



CHALMERS
UNIVERSITY OF TECHNOLOGY



Evaluation of wheel torque coordination strategies for heavy battery electric vehicles

Master's thesis in Mobility Engineering (Automotive Track)

Simran Joy

Abner Ankit Lawrence

DEPARTMENT OF MECHANICS AND MARITIME SCIENCES

CHALMERS UNIVERSITY OF TECHNOLOGY

Gothenburg, Sweden 2025

www.chalmers.se

MASTER'S THESIS IN MOBILITY ENGINEERING (AUTOMOTIVE TRACK)

Evaluation of wheel torque coordination strategies for heavy battery electric vehicles

Simran Joy
Abner Ankit Lawrence



Department of Mechanics and Maritime Sciences
Division of Vehicle Engineering and Autonomous Systems
Vehicle Dynamics group
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2025

Evaluation of wheel torque coordination strategies for heavy battery electric vehicles.
Master's Thesis
Simran Joy
Abner Ankit Lawrence

© Simran Joy, 2025.
© Abner Ankit Lawrence, 2025.

Supervisor: Sachin Janardhanan, Volvo Group Trucks Technology, Gothenburg, Sweden
Examiner: Bengt Jacobson, Mechanics and Maritime Sciences, Chalmers University of Technology,
Gothenburg, Sweden

Master's Thesis 2025
Department of Mechanics and Maritime Sciences
Chalmers University of Technology
SE-412 96 Gothenburg
Sweden
Telephone +46 31 772 1000

Typeset in L^AT_EX
Printed by Chalmers Reproservice
Gothenburg, Sweden 2025

Evaluation of wheel torque coordination strategies for heavy battery electric vehicles.

Simran Joy

Abner Ankit Lawrence

Department of Mechanics and Maritime Sciences

Division of Vehicle Engineering and Autonomous Systems

Vehicle Dynamics Group

Chalmers University of Technology

Abstract

With the rapid evolution of technology and growing environmental concerns, the demand for electric vehicles has increased significantly. The main challenge for a heavy battery electric vehicle is to combat the efficiency and load carrying capacity on different kinds of roads (country roads and highway) ranging from a tarmac road with high coefficient of friction to a low friction road.

Modular E-axles such as cruise and startability axles have been introduced in this research with different types of electric machines and gear ratios to make it a reliable, cost efficient and effective setup for achieving a higher driving range along with less power losses. The different types of power losses that have been considered in this study are drivetrain losses, longitudinal tyre slip losses, rolling resistance losses, friction brake losses. However, only the drivetrain losses have been minimised in this work.

Three different kinds of wheel torque coordination strategies have been discussed in this thesis for allocating force/torque requests to the actuators (electric machine, brakes) in order to evaluate the energy savings for different types of trucks. Out of the three strategies, two of them are based on power loss minimisation and is compared to the third strategy where equal friction is achieved at the wheels. The performance of these strategies were evaluated using real world driving cycles.

However, the primary challenge lies in determining the optimal balance between energy efficiency and the vehicle's safety factor. Different methods like finding the lateral margins, friction circles of the tyres at the axle level have been formulated and implemented in order to find a safety metric. This thesis aimed to identify an optimal energy-efficient strategy while also defining a suitable safety metric.

Keywords: Wheel torque coordination strategies, heavy battery electric vehicles, load carrying capacity, power loss, electric machines, friction brakes

Acknowledgements

At the very outset, we would like to extend our heartfelt thanks to everyone who has been involved in our research. Firstly, we would like to thank our supervisor Sachin Janardhanan and examiner Bengt Jacobsson for their unwavering support and guidance, investing their invaluable time, showing a lot of interest and also for providing their profound insight throughout the tenure of our thesis.

We also would like to thank our manager Sofi Sjögren from Volvo GTT for giving us the opportunity and also for providing us with an intellectually enriched environment and also for all the sources to make this research possible.

We would also like to thank all the fellow colleagues from Common Architecture and Shared Technology (CAST) at Volvo and also the faculty from Vehicle Engineering and Autonomous System (VEAS) at Chalmers for all their constructive feedback and solutions on how to improve our work. Also, we are very grateful to Sonja for taking care of all the administrative work at Chalmers.

Last but none the least, we would like to thank our families as well for their continuous support and sacrifices throughout our lives.

Simran Joy, Abner Ankit Lawrence, Gothenburg, June 2025

List of Acronyms

Below is the list of acronyms that have been used throughout this thesis listed in alphabetical order:

ABS	Anti-lock Braking Systems
BEV	Battery Electric Vehicle
EFU	Equal Friction Utilisation
ESC	Electronic Stability Control
EV	Electric Vehicle
HBEV	Heavy Battery Electric Vehicle
ILPLM	Power Loss Minimisation using Idle Losses
PLM	Power Loss Minimisation
SWA	Steering Wheel Angle
VTM	Volvo Transport Model

Nomenclature

Below is the nomenclature of indices and variables that have been used throughout this thesis.

Indices

i	Index for number of axles
j	Index for number of wheels

Variables

$a_{x,req}$	Longitudinal acceleration request
δ_{req}	Steering angle request at the wheel
F_{xj}	Longitudinal force on a wheel
F_{yj}	Lateral force on a wheel
F_{zj}	Normal load on a wheel
$F_{\text{margin,front}}$	Lateral margin on the front axle
$F_{\text{margin,rear}}$	Lateral margin on the rear axle
$P_{elEM,i}$	Electric power of the electric machine
$P_{loss,sx}$	Total longitudinal tyre slip loss
$P_{loss,rr}$	Total rolling resistance loss
$P_{lossbrk,i}$	Power loss of the friction brakes
$P_{lossEM,i}$	Power loss of the electric machine
$P_{mechEM,i}$	Mechanical power of an electric machine
$r_{ew,i}$	Effective rolling radius of a wheel
$T_{brk,i}$	Torque request to the wheel from the friction brakes
$T_{brkw,i}$	Braking torque on a wheel
$T_{EM,i}$	Torque request to the wheel from the electric machine
$T_{reqbrk,i}$	Torque request to the friction brakes from the actuator coordinator
$T_{reqEM,i}$	Torque request to the electric drivetrain from the actuator coordinator
$v_{x,act}$	Actual longitudinal vehicle speed

