (5.80). If no physically valid solution exists, the assumed quasi-steady operating condition cannot be represented by a feasible brake distribution satisfying the selected yaw-stability criterion. In the implementation, a value based on equal friction utilisation is then used to maintain numerical robustness.

5.2.2.5 From critical brake balance to axle longitudinal-force limits

The critical brake-balance ratio determines the longitudinal braking force that can be allocated to each axle. The corresponding axle-level force limits are:

$$F_{x,f}^{\text{lim}} = \lambda_b M_v a_b \quad (5.83)$$

$$F_{x,r}^{\text{lim}} = (1 - \lambda_b) M_v a_b \quad (5.84)$$

These limits establish the connection between the yaw-stability analysis and the control-allocation problem. They specify how the available longitudinal braking capability should be distributed between the front and rear axles when the vehicle operates close to the yaw stability boundary.

The limits in Eqs. (5.83)–(5.84) are not necessarily active during normal operation. In **CA-Rel**, the additional stability-based constraint is activated only when the longitudinal force requested by the driver exceeds the force capability obtained while retaining the conservative lateral-force requirement.

5.2.2.6 Determination of the lateral-force relaxation factor

The critical brake balance defines the longitudinal force that can be accommodated on each axle. The corresponding remaining lateral-force capability follows directly from the friction-circle constraint. For the front axle

$$F_{y,f}^{\text{avl}} = \sqrt{(\mu F_{z,f})^2 - (F_{x,f}^{\text{lim}})^2} \quad (5.85)$$

and for the rear axle,

$$F_{y,r}^{\text{avl}} = \sqrt{(\mu F_{z,r})^2 - (F_{x,r}^{\text{lim}})^2} \quad (5.86)$$

The axle-specific lateral-force relaxation factors can then be defined as:

$$\lambda_{\text{rel},f} = \frac{F_{y,f}^{\text{avl}}}{|F_{y,f}^{\text{pred}}|} \quad (5.87)$$

and

$$\lambda_{\text{rel},r} = \frac{F_{y,r}^{\text{avail}}}{|F_{y,r}^{\text{pred}}|} \quad (5.88)$$

For the implementation, the relaxation factors are bounded according to $0 \leq \lambda_{\text{rel},i} \leq 1$. A value of $\lambda_{\text{rel},i} = 1$ means that the complete predicted lateral-force demand can be retained. Conversely, $0 < \lambda_{\text{rel},i} < 1$ indicates that part of the predicted lateral-force demand must be relaxed to accommodate the required longitudinal force while remaining within the available tyre friction.

The relaxed lateral-force demand supplied to the control allocator is consequently:

$$F_{y,i}^{\text{rel}} = \lambda_{\text{rel},i} F_{y,i}^{\text{pred}} \quad (5.89)$$

Substitution into the friction-circle constraint gives

$$F_{x,i}^2 + \left(\lambda_{\text{rel},i} F_{y,i}^{\text{pred}} \right)^2 \leq (\mu F_{z,i})^2 \quad (5.90)$$

The corresponding longitudinal-force limit is therefore:

$$F_{x,i}^{\text{lim}} = \sqrt{(\mu F_{z,i})^2 - \left(\lambda_{\text{rel},i} F_{y,i}^{\text{pred}} \right)^2} \quad (5.91)$$

For $\lambda_{\text{rel},i} = 1$, Eq. (5.91) reduces to the conservative **CA-Cons** friction-circle limit, where as for $0 < \lambda_{\text{rel},i} < 1$ the longitudinal-force capability increases, provided that $F_{y,i,\text{pred}} \neq 0$. Thus, the relaxation creates additional longitudinal-force capacity by deliberately reallocating part of the available tyre friction from lateral to longitudinal force generation.

5.2.3 Evaluation of yaw-stability-based lateral force relaxation factor under combined braking and cornering

The performance of the proposed **CA-Rel** coordinator is evaluated in an emergency braking-in-curve manoeuvre and compared with the conservative **CA-Cons** coordinator. The test is performed using the 4×4 electric tractor configuration introduced in Paper D, equipped with one electric machine per axle and independently controlled friction brakes with ABS at each wheel. The vehicle negotiates a constant-radius curve, with radius $\approx 120\text{m}$, at a speed of 70 km/h with a tyre-road friction coefficient of $\mu = 0.4$. A maximum braking request is applied by the driver model approximately 6 s after the start of the manoeuvre.

To preserve the power-loss minimisation objective during normal driving conditions, the relaxation mechanism in **CA-Rel** is activated only when the requested longitudinal force exceeds the capability provided by the conservative friction-circle constraint. Thus, under non-critical conditions, **CA-Rel** converges to the same allocation behaviour as **CA-Cons**.

The resulting vehicle trajectories are presented in Figure 5.8. **CA-Rel** reduces the stopping distance by approximately 11 m compared with **CA-Cons**, while maintaining a lateral deviation below 0.6 m . During the initial cornering phase, both coordinators exhibit similar path-tracking behaviour. Following the activation of the braking request, **CA-Rel** achieves a higher deceleration by exploiting the increased longitudinal tyre force capability. This improvement requires additional steering compensation, as shown in Figure 5.9, where the driver model applies a larger steering correction to maintain the desired path after the reduction of lateral tyre force demand.

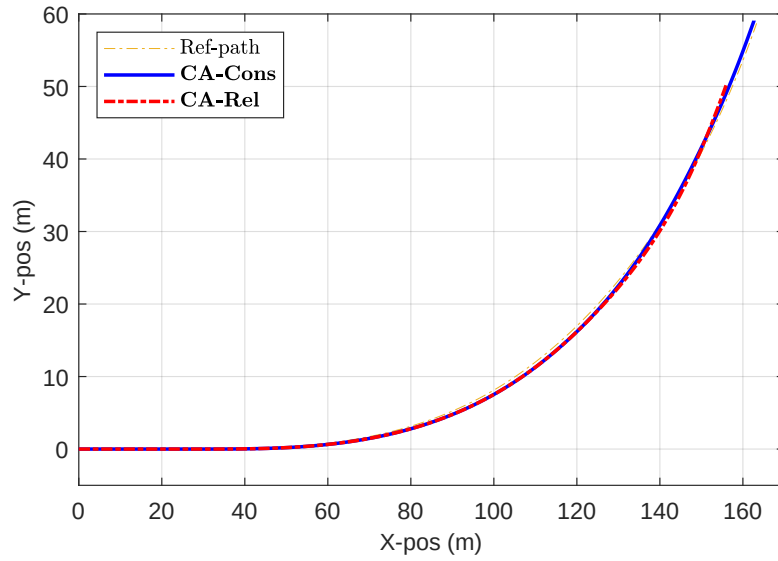


Figure 5.8: Path followed by vehicle during the brake in curve manoeuvre. The label 'Ref-path' in the legend shows the expected path to be followed during the manoeuvre.

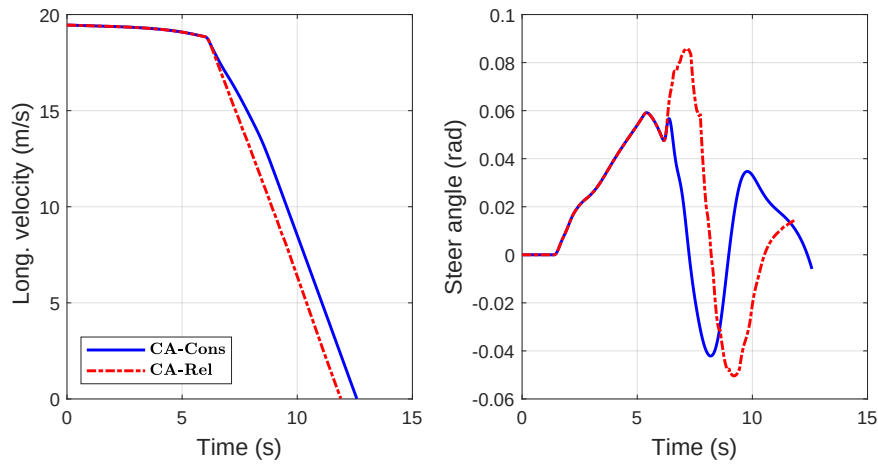


Figure 5.9: Vehicle speed and steering angle responses achieved for the two coordinators.