$v_{x,req}$	Longitudinal velocity request
$v_{xi,j}$	Longitudinal speed of a wheel
$\omega_{z,i}$	Rotational speed of a wheel
ψ_{ry}	Road grade

Contents

List of Acronyms	ix
Nomenclature	xi
List of Figures	xv
List of Tables	xvii
1 Introduction	1
1.1 Problem Motivation	1
1.2 Goals and Objectives	2
1.3 Limitations	2
2 Power Losses	4
2.1 Tyre Slip Losses	4
2.2 Rolling Resistance Loss	5
2.3 Electric Drivetrain Losses	6
2.4 Electrical Power Conversion Losses	7
2.5 Friction Braking Losses	7
3 Methodology	8
3.1 Vehicle Topology	8
3.2 Selection of Driving Cycle	9
3.3 Simulation Model	12
3.3.1 Driving Cycle - Velocity profile	13
3.3.2 Driver Model	13
3.3.3 Motion Controller	13
3.3.4 Actuators	14
3.3.5 Vehicle Dynamics / VTM Model	14
3.3.6 Sim Stopper	15
4 Control Strategies	16
4.1 Power Loss Minimisation (PLM)	16
4.1.1 4x4 Configuration	18
4.1.2 6x4 Configuration	19
4.2 Power Loss Minimisation including Idle Losses (ILPLM)	21
4.3 Equal Friction Utilisation (EFU)	22
4.3.1 4x4 Configuration	23
4.3.2 6x4 Configuration	24

5	Results	25
5.1	Model Validation	26
5.1.1	Distance vs Time plots for 4x4 and 6x4 configuration	26
5.1.2	Longitudinal force request to the coordination strategies vs Distance plots for 4x4 and 6x4 configuration	27
5.1.3	Yawrate and SWA vs Distance plots for 4x4 and 6x4 configuration	27
5.1.4	Wheel torque allocation during steady state turning	29
5.1.4.1	4x4 truck configuration	29
5.1.4.2	6x4 truck configuration	31
5.1.5	Lateral deviation vs Distance travelled for 6x4 rigid truck configuration	33
5.2	Lateral stability	34
5.2.1	Lateral stability for 4x4 rigid truck configuration	34
5.2.2	Lateral stability for 6x4 rigid truck configuration	38
5.3	Energy Savings	42
5.4	Influence of disconnecting Startability Axle for 4x4 and 6x4 rigid truck configuration	43
6	Conclusion	45
7	Future Work	47
A	Appendix 1	50

List of Figures

2.1	Power losses tyre slip	5
2.2	Carcass construction of a radial tyre	6
2.3	Representation of an electric machine	6
3.1	Representation of Cruise and Startability Axle on a 4x4 and 6x4 rigid truck	9
3.2	E20 Gothenburg–Alingsås (90–110 km/h)	10
3.3	R1758 Bollebygd–Töllsjö (50–70 km/h)	10
3.4	R180 Hällered–Alingsås (70–90km/h)	11
3.5	Visual representation of the roads	11
3.6	Simulation Model	12
3.7	VTM Animation of a 4x4 rigid truck	14
4.1	Overview of PLM Algorithm	20
4.2	Overview of ILPLM Algorithm	21
4.3	EFU algorithm for 4×4 / 6x4 configuration	23
5.1	Distance vs Time plots for 4x4 configuration	26
5.2	Distance vs Time plots for 6x4 configuration	26
5.3	Force vs Distance plots for 4x4 configuration	27
5.4	Force vs Distance plots for 6x4 configuration	27
5.5	Yawrate vs Distance plots for 4x4 and 6x4 configuration	28
5.6	Steering angle vs Distance plots for 4x4 and 6x4 configuration	28
5.7	Friction circle (4x4 rigid, EFU, $\mu = 0.40$)	30
5.8	Friction circle (4x4 rigid, PLM, $\mu = 0.40$)	30
5.9	Friction circle (4x4 rigid, ILPLM, $\mu = 0.40$)	31
5.10	Friction circle (6x4 rigid, EFU, $\mu = 0.40$)	32
5.11	Friction circle (6x4 rigid, PLM, $\mu = 0.40$)	32
5.12	Friction circle (6x4 rigid, ILPLM, $\mu = 0.40$)	33
5.13	Lateral Deviation (6x4 rigid, $\mu = 0.40$)	34
5.14	Friction Circle with positive F_{margin} limits for a wheel	34
5.15	Friction Circle with negative F_{margin} limits for a wheel	35
5.16	Lateral force vs Distance (ILPLM, 4x4 rigid, $\mu = 0.40$, R1758)	36
5.17	Comparison of normalised lateral force vs Distance (4x4 rigid, $\mu = 0.40$, R1758)	37
5.18	Percent of samples versus index (4x4 rigid, $\mu = 0.40$, R1758)	38
5.19	Lateral force vs Distance (ILPLM, 6x4 rigid, $\mu = 0.40$, R180)	39
5.20	Comparison of normalised lateral force vs Distance (ILPLM, 6x4 rigid, $\mu = 0.40$, R180)	40
5.21	Percent of samples vs index (6x4 rigid, $\mu = 0.40$, R180))	41
5.22	Energy plot (4x4 rigid, $\mu = 0.40$, R1758)	42
5.23	Energy plot (6x4 rigid, $\mu = 0.40$, R1758)	42
5.24	Axle Switching (4x4 rigid, $\mu = 0.40$)	43

5.25	Axle Switching (6x4 rigid, $\mu = 0.40$)	44
6.1	Trade off plot(4x4 rigid $\mu = 0.40$)	45
6.2	Trade off plot(6x4 rigid $\mu = 0.40$)	46
A.1	Percent of samples versus index (4x4 rigid, $\mu = 0.40$, R180)	50
A.2	Percent of samples versus index, Zoomed view (4x4 rigid, $\mu = 0.40$, R180)	50
A.3	Percent of samples versus index (4x4 rigid, $\mu = 0.40$, E20)	51
A.4	Percent of samples versus index, Zoomed view (4x4 rigid, $\mu = 0.40$, E20)	51
A.5	Percent of samples versus index (6x4 rigid, $\mu = 0.40$, R1758)	52
A.6	Percent of samples versus index (6x4 rigid, $\mu = 0.40$, E20)	52
A.7	Energy plot (4x4 rigid, $\mu = 0.40$, R180)	53
A.8	Energy plot (4x4 rigid, $\mu = 0.40$, E20)	53
A.9	Energy plot (6x4 rigid, $\mu = 0.40$, R180)	54
A.10	Energy plot (6x4 rigid, $\mu = 0.40$, E20)	54

List of Tables

3.1	Common Vehicle and Environmental Parameters Used in Simulation	11
3.2	Vehicle-Specific Parameters for 4x4 Rigid Truck	12
3.3	Vehicle-Specific Parameters for 6x4 Rigid Truck	12
5.1	Configuration table for different roads with different coefficients of friction	25

1

Introduction

In recent years, the automotive industry has witnessed a rapid increase in interest towards Heavy Battery Electric Vehicles (HBEVs), largely driven by concerns over diminishing fossil fuel resources and rising environmental pollution. HBEVs represent a promising alternative to traditional combustion-engine vehicles, offering significant benefits such as swift torque response up to 10 to 100 times faster and precise torque measurement capabilities via motor current [1]. Despite these advantages, one significant challenge hindering broader adoption is their limited driving range, with typical mid-range EVs achieving approximately 150–200 kilometers per charge, considerably less than conventional vehicles powered by internal combustion engines.

Addressing the issue of limited driving range is critical to realizing the full potential of HBEVs, particularly in the heavy-duty sector, where operational demands are stringent and current battery technologies face constraints regarding energy density and charging infrastructure availability. Enhancing the autonomy of HBEVs thus requires focused research efforts, broadly categorized into two approaches: improvements in battery and energy storage technologies, including advancements in lighter, higher-density storage solutions like supercapacitors, and optimal control strategies aimed at efficient power utilization.

HBEVs are offered with varied configurations, including multiple axles and distributed propulsion layouts, each presenting unique opportunities for optimizing energy efficiency. Leveraging the modularity, flexibility, and cost-neutral aspects of electric motors, distributed drivetrain architectures are increasingly explored as a viable solution. Optimal control strategies that dynamically distribute power among multiple actuators have demonstrated significant potential in reducing energy losses, extending driving range, and improving vehicle safety and performance [2–5]

Previous research has conceptually explored different algorithms and energy management strategies aimed at optimal torque distribution and instantaneous minimization of power losses. Nevertheless, significant room remains for developing practical, real-time control strategies tailored specifically to various HBEV configurations, such as 4x4 rigid trucks and 6x4 rigid trucks. Addressing these challenges through a systematic and modular approach could significantly enhance the feasibility, efficiency, and operational range of HBEVs, especially in heavy-duty applications.

This thesis focuses precisely on these aspects: evaluating the energy savings achievable through distributed propulsion, comparing the wheel torque coordination strategies using proposed safety and energy metrics, and developing a modular control scheme adaptable to diverse vehicle configurations. Ultimately, this research aims to meaningfully contribute to the broader efforts toward sustainable and efficient electric transportation.

1.1 Problem Motivation

This thesis introduces and investigates a novel modular axle drive concept featuring distributed power allocation across multiple axles. Special attention is given to differentiating between axles designated

as "cruise axles," which are optimized for consistent highway speeds, and "startability axles," designed specifically for initiating vehicle motion and improving acceleration. This approach is particularly advantageous for heavy-duty commercial vehicles, with the main goal of increasing vehicle range through carefully coordinated design and operational control of the axles. To efficiently manage power distribution among these axles, wheel torque coordination strategies are employed, primarily focused on reducing energy consumption.

In contrast to conventional diesel-powered systems, electric propulsion architectures inherently provide greater flexibility and allow for more effective utilization of electric motors distributed across multiple axles. This higher degree of flexibility and actuation capability enables motors to operate at optimal efficiency points, significantly enhancing energy efficiency. Moreover, this distributed configuration supports other beneficial objectives, such as reducing component wear, ensuring balanced friction distribution among wheels, and enhancing overall vehicle safety [6].

Within this thesis, three distinct wheel torque coordination algorithms are evaluated, each designed to improve energy efficiency while maintaining vehicle stability under typical driving conditions. These algorithms are intended to complement existing safety systems like Anti-lock Braking Systems (ABS) and Electronic Stability Control (ESC), with also enabling enhanced functionalities like torque vectoring, brake blending which manage torque distribution and braking coordination during critical or emergency situations. During such critical conditions, maintaining safety supersedes the goal of energy efficiency.

Two of the torque distribution algorithms examined specifically target minimizing instantaneous power losses in actuators. These are benchmarked against a third algorithm that prioritizes even friction usage across all wheels. All algorithms function as closed-loop systems, continuously adjusting based on real-time feedback from actuator and vehicle sensors.

The presented actuator coordination methods are suitable for both propulsion and braking scenarios but are specifically applied to electric drivetrains and friction brakes within the rigid truck. Equal torque distribution to wheels on each axle is assumed, and the electric drivetrains are presumed to operate optimally for either propulsion or regenerative braking scenarios.

1.2 Goals and Objectives

The main objectives of this thesis are:

- To evaluate the energy savings possible for Heavy Battery Electric Vehicles (HBEVs) for various vehicle configurations such as a 4x4 rigid truck, 6x4 rigid truck using the distributed propulsion concept with safety constraints and a power loss minimisation algorithm for different driving cycles.
- To propose safety and energy metrics to compare wheel torque coordination strategies.
- To present a modular scheme of the axle control algorithm to support the distributed e-axles concept, extendable to other vehicle configurations.