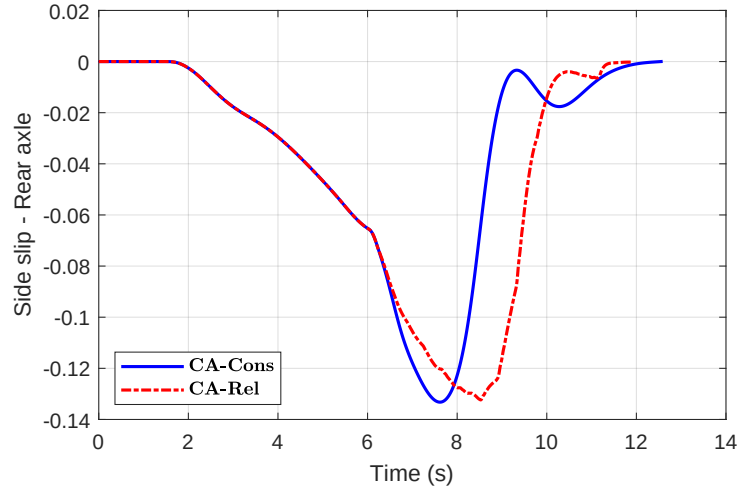


Figure 5.10: Side slip generated on the rear axle during the braking in curve manoeuvre.

Figure 5.10 shows the corresponding rear-axle side-slip angle for both controllers. In both cases, the vehicle exhibits a pronounced oversteering tendency during the combined braking and cornering manoeuvre, resulting in large rear-axle side-slip angles. This operating condition is precisely where the proposed lateral-force relaxation becomes active, as the rear axle approaches its friction limit. Consequently, the relaxation factor modifies the longitudinal force limits, allowing additional braking force while maintaining yaw stability.

The differences between the two coordinators are further illustrated through the wheel-force distribution in Figures 5.11 and 5.12. The largest variations occur between 6 s and 9 s, where **CA-Rel** operates closer to the available tyre friction limit and generates higher longitudinal tyre forces compared with **CA-Cons**. The increase in longitudinal force is accompanied by a reduction in lateral force utilisation, demonstrating the exploitation of the combined-slip characteristics of the tyres. The small oscillations observed in the longitudinal tyre forces originate from the wheel-slip controller operating close to the imposed slip limit.

As the vehicle speed decreases, the lateral acceleration and corresponding lateral force demand reduce. Consequently, the relaxation mechanism becomes inactive and the longitudinal force capabilities obtained from the conservative and relaxed friction-circle constraints become equivalent. The allocator therefore returns to the power-loss minimisation behaviour without additional lateral-force relaxation.

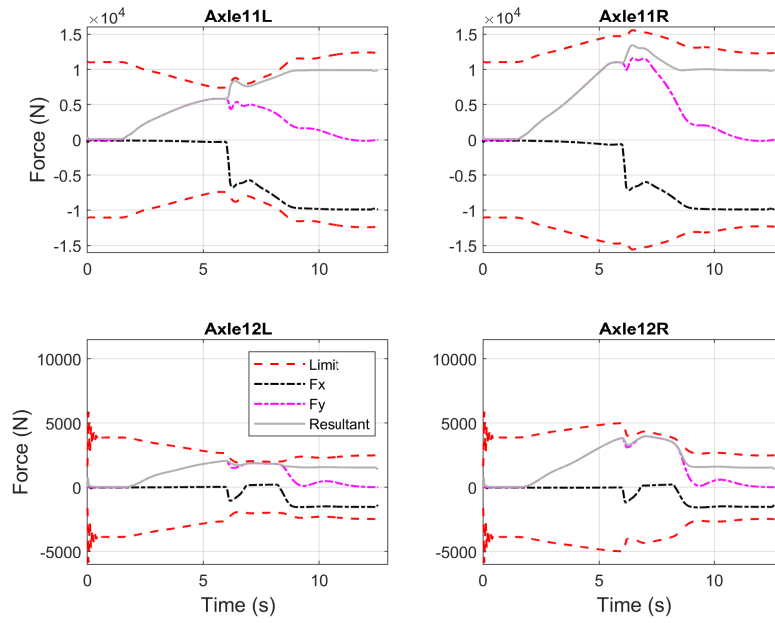


Figure 5.11: Wheel forces and their limits for CA-Cons

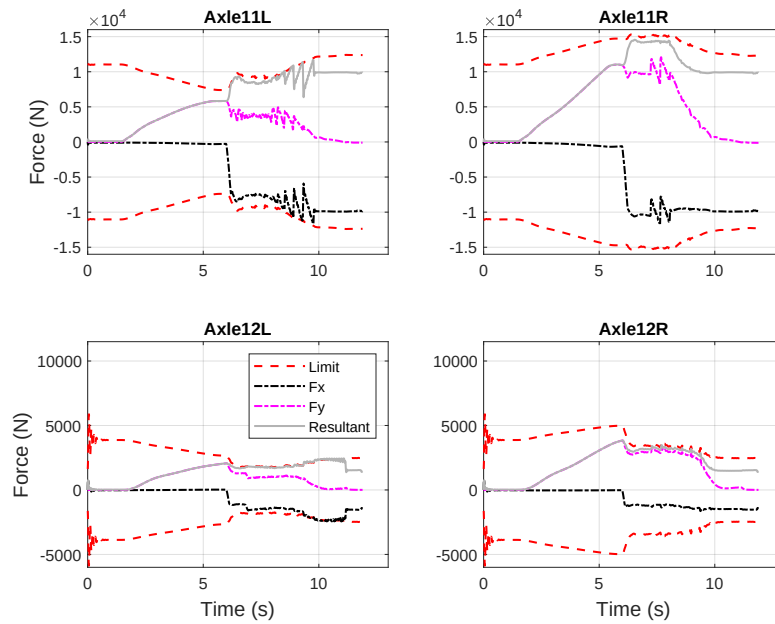


Figure 5.12: Wheel forces and their limits for CA-Rel

The resulting axle-level longitudinal force capabilities are compared in Figure 5.13. During the critical braking phase, **CA-Rel** provides increased longitudinal force capability at both axles by activating the relaxation strategy, which enables higher braking forces while remaining within the available tyre friction limits. After the critical phase, the capability limits converge with those of **CA-Cons** due to the reduced lateral force demand.

These results highlight the performance of enhanced control allocator for brake in curve manoeuvre. By quickly adapting the constraints and utilising the combined slip characteristics, it is possible to improve the braking performance without affecting vehicle stability. This enhanced coordinator can then switch to the mode minimising power loss objectives under normal driving conditions proving the applicability to be used with cruise and startability axles.

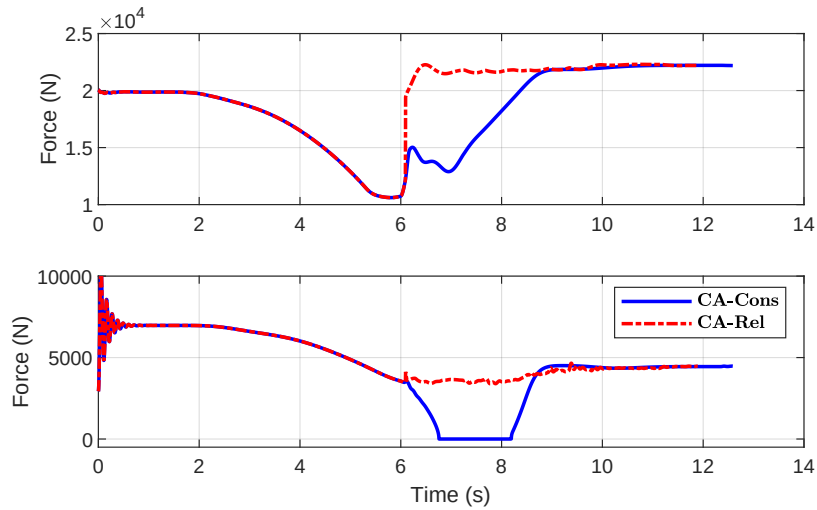


Figure 5.13: Longitudinal force limits $F_{x,i}^{\text{lim}}$ calculated for actuator requests on front and rear axles for **CA-Cons** and **CA-Rel**.

Overall, the results demonstrate that **CA-Rel** improves braking performance during combined braking and cornering manoeuvres by exploiting the available tyre friction more effectively, while retaining the original power-loss optimisation objective during normal operating conditions.

5.3 Comparison and conclusions

The two approaches presented in this chapter address vehicle motion limitations under tyre-force constraints at different levels of the control architecture. The

hierarchical MPC-based framework presented in Section 5.1 provides a predictive vehicle-level motion control solution, where longitudinal force and yaw-moment requests are generated while considering vehicle dynamics, actuator limitations, and motion constraints. The adaptive weighting and state constraints improve path tracking performance, lateral stability margins, and control robustness during demanding low-friction manoeuvres.

In contrast, the approach presented in Section 5.2 enhances the actuator coordination layer by modifying the friction-related constraints used for actuator coordination. By allowing a controlled relaxation of the lateral force demand, the **CA-Rel** coordinator increases the available longitudinal force capability during combined braking and cornering manoeuvres. The results demonstrate improved braking performance while maintaining acceptable vehicle lateral behaviour.

The two approaches are therefore complementary rather than alternatives. The MPC-based framework provides a general solution for predictive motion regulation over a broad range of driving conditions, whereas **CA-Rel** provides a targeted improvement to the actuator coordination strategy for situations where longitudinal force demand is limited by lateral tyre force utilisation. The results demonstrate that safe and effective motion control of heavy electric vehicles can be achieved through both vehicle-level predictive control and constraint-aware actuator coordination, with the former providing a broader control framework and the latter improving performance in specific limit-handling scenarios.

Chapter 6

Tyre-wear-aware actuator and axle coordination

The actuator and axle coordination framework developed in this thesis enables efficient utilisation of the redundant actuation available in distributed electric propulsion (DEP) systems. By exploiting the additional degrees of freedom provided by multiple electric machines and friction brakes, the actuator coordinator distributes the requested longitudinal force among available actuators while minimising instantaneous actuator power losses. The resulting power-loss-minimising formulation provides an energy-efficient solution while satisfying actuator constraints and vehicle motion requirements.

However, energy consumption alone does not fully represent the operational cost of heavy electric vehicles. In addition to energy usage, tyre degradation represents a significant contribution to vehicle operating costs and has become an increasing environmental concern. This is particularly relevant for BEVs, where higher vehicle mass and increased drivetrain torque capability can result in increased tyre loading and accelerated tread degradation.

As exhaust emissions are progressively reduced through vehicle electrification, non-exhaust emissions associated with tyre wear, brake wear and road interaction represent an increasing fraction of traffic-related particulate emissions. Among these sources, tyre wear particles contribute significantly to microplastic contamination and have therefore attracted increasing attention from both environmental and regulatory perspectives [58, 2, 73]. Future regulations targeting tyre degradation further emphasise the need for strategies capable of reducing tyre wear during vehicle operation.

From an economic perspective, tyre replacement represents a considerable operational expense for HCVs. The effect is expected to become increasingly important for electrified heavy vehicles due to their higher torque capability and increased vehicle mass. Since tyre degradation is directly influenced by tyre-road

interaction, vehicle motion control and actuator coordination provide opportunities to reduce unnecessary tyre excitation.

The power-loss-minimisation formulation provides a suitable foundation for incorporating additional operational objectives. Since power loss represents an energy dissipation rate, it can be converted into an equivalent economic cost rate. This enables the optimisation objective to be extended beyond energy efficiency by incorporating other cost contributions within the same physically interpretable framework.

In this chapter, tyre wear is introduced as an additional cost-rate component within the actuator and axle coordination framework. By expressing both actuator power losses and tyre degradation in monetary units, a unified cost-rate optimisation strategy is obtained. Compared with conventional multi-objective optimisation approaches based on arbitrary weighting factors, the proposed formulation enables physically interpretable trade-offs between short-term energy consumption and long-term component degradation.

The resulting tyre-wear-aware actuator coordination strategy extends the energy-efficient framework developed in Chapter 4 towards a more comprehensive operational cost optimisation approach for distributed electric propulsion systems.

6.1 Generalised cost-rate formulation

The actuator and axle coordination problem introduced in Section (4.1.2.2) can be interpreted as a specific case of a more general cost-rate optimisation framework. In the original formulation, the objective function consists of actuator power losses associated with the electric machines and friction brakes. These losses represent irreversible energy conversion processes and are therefore expressed as rates of energy dissipation:

$$P^{\text{loss}} = \frac{dE}{dt}$$

Since power loss is expressed in watts (J/s), it can be directly converted into an economic cost rate when the energy price is known. Consequently, the power-loss minimisation problem can be interpreted as an economic optimisation problem in which actuator usage is evaluated according to its associated operational cost.

The original objective function introduced in Chapter 4 can therefore be generalised as:

$$\sum P_j^{\text{loss}}(u) \rightarrow \sum C_j^{\text{rate}}(u) \quad (6.1)$$

where C_j^{rate} represents actuator-related cost-rate component associated with actuator j expressed in a common economic unit.

This generalisation preserves the mathematical structure of the original actuator coordination problem. The optimisation variables, equality constraints,

actuator limits and vehicle-level constraints remain unchanged. Only the objective function is modified to incorporate additional operational cost components.

Under this formulation, power-loss minimisation represents one possible objective composition, where

$$C_{rate} = C_{power}$$

is the only considered cost. However, additional cost terms can be incorporated provided that they can be expressed as rates in a common unit.

The advantage of this formulation is that different operational effects can be combined without introducing artificial weighting factors. Instead, each objective contribution represents a physically measurable cost associated with vehicle operation. This improves the interpretability of the optimisation problem and enables systematic evaluation of trade-offs between different operational objectives.

In this chapter, tyre wear is introduced as an additional cost-rate component. Unlike actuator power losses, which represent instantaneous energy dissipation, tyre wear represents a cumulative degradation process associated with tyre-road interaction. Nevertheless, through conversion from tread degradation to economic cost, tyre wear can be expressed as an equivalent cost rate and incorporated within the same optimisation framework.

Consequently, the actuator and axle coordination problem developed in Chapter 4 is extended from power-loss minimisation towards unified cost-rate minimisation, where both energy consumption and tyre degradation are considered.

6.2 Tyre-wear cost-rate formulation

6.2.1 Slip-based tyre wear model

Tyre wear is introduced as an additional operational cost associated with longitudinal force generation. The formulation adopted in this thesis is based on an experimentally calibrated slip-dependent tyre wear model presented in Paper F, where tread material loss is related to the accumulated longitudinal and lateral slip during vehicle operation.

The general tyre wear model expresses the accumulated tyre mass loss in kg as:

$$M_{tr}^{wear} = w_{tr} K_{tr} \int_{s_0}^{s_f} (s_x^2(s) + s_y^2(s)) ds, \quad (6.2)$$

where s denotes the distance travelled, s_x and s_y represent longitudinal and lateral slip, w_{tr} is the tyre width, and K_{tr} is an experimentally calibrated wear coefficient. The use of travelled distance s as the independent variable ensures that wear is directly related to the physical interaction between tyre and road surface along the travelled path.

The complete tyre wear mechanism depends on several factors, including normal load, temperature, tyre compound, pressure and operating conditions. However, for actuator coordination purposes, the objective is not to predict absolute tyre lifetime but rather to estimate the relative wear cost associated with different actuator force distributions.