1.3 Limitations

- As there is no existing vehicle available for testing, all controllers and models developed in this thesis are implemented in simulation using MATLAB and Simulink.

- All states and variables used in the controller are assumed to be known or measurable and are readily provided within the simulation environment.
- Aerodynamic forces have only be calculated but not modeled.
- Computational performance and solver efficiency of the optimization algorithms are not evaluated.
- Hardware-in-the-Loop (HiL) testing is not performed.
- Detailed actuator models (i.e., high-fidelity representations) are not considered.
- The influence of trailer not considered.

2

Power Losses

To effectively compare different control strategies, it is essential to evaluate losses that are strongly influenced by wheel torque allocation. During vehicle operation, power is dissipated through various mechanisms, including aerodynamic losses, rolling resistance losses, tyre slip losses, electric drivetrain losses, cooling systems, battery losses, and auxiliary components. The important sources of power losses considered in this thesis are shown below:

2.1 Tyre Slip Losses

During acceleration or deceleration of a vehicle, torque is applied to the wheels, generating a longitudinal force. This force is directly related to the phenomenon of longitudinal slip, which arises from the relative velocity between the translational motion of the wheel center and its rotational motion. Longitudinal slip is quantified using the difference between the translational velocity of the wheel center and the wheel's rotational velocity, multiplied by the effective wheel radius [6]. The tyre slip power loss for the whole vehicle assuming the torque is equally distributed between the two wheels on an axle can be expressed as:

$$P_{\text{loss_sx}} = \sum_j 2 \cdot F_{xj} (v_{xj} - \omega_j \cdot r_{ej}) \quad (2.1)$$

where, F_{xj} is the longitudinal force of a single wheel
 v_{xj} is the longitudinal velocity at the wheel center
 ω_j is the rotational velocity of the tyre
 r_{ej} is the effective rolling radius of the tyre

Tyre slip losses, namely the longitudinal slip are only calculated due to wheel torque coordination and are not minimised. The wheel forces produced due to high friction levels are within the tyre force limits. Therefore the lateral slip produced did not influence wheel torque allocation and it was nearly the same for all the control strategies.

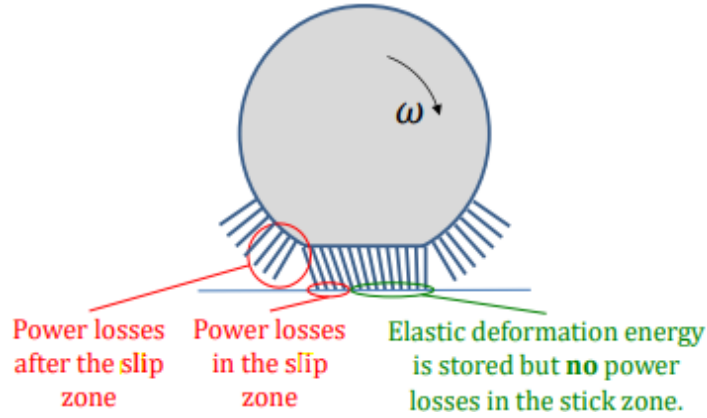


Figure 2.1: Power losses tyre slip

2.2 Rolling Resistance Loss

Rolling resistance represents another major source of power loss, stemming from the deformation of the rubber elements in the tyre's contact patch (carcass) which often leads to a phenomena known as hysteresis in which the rubber elements come back to their initial condition after some time but lose some energy in the form of heat which is generated due to the tyre's internal damping. Consequently, the centre of pressure shifts towards the front of the tyre, generating a moment that opposes the direction of rolling [7].

The magnitude of rolling resistance also depends on a plethora of factors like tyre structure, inflation pressure, material composition, type of compound, vertical(normal) load, operating conditions, and the applied drive or brake torque. These losses are commonly modeled using empirical rolling resistance coefficients.

$$P_{\text{loss_rr}} = \sum_j 2 \cdot C_{rr} \cdot F_{z,j} \cdot v_{x,j} \quad (2.2)$$

where, C_r is the coefficient of rolling resistance
 $v_{x,j}$ is the longitudinal velocity of the wheel
 $F_{z,j}$ is the normal load on the wheel.

In our model, they are only calculated and not minimized which applies across all the control strategies. Also, we have assumed the same coefficient of rolling resistance for all the wheels across all strategies.

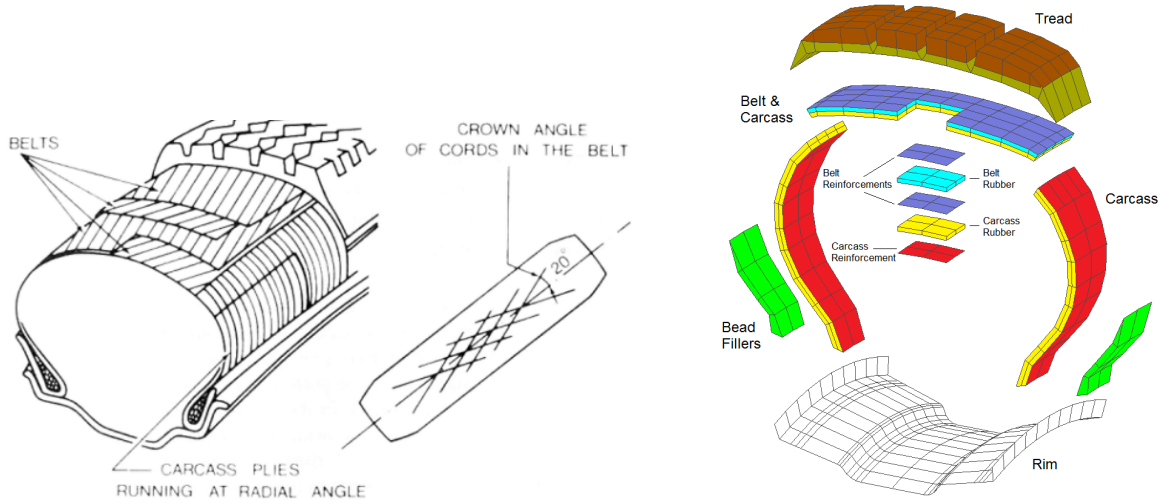


Figure 2.2: Carcass construction of a radial tyre

2.3 Electric Drivetrain Losses

In electric vehicles with motorized axles, losses occur not only in the electric drivetrain on each axle but also the losses originating from power source [8]. Electric machines like Permanent Magnet Synchronous Machine (PMSM) and Induction Motor (IM) have been used in this thesis due to their different characteristics as shown in the study [9]. There are different losses associated with these machines and they are broadly classified as copper, iron, windage and friction losses and are represented in [2].

The transmission on each axle i is represented by a single gear ratio $g_{r,i}$, and its efficiency $\eta_{\text{trmn},i}$ is defined as a function of the gear ratio, as:

$$\eta_{\text{trmn},i} = 0.99^{\frac{g_{r,i}}{3}}, \quad \text{where } i = 1, 2, 3 \quad (2.3)$$

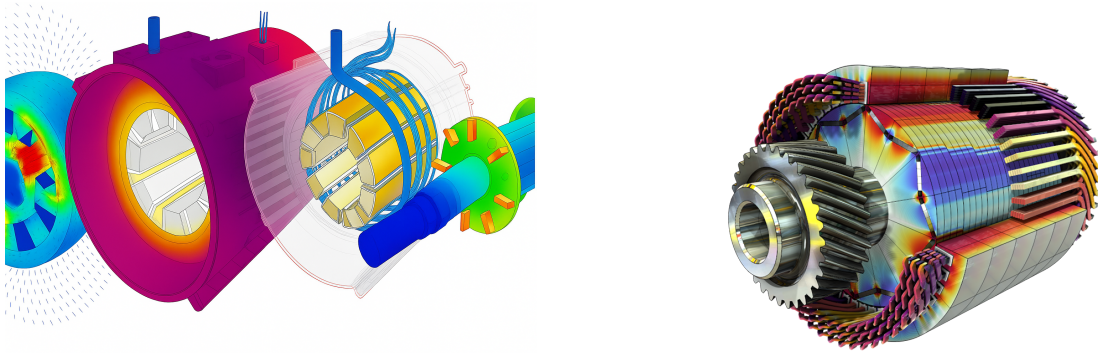


Figure 2.3: Representation of an electric machine

2.4 Electrical Power Conversion Losses

In an electric drivetrain, energy is converted at several stages. In drive mode, the battery feeds the inverter (DC–AC), the motor turns that electrical energy into torque, and the vehicle moves; in regeneration (“regen”), the process reverses to recharge the battery. Along the way, some energy is lost as heat: the battery incurs I^2R losses from its internal resistance, and the inverter loses power through device conduction and high-frequency switching during DC–AC or AC–DC conversion. These losses accumulate and reduce overall efficiency.

Battery and inverter losses are combined and modeled as lumped conversion efficiency:

$$P_{\text{mechEM}_i} = P_{\text{elEM}_i} \cdot \eta_{\text{mot}_i} \quad (2.4)$$

$$P_{\text{elEM}_i} = P_{\text{mechEM}_i} \cdot \eta_{\text{gen}_i} \quad (2.5)$$

where, P_{mechEM_i} is the mechanical power of the EM

P_{elEM_i} is the electrical power of the EM (Electric Machine)

η_{mot_i} the EM conversion efficiency

η_{gen_i} is the EM regeneration conversion efficiency.

2.5 Friction Braking Losses

Heavy commercial vehicles (HCVs) use pneumatic (air) braking systems. A compressor charges air reservoirs (tanks), and the pressurized air is routed through valves and lines to the brake actuators. Generating, storing, and transmitting this air consumes energy due to compressor inefficiency, pressure drops, and leakage. In this study, we treat these pneumatic losses as approximately constant and neglect them, since estimating them would require a detailed model of the full pneumatic circuit.

The friction brakes convert the vehicle’s kinetic energy into heat when caliper-mounted brake pads clamp a rotating brake disc (rotor) attached to the wheel hub.

Heat losses due to friction braking are given by:

$$P_{\text{loss_brk}} = \sum_j 2 \cdot T_{\text{brk}_j} \cdot \omega_j \quad (2.6)$$

The heat losses from friction braking can be modeled as a linear function of the wheel braking torque $T_{\text{brkw},j}$ and the corresponding wheel rotational speed $\omega_{w,j}$.

3

Methodology

3.1 Vehicle Topology

This research investigates three wheel torque coordination strategies applied to two vehicle configurations: a 4x4 rigid truck and a 6x4 rigid truck. In the 4x4 configuration, both the front and rear axles are driven, whereas in the 6x4 setup, propulsion is provided by the second and third axles.

For this study, the front axle of the 4x4 rigid truck and the rear axle of the rear axle group of the 6x4 rigid truck are referred to as the cruise axle. Conversely, for the 6x4 rigid truck, the rear axle of the 4x4 truck and the front axle of the rear axle group of the 6x4 configuration are designated as the startability axles as shown in Figure 3.1. The startability axle is primarily responsible for delivering high initial torque, particularly during vehicle launch and low-speed operation.

The cruise axle is equipped with a PMSM (Permanent Magnet Synchronous Machine) and a single-reduction transmission ratio, offering enhanced energy efficiency, consistent power delivery, and the capability to handle increased torque demands. This setup makes the cruise axle ideal for steady-state driving conditions.

In contrast, the startability axle is fitted with an asynchronous Induction motor (IM) and also utilizes a single-reduction transmission. While slightly less efficient than permanent magnet machines, the induction motor is favored for its robustness, reliability, and immunity to permanent magnet-related idling losses [5,6]. This makes it particularly suitable for high-load, transient driving conditions where durability and simplicity are prioritized.