For the manoeuvres and operating conditions considered in this thesis, lateral slip is expected to remain small relative to longitudinal slip, and its contribution to tyre wear is therefore neglected. The wear formulation is accordingly simplified to consider only longitudinal slip:

$$M_{tr,x}^{\text{wear}} = w_{tr} K_{tr} \int_{s_0}^{s_f} s_x^2(s) ds \quad (6.3)$$

This approximation is suitable for the longitudinal actuator coordination problem considered in this thesis, where the optimisation variable is the distribution of longitudinal tyre forces.

6.2.2 Torque-based tyre wear formulation

For small longitudinal slip values, the tyre force relationship can be approximated using the linear tyre model as,

$$F_x = C_x s_x \quad (6.4)$$

where C_x denotes the longitudinal tyre stiffness.

Consequently, by neglecting wheel rotational inertia as in single wheel model presented in Chapter 2, the wheel longitudinal force can be related to the applied wheel torque as:

$$F_x = \frac{T_w}{r_w} \quad (6.5)$$

Substituting Eq.(6.5) into the tyre wear model yields the torque-based representation x

$$M_{tr,x}^{\text{wear}} = w_{tr} K_{tr} \int_{s_0}^{s_f} \left(\frac{T_w}{r_w \cdot C_x} \right)^2 ds \quad (6.6)$$

where $M_{tr,x}^{\text{wear}}$ denotes the tyre mass loss caused by longitudinal slip.

This formulation establishes a direct relationship between actuator torque distribution and tyre degradation. Consequently, tyre wear can be evaluated directly within the actuator coordination layer without requiring additional tyre state estimation.

6.2.3 Tyre-wear cost rate formulation

The accumulated tyre mass loss is converted into tread depth reduction using the tyre geometry and material density as:

$$\Delta h_{tr,x} = \frac{M_{tr,x}^{\text{wear}}}{\pi w_{tr} d_{tr} D_{tr}} \quad (6.7)$$

where d_{tr} is the tyre diameter and D_{tr} is the rubber density.

The tread depth reduction is subsequently converted into an economic cost using the tyre price and usable tread depth,

$$C_{tr}^{\text{wear}} = p_{tr} \cdot \frac{1}{h_{tr}} \frac{\Delta h_{tr,x}}{ds} \quad (6.8)$$

where p_{tr} is the tyre price and h_{tr} denotes the usable tread depth.

For an axle i and side l , consisting of $n_{w,l}$ tyres, the tyre wear cost rate is expressed as:

$$C_{tr,il}^{\text{wear}} = p_{tr} \frac{1}{h_{tr,il}} \frac{d\Delta h_{tr,x,il}}{ds} n_{w,l} \quad (6.9)$$

where $h_{tr,il}$ is the usable tread depth of each tyre on axle side l .

The resulting cost rate represents the monetary cost associated with tyre degradation during vehicle operation (in €/m). Since it is expressed in the economic unit, as the actuator power-loss according to Eq.(6.1), it can be directly incorporated into the generalised cost-rate optimisation framework.

An important consequence of this formulation is that tyre wear becomes directly dependent on the actuator force distribution. Consequently, the optimisation can reduce tyre degradation by avoiding unnecessary concentration of longitudinal force on individual axles or wheels and by exploiting the actuation redundancy available in distributed electric propulsion systems.

6.3 Cost-rate-aware actuator and axle coordination

The actuator and axle coordination framework introduced in Chapter 4 was formulated as a power-loss minimisation problem, where the redundant actuation capability of the distributed electric propulsion architecture was exploited to distribute the requested longitudinal force among the available actuators. The optimisation objective considered the instantaneous power losses associated with the electric machines and friction brakes, while satisfying actuator constraints, force request fulfilment, and axle-level tyre force limitations.

Although power-loss minimisation provides an effective strategy for improving energy efficiency, it does not account for other operational costs associated with vehicle usage. In particular, tyre degradation represents a significant lifetime-related cost for heavy electric vehicles and is directly influenced by actuator force distribution. Therefore, the optimisation framework is extended by incorporating tyre wear as an additional cost component introduced earlier in this chapter.

To enable a unified optimisation objective, both actuator power losses and tyre wear are expressed as monetary cost rates in €/s. The power losses represent the rate of irreversible energy dissipation and can therefore be converted into an equivalent economic cost rate using the electricity price:

$$C_{pl,ij} = (P_{EM,ij}^{\text{loss}} + P_{Brk,ij}^{\text{loss}}) \frac{C_{\text{el}}}{3.6 \times 10^6}, \quad (6.10)$$

where C_{el} is the electricity price in €/kWh, and the factor 3.6×10^6 converts joules into kilowatt-hours. The resulting quantity represents the monetary cost associated with actuator power dissipation per unit time.

Similarly, the tyre wear model developed in Section 6.2 provides a relationship between longitudinal force generation and tread degradation. The resulting tread depth reduction rate is converted into a tyre wear cost rate in €/s using the tyre price and usable tread depth:

$$C_{tr,il}^{\text{wear}} = p_{tr} \frac{1}{h_{tr,il}} \frac{d\Delta h_{tr,x,il}}{ds} n_{w,l} v_x \quad (6.11)$$

where p_t is the tyre price, $h_{tr,il}$ is the usable tread depth, $n_{w,l}$ is the number of tyres on axle side l , and v_x converts the wear cost from €/m into €/s. Consequently, both energy losses and tyre degradation are represented in the same unit, enabling a physically meaningful trade-off within the optimisation problem.

The actuator coordination problem is therefore reformulated as an economic cost-rate minimisation problem:

$$\begin{aligned} \mathbf{u}^* = \arg \min_{\mathbf{u}} \quad & \sum_i^n \sum_j^{m_i} C_{pl,ij} + \sum_i^n \sum_l C_{tr,il}^{\text{wear}} \\ \text{s.t.} \quad & B\mathbf{u} = \mathbf{v}, \\ & -\mathbf{h} \leq G\mathbf{u} \leq \mathbf{h}, \\ & \mathbf{u}_l \leq \mathbf{u} \leq \mathbf{u}_u. \end{aligned} \quad (6.12)$$

where $\mathbf{u}$ is the actuator force request vector, $\mathbf{v}$ is the requested longitudinal vehicle force, and the constraints retain the same formulation as introduced in Chapter 4. The equality constraint ensures that the allocated actuator forces satisfy the vehicle motion request, while the inequality constraints enforce axle-level tyre force limitations and actuator capability boundaries.

The actuator decision vector is defined as

$$\mathbf{u} = [F_{x,EM,11}^{\text{req}}, \dots, F_{x,EM,nM_{EM,i}}^{\text{req}}, F_{x,Brk,11}^{\text{req}}, \dots, F_{x,Brk,n,M_{Brk,i}}^{\text{req}}]^T, \quad (6.13)$$

where the elements represent the longitudinal force requests allocated to the electric machines and friction brakes. The actuator constraints are defined by the corresponding minimum and maximum achievable forces:

$$\mathbf{u}_l \leq \mathbf{u} \leq \mathbf{u}_u. \quad (6.14)$$

The axle force limitations are represented through the constraint matrix G :

$$-\mathbf{h} \leq G\mathbf{u} \leq \mathbf{h}, \quad (6.15)$$

where $\mathbf{h}$ contains the available longitudinal force capability of each driven axle. The formulation of G and $\mathbf{h}$ depends on the vehicle configuration and powertrain topology.

Compared with the original power-loss minimisation formulation, the proposed economic cost-rate formulation introduces an additional incentive to distribute longitudinal forces according to both actuator efficiency and tyre degradation. Consequently, the allocator can exploit the redundant actuation capability of the DEP system to reduce excessive tyre slip while maintaining efficient actuator utilisation.

The proposed formulation preserves the computational structure of the original actuator coordination framework. Under the adopted linear tyre slip assumption, the tyre wear contribution is quadratic with respect to longitudinal force and therefore maintains the convexity of the optimisation problem. Hence, tyre wear can be incorporated without significantly increasing the computational complexity.