In PMSM, the permanent magnet losses can be relatively high under idle conditions. In order to eliminate these losses, the introduction of a clutch or dog clutch can be introduced, which mechanically declutches the rotor from the wheel. However, the downside of this setup is the increased costs and requirement of additional space.

On the other hand, for an IM, the idle losses can be minimised, which makes it energy-efficient. The IM is equipped with a higher gear ratio for initial torque delivery, for instance, starting from a standstill position.

Another major reason for the placement of startability axle at the rear axle of a 4x4 truck and at the front axle of the rear axle group for a 6x4 truck is that it has dual wheels on that axle, because of which the mass of the axle is higher as compared to the other axles, thereby increasing the vertical or normal loads at the axle making it suitable for startability axle for initial torque delivery.

Together, these configurations support an architecture that enables the flexible and efficient distribution of drive torque between axles, forming the basis for the torque coordination strategies examined in this thesis.

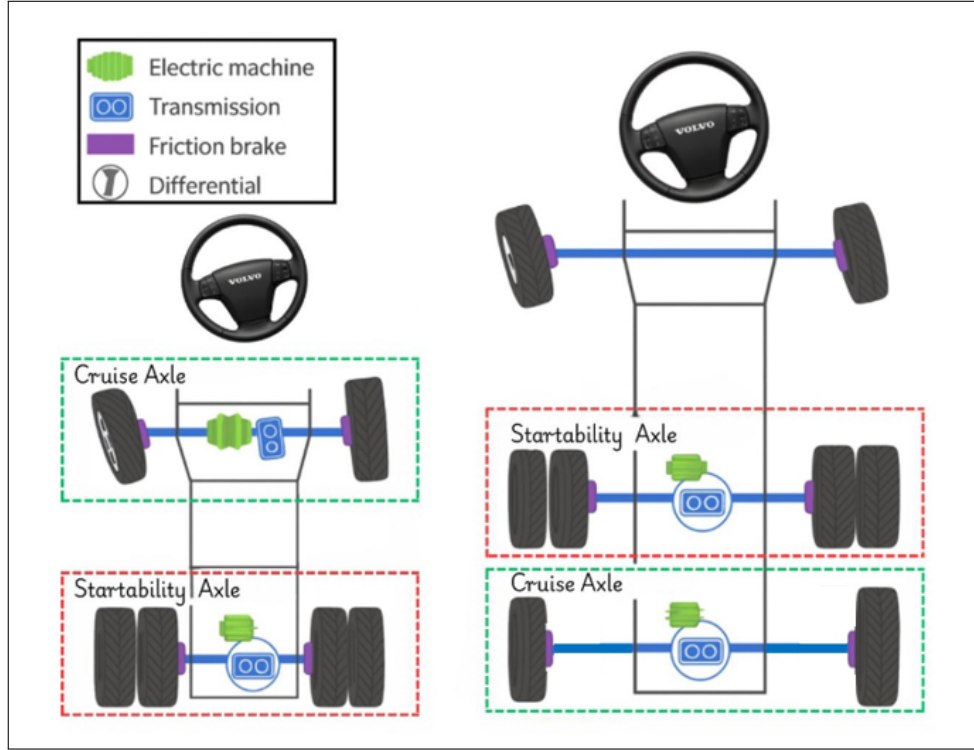


Figure 3.1: Representation of Cruise and Startability Axle on a 4x4 and 6x4 rigid truck

3.2 Selection of Driving Cycle

To evaluate the performance of the proposed control strategies mentioned in Section 4, representative driving cycles were derived from real-world Swedish road segments. Specifically, a section of the R1758 Bollebygd–Tollsjö country road, R180 Hällrad to Alingsås and the E20 highway between Gothenburg and Alingsås were selected for simulation.

The driving cycles were extracted and post-processed using data from Trafikverket’s road database, accessible via the VTM (Volvo Transport Model) library [11]. These profiles incorporate critical road characteristics such as topography, curvature, and speed limits, as well as detailed longitudinal and lateral geometry. This comprehensive dataset ensures that the simulation environment closely reflects actual driving conditions, allowing for a more accurate assessment of each control strategy’s effectiveness. The original road profile data obtained from the database covered approximately 16.25 km for the E20 highway, 15.45 km for the R1758 country road, and 8 km for the R180 Hällrad to Alingsås country road. To create a longer and more symmetric driving cycle, each dataset was mirrored to simulate a return journey, effectively doubling the total distance as shown in figure 3.2. As a result, the final driving cycles used in the simulations span 32.5 km for the E20 route and 30.9 km for the R1758 route, and 16 km for R180 route with the vehicle assumed to return to its initial starting point.

For all simulations, the coefficient of friction between all the tyres and the road surface was assumed to be constant. This simplification ensures consistency across all evaluated control strategies and driving scenarios.

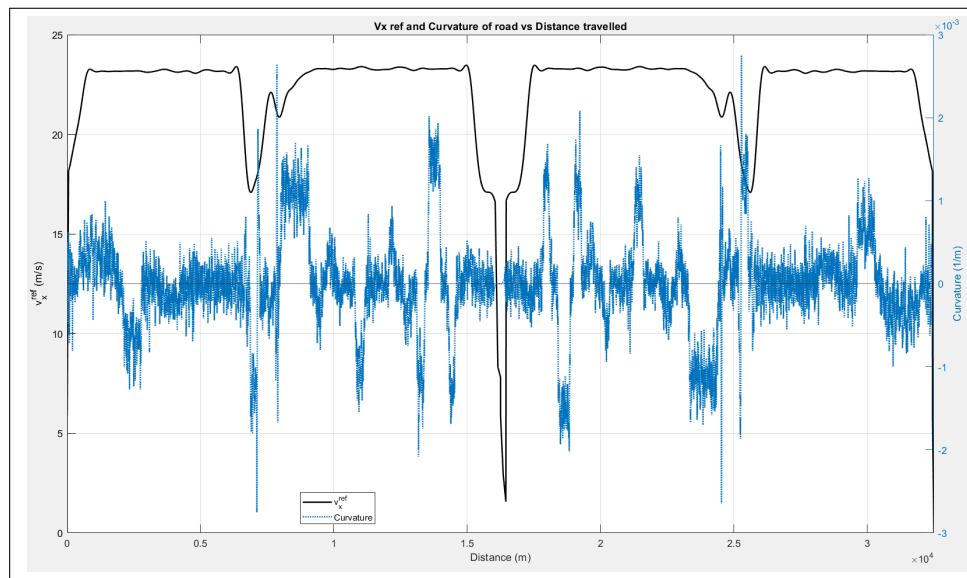


Figure 3.2: E20 Gothenburg–Alingsås (90–110 km/h)

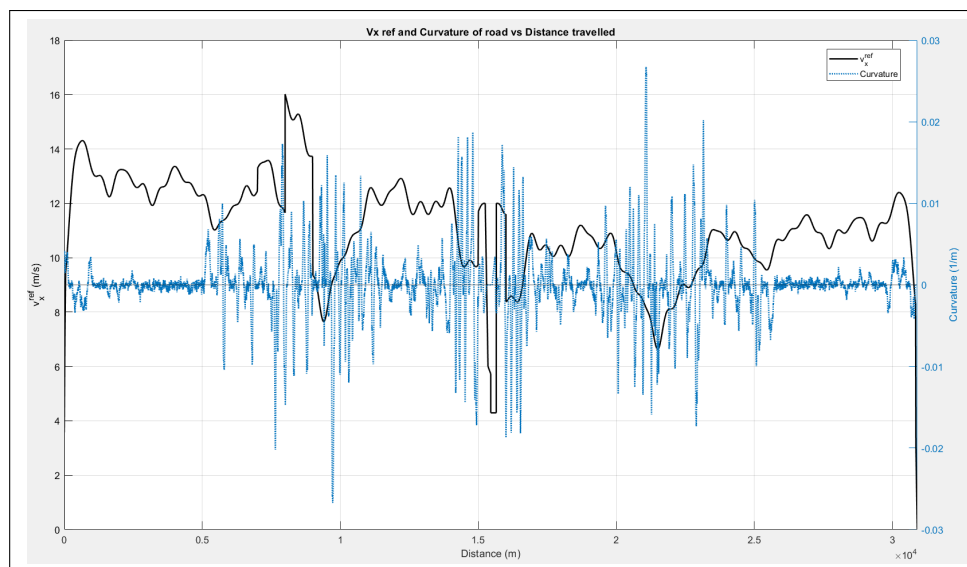


Figure 3.3: R1758 Bollebygd–Töllsjö (50–70 km/h)

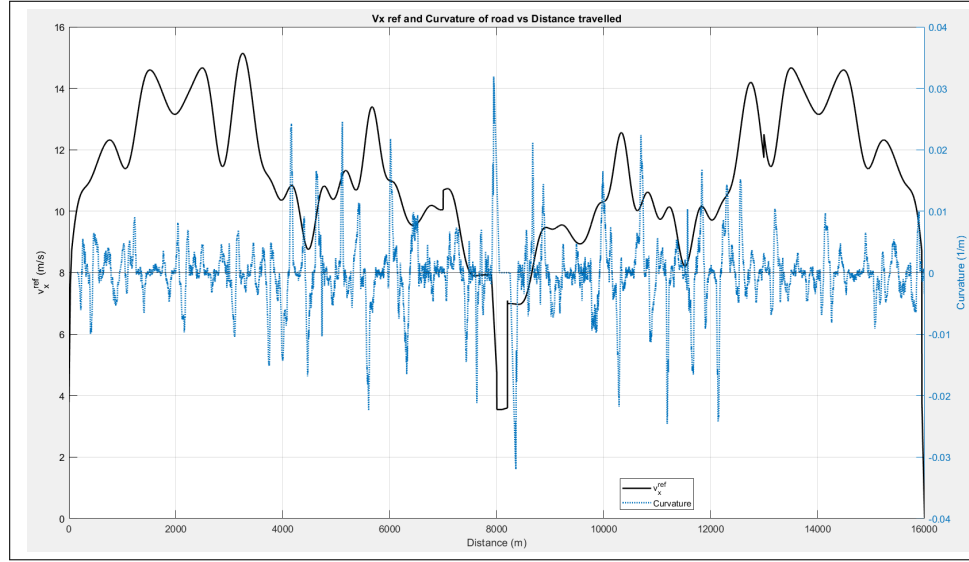


Figure 3.4: R180 Hällered-Alingsås (70-90km/h)



Figure 3.5: Visual representation of the roads

Table 3.1: Common Vehicle and Environmental Parameters Used in Simulation

Parameter	Unit	Symbol	Value
Rolling resistance coefficient	–	RRC	0.005
Frontal area	m^2	A_f	9
Drag coefficient	–	C_d	0.59
Air density	kg/m^3	ρ	1.2
Wheel radius	m	r_w	0.470
Gravitational constant	m/s^2	g	9.81
Cruisability axle gear ratio	–	gr_{crs}	14
Startability axle gear ratio	–	gr_{stb}	23
Battery capacity	kWh	E_{batt}	400
PMSM maximum power	kW	$P_{maxEM,1}$	300
IM maximum power	kW	$P_{maxEM,2}$	300