Overall, the proposed economic cost-rate formulation extends the energy-efficient actuator coordination framework towards a more comprehensive operational cost optimisation strategy. By representing energy losses and tyre degradation in a common monetary unit, the coordinator can systematically balance short-term energy consumption against long-term component degradation. Furthermore, the formulation provides a scalable framework in which additional cost components, such as actuator ageing, maintenance requirements, or idle losses, can be incorporated within the same optimisation structure.

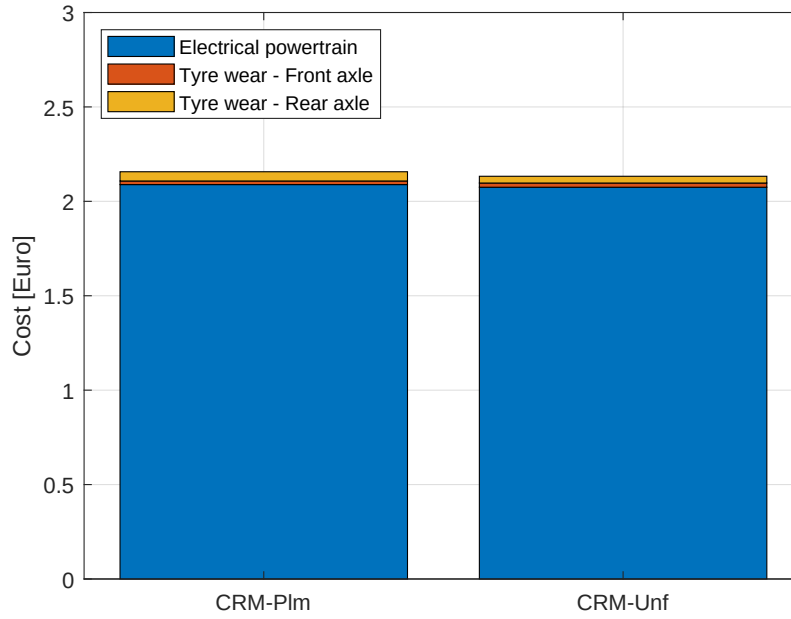


Figure 6.1: Comparison of the total operating cost and its constituent cost components for CRM-Unf and CRM-Plm.

6.3.1 Evaluation of tyre wear-aware cost-rate optimisation in actuator and axle coordination

Example: Operating cost for refuse truck application using a 2-axle heavy vehicle

This section evaluates the proposed tyre-wear-aware cost-rate minimisation **CRM-Unf** strategy by comparing it with the power-loss-minimising (PLM) coordinator **CRM-Plm** introduced in Chapter 4. The evaluation is performed for a two-axle distributed electric vehicle with cruise and startability axles using PMSMs on both the axles and operating in a refuse driving cycle, where frequent acceleration and deceleration events result in high variations in longitudinal tyre slip. Both coordination strategies are evaluated under identical vehicle configurations and operating conditions to ensure a consistent comparison.

The results, summarised in Figure 6.1 and Table 6.1, demonstrate that incorporating tyre wear into the optimisation objective reduces the total operating cost by 1.12% and the tyre wear cost by 14.43% compared with the **CRM-Plm** strategy, while maintaining comparable vehicle performance. The improvement is achieved by exploiting the actuation redundancy of the distributed electric pow-

Table 6.1: Cost metrics associated with the two coordination algorithms

Actuator coordina- tor	Energy cost	Tyre wear cost-Front axle	Tyre wear cost-Rear axle
CRM-Plm	2.0888	0.0190	0.0489
CRM-Unf	2.0746	0.0223	0.0358

Unit of cost is expressed in €.

ertrain to redistribute longitudinal force more evenly between the driven axles, thereby reducing excessive tyre slip without compromising longitudinal motion tracking.

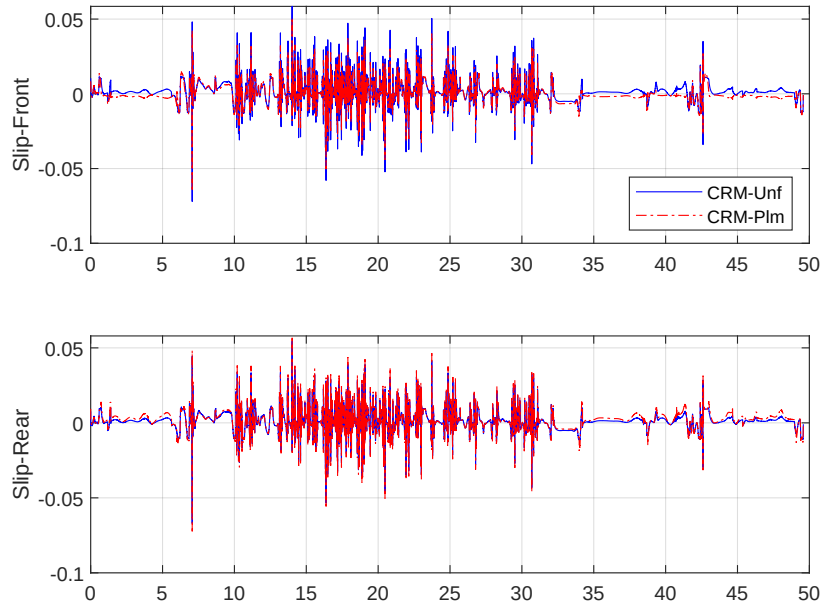


Figure 6.2: Comparison of the longitudinal slip on the front and rear driven axles for the two coordination strategies. Slip utilisation on the front axle with **CRM-Unf** is higher than with **CRM-Plm** and lower on the rear axle. The axle slip is computed as the sum of the longitudinal slips of the left and right wheels.

The actuator allocation obtained with the proposed strategy differs from the PLM solution by balancing actuator power losses and tyre degradation, rather than minimising energy losses alone. Consequently, the propulsion forces are distributed more uniformly across the available driven axles, reducing tyre wear while maintaining a comparable energy cost. This behaviour is illustrated in Figure 6.2, where the proposed **CRM-Unf** achieves a more equal longitudinal slip distribution between the driven axles than **CRM-Plm**, thereby avoiding

excessive utilisation of a single axle for energy efficiency. The results also indicate that minimising instantaneous operating cost does not necessarily correspond to equal tyre slip utilisation across individual axles.

Furthermore, the sensitivity studies presented for the vehicle and powertrain configurations in Paper G show that the effectiveness of the proposed formulation is maintained under small variations in electricity price and tyre price, with only minor changes in the achieved operating-cost reduction. However, replacing the rear axle electric machine with IM is seen to be sensitive to electricity price.

Overall, the results demonstrate that extending the actuator coordination problem from power-loss minimisation to unified cost-rate optimisation enables economically meaningful trade-offs between energy consumption and tyre degradation, thereby reducing the overall operating cost of distributed electric propulsion systems while preserving vehicle performance. Future extensions to the objectives may therefore incorporate tyre condition estimation, replacement intervals, or axle-specific maintenance costs to account for lifetime tyre management.

6.3.2 Operating cost for a 6x4 tractor for a regional distribution driving cycle

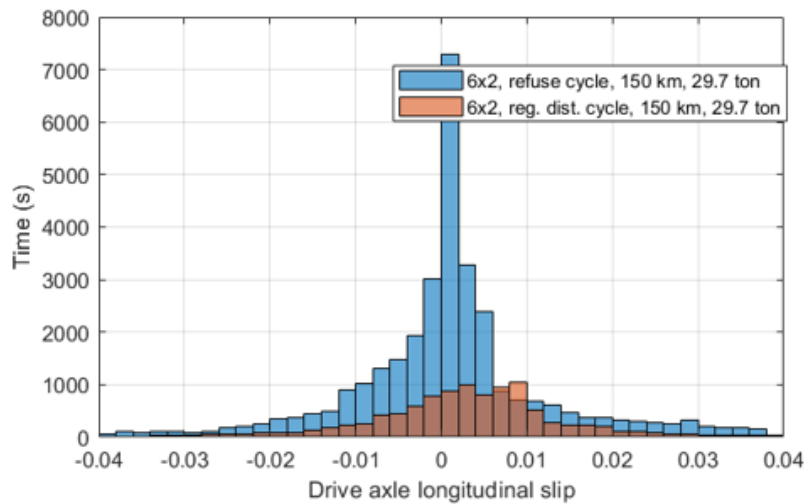


Figure 6.3: Refuse and regional distribution driving cycles for the single-driven-axle configuration.

The applicability of the proposed tyre-wear-aware cost-rate optimisation framework is further evaluated for a three-axle distributed electric tractor operating in a regional distribution application. The results are based on the study presented in Paper F, which considers a different vehicle configuration with electric machines

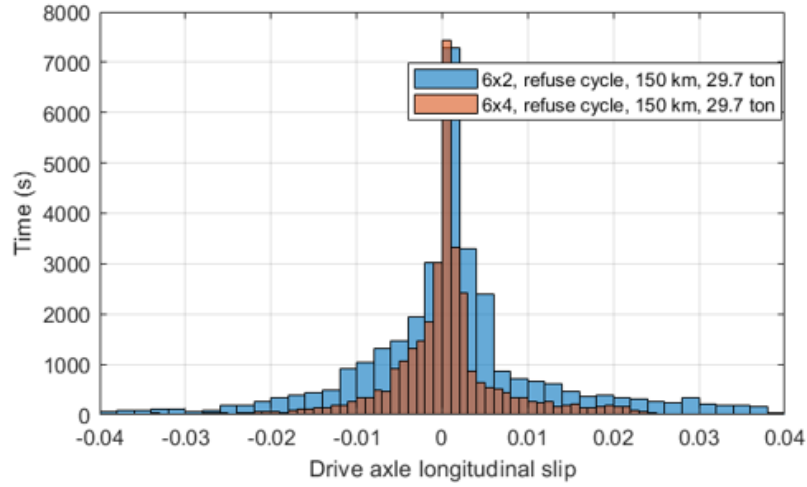


Figure 6.4: Longitudinal slip of the single driven axle for two vehicles with different axle configurations.

and friction brakes on each wheel of the two driven rear axles. Additionally, the electric machine characteristics and actuator maps differ from the previous case study. Furthermore, the energy-oriented baseline in Paper F minimises electrical energy consumption rather than instantaneous actuator power losses as formulated in Chapter 4. Despite these differences, the underlying principle of exploiting actuation redundancy to optimise actuator utilisation is consistent with the framework developed in this thesis.

The influence of driving cycle characteristics and drive-axle configuration on tyre degradation is first investigated. As shown in Figure 6.3, driving cycles with frequent acceleration and braking events result in larger variations in longitudinal slip compared with smoother regional distribution operation. Since the adopted tyre wear model relates degradation to longitudinal slip, these variations directly influence the resulting tyre wear cost. Since the adopted tyre wear model relates degradation to longitudinal slip, these variations directly influence the resulting tyre wear cost.

The influence of drive-axle configuration is further illustrated in Figure 6.4. Vehicles with fewer driven axles require higher propulsion and regenerative torque per driven wheel, resulting in increased longitudinal slip and tyre degradation. This highlights the importance of distributing longitudinal forces across multiple driven axles in distributed electric propulsion systems.

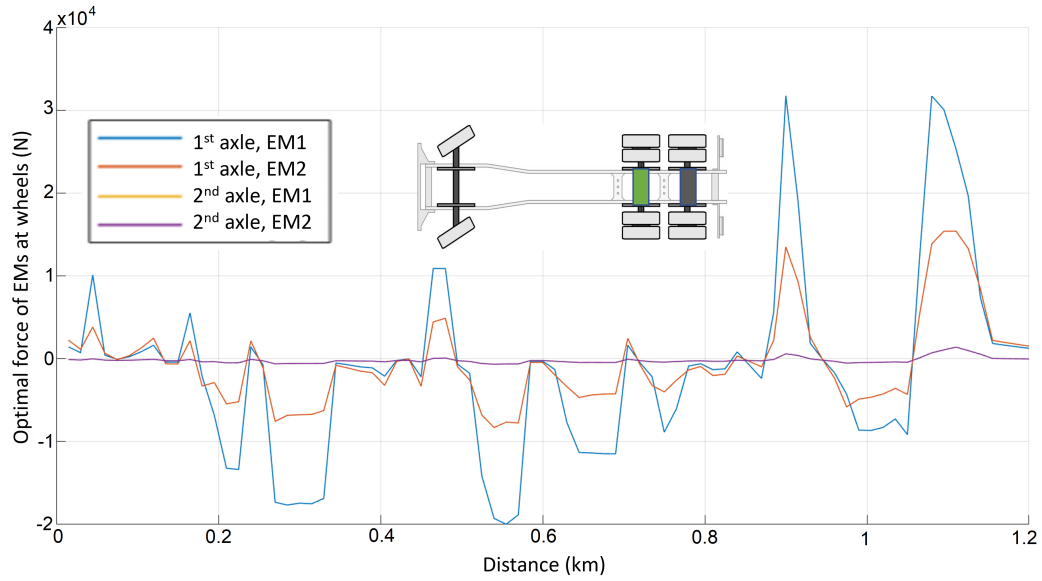


Figure 6.5: Controller performance with an energy-only cost function, where tyre wear is neglected. The optimal energy distribution is achieved by deactivating the EMs on the second axle.

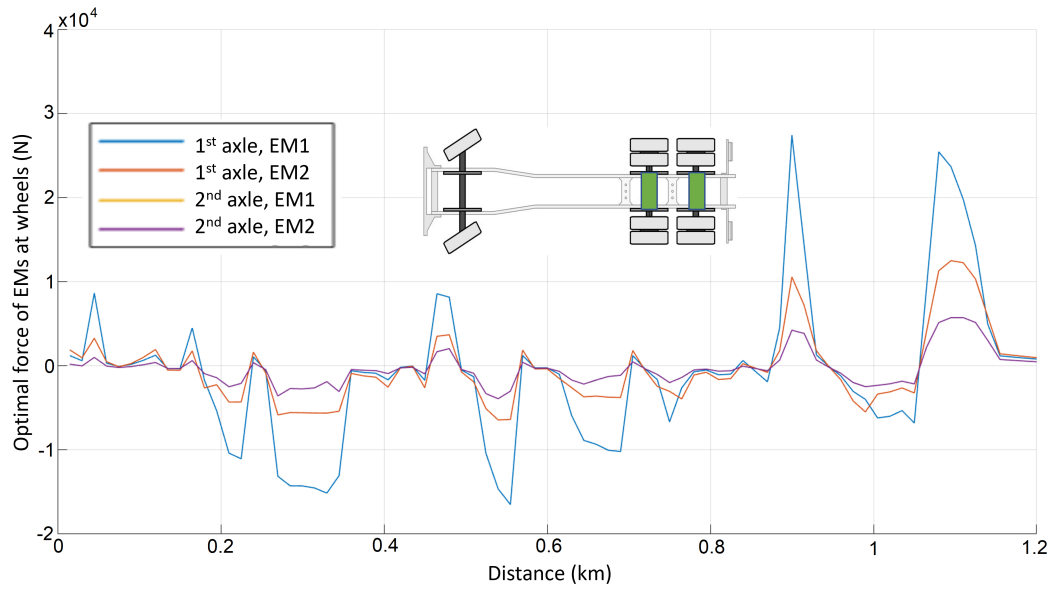


Figure 6.6: Controller performance when the cost function accounts for both energy consumption and tyre wear for the same driving cycle as in Figure 6.5. The controller achieves a cost-optimal power distribution by activating the EMs on the second axle, with both EMs operating at equal power levels.

Table 6.2: Metrics on different controllers.

Control strategy	energy cost*	tyre wear cost*	total cost*
w/o tyre wear	0.238	0.033	0.271
w tyre wear	0.244	0.016	0.260

*Unit of cost is Euro/ton/year/10km.

The energy-oriented coordination strategy and the proposed tyre-wear-aware strategy are compared for a 6×4 heavy vehicle equipped with four electric machines. As shown in Figure 6.5 and Figure 6.6, the energy-oriented strategy prioritises actuator efficiency and tends to concentrate torque on the most efficient axle configuration, including deactivation of the second driven axle. In contrast, the tyre-wear-aware strategy distributes the propulsion demand more evenly across the driven axles, reducing tyre slip and associated degradation.

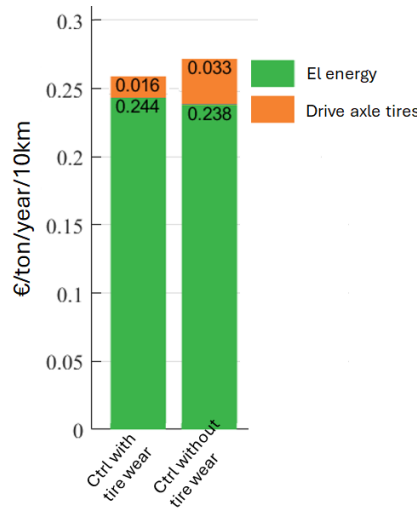


Figure 6.7: The difference between total costs and cost factors as a result of implementing two different controllers. 4% cost reduction is observed when tyre wear cost is included in the controller.

The resulting cost comparison in Table 6.2 and Figure 6.7 shows that incorporating tyre wear into the optimisation objective reduces the total operating cost by approximately 4% for the evaluated regional distribution cycle. Although the tyre-wear-aware strategy results in a small increase in electrical energy cost, the reduction in tyre degradation provides a larger overall economic benefit.

These results demonstrate that the proposed cost-rate formulation can be extended beyond the specific vehicle configurations studied in this thesis. By expressing energy consumption and tyre degradation within a common economic

framework, the actuator coordination problem can account for both short-term energy efficiency and long-term component degradation. The achievable benefits depend on the degree of actuation redundancy, driving cycle characteristics, powertrain efficiency, and the relative economic contribution of energy and tyre wear costs.

Chapter 7

Conclusion and Future work

7.1 Conclusion

This thesis investigated how heavy electric vehicles with modular electric axles can be designed and controlled to minimise operational cost while retaining safe vehicle motion. Based on the research questions, the thesis concludes that modular distributed electric propulsion is most beneficial when propulsion-system design, actuator coordination, and vehicle-motion control are treated as one integrated problem. The results show that such an approach can improve energy efficiency and driving range, reduce operational cost, and maintain safe and predictable motion under demanding driving conditions.

Regarding the **RQ1**, the thesis shows that the propulsion system of on-road electric heavy commercial vehicles can be designed more effectively by functionally differentiating the roles of electric axles. Chapter 3 and Paper A show that a distributed architecture with a dedicated cruise axle and a dedicated startability axle provides a better match to the dominant operating modes of long-haul vehicles than traditional layouts with a single drive train on a single axle or axle group. The cruise axle can be optimised for efficient steady-state highway operation, while the startability axle can provide the high tractive effort required at low speed and on steep grades. The results further show that hardware sizing and actuator (including axle limitation) coordination must be optimised jointly rather than separately. For the studied long-haul use case, Paper A showed that assigning around 50% of the total electric propulsion power to the cruise axle and selecting a cruise axle gear ratio in the range of 12 to 13 maximised range while satisfying cruise and startability requirements. Modular electric axles are therefore not only a packaging alternative, but also an enabler of operating-mode-based propulsion-system control design.

With respect to the **RQ2**, the thesis concludes that optimisation-based axle and actuator coordination provides a systematic and physically meaningful way

to distribute wheel torque demands while minimising power losses. Chapter 4, together with Papers B and C, shows that the coordination problem is naturally formulated as a control allocation problem in which redundant electric drive-trains and friction brakes are coordinated to satisfy a requested motion. Paper B demonstrated that power-loss minimisation provides a more physically grounded optimisation objective than heuristic prioritisation, while Paper C showed that weighted multi-objective formulations are flexible but sensitive to tuning, whereas equality-constrained power-loss minimisation offers better numerical conditioning and clearer physical interpretation when the requested motion remains feasible. These results are strengthened by Paper D, where the strategies were assessed in a high-fidelity heavy-vehicle simulation environment. There, it was shown that considering idle losses of the electric machines in the coordination logic can reduce energy consumption by up to 7% compared with a strategy based on equal friction utilisation. Efficient coordination should therefore not only determine how much torque to assign to each actuator, but also when a given actuator should remain inactive.

For the **RQ3**, the thesis shows that safe and predictable vehicle behaviour cannot be ensured through energy-efficient wheel torque allocation alone. Chapter 5 and Paper E demonstrate that lower-level control allocation must be complemented by vehicle-level supervision that regulates global force and yaw-moment requests with explicit awareness of stability boundaries. To achieve this, the thesis introduced a hierarchical wheel torque coordination framework in which an adaptive linear time-varying MPC generates safe global force references, and a lower-level control allocator distributes them among the available actuators while minimising power losses. Paper E showed that this framework improves path tracking, yaw stability, and steering predictability in demanding manoeuvres such as double lane changes and split-friction cornering on low-friction roads. A key result is that fixed state-weight configurations are not sufficient across different operating conditions. Instead, adaptive weighting of longitudinal and lateral control objectives, together with dynamically updated safe operating limits, is needed to balance agility, efficiency, and stability. In addition, the braking-in-curve study in Chapter 5 showed that the control allocator can be enhanced by relaxing lateral tyre-force demands in a controlled manner to increase longitudinal braking capability without destabilising the vehicle. Safe motion therefore requires tyre-force-aware and hierarchical coordination, in which the vehicle-level control objective is adapted according to the available stability margin.

To answer the **RQ4**, the thesis concludes that actuator coordination can be extended from pure energy minimisation to operational-cost minimisation by including tyre wear in the optimisation objective. Chapter 6, together with Papers F and G, establishes this contribution. Paper F introduced a wear-aware control allocation formulation in which tyre wear was incorporated into actuator coordina-

tion, showing that wheel torque distribution influences not only energy consumption but also tyre degradation. Energy-only optimisation tends to favour shutting down some electric machines and concentrating propulsion on fewer driven axles, whereas tyre-wear-aware optimisation distributes propulsion more evenly to reduce longitudinal slip. Paper G further extended this formulation by expressing actuator power losses and tyre wear in common monetary units within a unified operational-cost-based framework. This enables a direct trade-off between short-term energy use and long-term tyre degradation. For the studied cases, the results showed reductions in total operational cost of around 1% to 4%, depending on the vehicle configuration, drive cycle, and gross vehicle weight. The central conclusion is therefore that operational-cost minimisation provides a broader and more meaningful control objective than energy efficiency alone.

Summarising the contribution of the appended papers, the thesis shows a clear methodological progression. Paper A addresses the joint hardware and control design of modular cruise and startability axles. Papers B and C establish the theoretical and numerical foundation for optimisation-based control allocation. Paper D extends this foundation to high-fidelity simulation and highlights the importance of idle-loss-aware actuator coordination. Paper E introduces hierarchical predictive motion control to ensure safe vehicle behaviour under demanding conditions. Papers F and G then extend the optimisation objective from energy consumption to operational cost by incorporating tyre wear and expressing wear and actuator losses in unified monetary terms. Taken together, the thesis chapters connect these papers into a coherent systems perspective in which architecture design, actuator coordination, and motion control are tightly coupled.

In conclusion, the answers to the research questions show that modular electric axles can serve as a practical and powerful enabler for future heavy electric vehicles. A functionally differentiated propulsion architecture improves the match between drivetrain design and vehicle operating modes. Optimisation-based actuator coordination with explicit physical loss models improves efficiency and range. Hierarchical predictive motion control with adaptive objectives improves stability, agility, and steering predictability in critical manoeuvres. Extending the optimisation framework to include tyre wear and unified operational-cost terms enables a transition from energy management to operational-cost management. Taken together, the thesis and the appended papers show that modular distributed electric propulsion, when supported by integrated design and control methods, is a promising pathway toward heavy electric vehicles that are simultaneously more efficient, more economical, and safer to operate.

7.2 Future work

While this thesis provides strong concept-level and method-level evidence for the potential of modular electric axle architectures and optimisation-based actuator coordination, several directions remain open for future research.

Most results are based on simulation studies using representative vehicle and tyre models under the driving cycles, road conditions, and vehicle configurations considered in this work, rather than full-scale experimental validation on physical heavy vehicles. Extending the evaluation to a broader range of driving cycles, road surface conditions, and vehicle configurations would help establish the generality of the proposed methods, while experimental validation would be required to confirm that the results hold under effects that simulation cannot fully capture, such as unmodelled actuator dynamics, sensor noise, and communication delays.

The developed control methods focus on vehicle motion in the road plane, encompassing longitudinal, lateral, and yaw dynamics, but do not explicitly address rollover prevention, active steering control, or active suspension control. These are safety-relevant functions in heavy vehicles, particularly for high-centre-of-gravity configurations and manoeuvres involving sudden lateral load transfer, and their interaction with the proposed actuator coordination framework has not been investigated. Integrating vertical and roll dynamics into the control formulation is therefore an important direction for extending the safety guarantees established in this thesis.

The MPC-based global force generator was tuned and verified for the vehicle configurations considered in this thesis. Validating the adaptive tuning concept across different vehicle configurations, together with further investigation into robustness against model uncertainties and computational efficiency, would help refine the framework for real-time implementation.

The actuator coordination strategies developed for two driven axles, including the ILPLM formulation, rely in part on conditional logic that does not scale directly to vehicles with more than two driven axles. For such configurations, reformulating the ILPLM strategy as a mixed-integer optimisation problem would be necessary to preserve generality and computational tractability as the number of axles increases. Related to this, analysing different axle-usage modes under actuator failure scenarios would extend the fault-tolerance aspects of the framework and provide insight into how coordination strategies should adapt when one or more actuators become unavailable.

Several relevant operational cost factors were outside the scope of the cost formulation developed here. Thermal management of electric machines, power electronics, and batteries can significantly affect both efficiency and component lifetime, particularly under sustained high-load operation. Auxiliary loads, such as HVAC and onboard electronics, contribute to total energy consumption but

were not included in the optimisation. Battery ageing, which depends on charging and discharging patterns as well as thermal history, was not modelled, meaning that the wear-aware cost formulation addresses mechanical (tyre) degradation but not electrochemical degradation. Charging strategy and maintenance scheduling, which affect total cost of ownership at a fleet level, were also not considered. Relatedly, incorporating detailed thermal models and actuator hysteresis effects, along with more detailed power electronics and electrical system models, would improve the precision of the energy consumption results. A more complete operational cost model would need to integrate all of these factors alongside the energy and tyre wear terms already developed.

For tyre wear specifically, the framework mainly addresses wear associated with longitudinal slip during actuator coordination, since lateral slip was assumed small for the manoeuvres considered. A full integration of lateral-slip-aware wear minimisation, which would require coupling the wear model with lateral and yaw motion control, remains an open direction for future work, particularly for scenarios involving cornering, lane changes, or low-friction manoeuvring where lateral slip may become significant.

Taken together, these limitations mean that the presented conclusions should be interpreted as strong concept-level and method-level evidence for the potential of modular electric axle architectures and optimisation-based actuator coordination, rather than as production-ready solutions. Bridging the gap toward deployment would require experimental validation, extension to vertical and lateral vehicle dynamics, and a more comprehensive operational cost model incorporating thermal, electrochemical, and fleet-level factors.

Appendix A

Verification of idle losses in IMs through real vehicle tests

To verify the idles losses in IMs, a preliminary investigation was performed on a vehicle, configured with CEP, using real vehicle tests. The vehicle as shown in Figure A.1, is configured with four IMs along with inverters, which all drive a common shaft. The inverter software was also programmed with a specialised interface, giving individual torque control with speed limit to each machine and with a capability to switch on/off each inverter. The specialised software interface

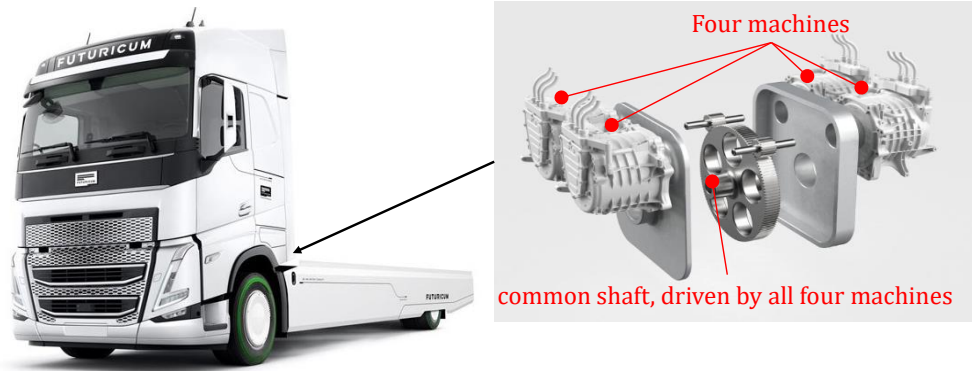


Figure A.1: Layout of IMs on a 4x2 tractor with specialised interface for discrete machine usage [47].

also relayed individual actuator signals, statuses and limits. The necessary signals such as machine voltage, torque output, and machine speed were available on vehicle CAN [60]. To measure the DC current, clamp sensors [20] were used on cables connecting the high-voltage battery to the individual inverters.

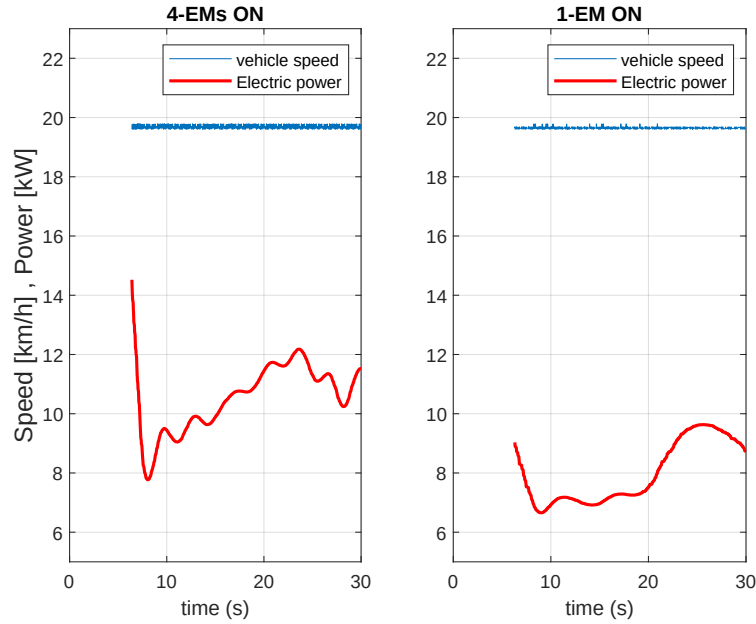


Figure A.2: Results from real vehicle measurements. The electric power consumption are evaluated for the case using 4 EMs and 1EM, for the same vehicle operating conditions.

A series of test runs were performed, using 4 EMs followed by switching off 3 machines and evaluating the input and output power at the EMs. Post processed test results are plotted in figure A.2. Comparisons of electric power consumption, was done for the same vehicle motion. The tests indicated that there are vehicle operating conditions where one can reduce power losses by switching off some machines. However, more analysis and investigation is needed through real vehicle and rig tests, so the author chose to not lift these preliminary results as a contribution to the thesis.

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