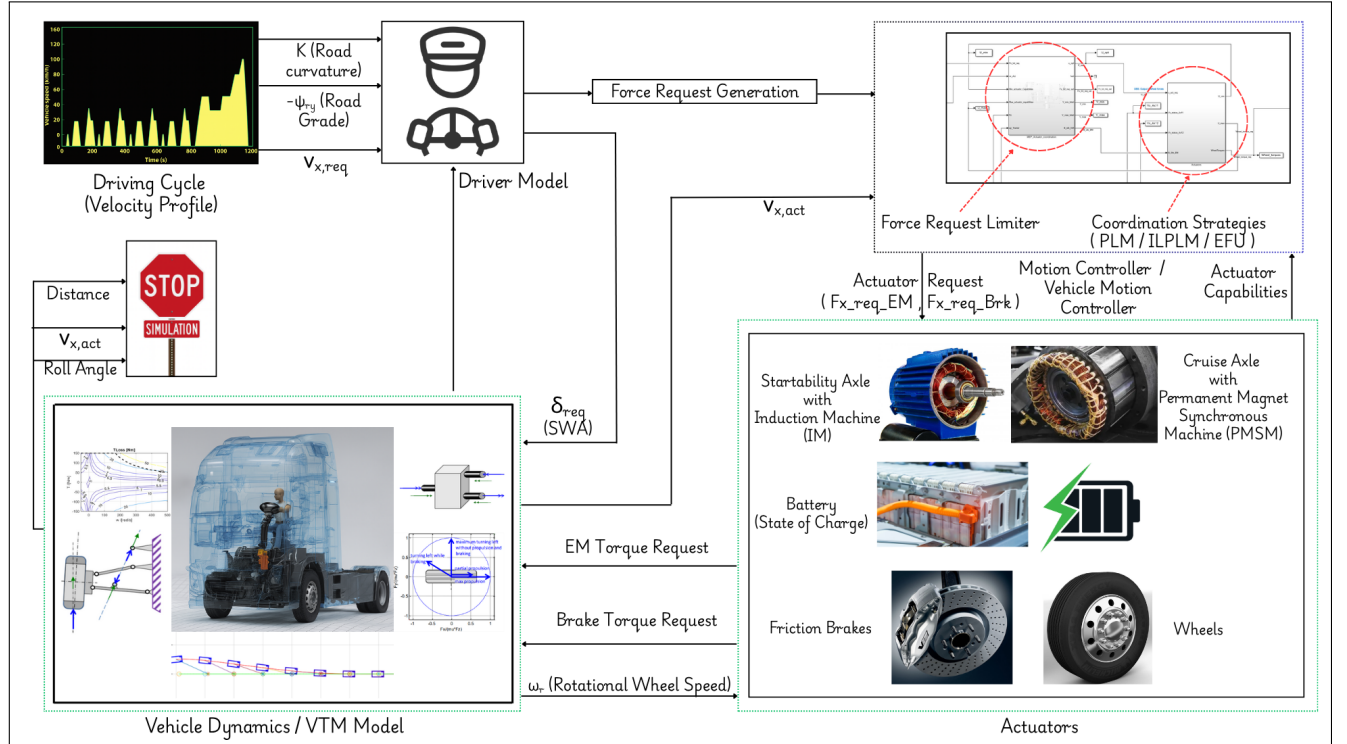
Table 3.2: Vehicle-Specific Parameters for 4x4 Rigid Truck

Parameter	Unit	Symbol	Value
Mass of vehicle combination	kg	m_{4x4}	17 559
Front track width	m	$tw_{f,4x4}$	2.03
Rear track width	m	$tw_{r,4x4}$	1.85

Table 3.3: Vehicle-Specific Parameters for 6x4 Rigid Truck

Parameter	Unit	Symbol	Value
Mass of vehicle combination	kg	m_{6x4}	25 334
Front track width	m	$tw_{f,6x4}$	2.01
Rear track width	m	$tw_{r,6x4}$	1.85

3.3 Simulation Model

**Figure 3.6:** Simulation Model

3.3.1 Driving Cycle - Velocity profile

The velocity profile for this thesis is implemented using a 1-D lookup table, which defines the reference speed profile of the vehicle as a function of distance travelled. This, in turn, helps the vehicle to follow a set trajectory that has been synthetically altered at some points depending on the type of road that has been taken into account. Since the data that is retrieved from Trafikverket's road database is probably for a car, the velocity profile needs to be synthetically changed at some points so that the truck does not rollover. The simulation model has been referred from [12].

3.3.2 Driver Model

The driver model then receives the output from the driving cycle, such as requested longitudinal velocity ($v_{x,\text{req}}$), road curvature (K), and road grade ($-\psi_{ry}$). The driver model is divided into two parts :

- The force request is generated in the model, where the different kinds of resistive forces, like aerodynamic drag, rolling resistance, and grade resistance, are added up with the feedback force, which then gets added up with the feed forward force $m \cdot a_{x,\text{req}}$, which determines the total longitudinal force request. This constitutes the longitudinal control block.
- The other control block constitutes the lateral force, which operates independently of the longitudinal control block and generates a steering wheel angle request based on road curvature and vehicle state inputs, and the angle is divided by the steering ratio to get the required steering wheel angle (δ_{req}) [13].

The computed ($F_{x,\text{req}}$) based on the equations used in [6] is then passed to the motion controller or vehicle motion controller block.

3.3.3 Motion Controller

The motion controller or vehicle motion controller block is bifurcated into two parts:

- Force Request Limiter: ($F_{x,\text{req}}$) is fed as input to the force limiter block, where the requested longitudinal force is limited either by using the EM's capabilities or the available traction on the wheels at the axle level. This limitation can be achieved using the equation expressed below:

$$F_{x,\text{tot,req,sat}} = \min\left(\min(F_{x,\text{tot,req}}, V_{\text{max,Total}}), \mu\left(\sum_{i=1}^m F_{z\text{Ax}i}\right)\right) \quad (3.1)$$

where $F_{z,\text{Ax}i}$ denotes the vertical force acting on the axle.

$V_{\text{max,Total}}$ is the total of the maximum force request limits of the startability axle, cruise axle, and also the maximum force limits of the electric machines. This equation holds valid if the force request is greater than 0, i.e., acceleration mode.

While if the force request is less than 0, i.e., braking mode, then the equation for saturated longitudinal force request is given by:

$$F_{x,\text{tot,req,sat}} = \max\left(\max(F_{x,\text{tot,req}}, V_{\text{min,Total}}), -\mu\left(\sum_{i=1}^m F_{z\text{Ax}i}\right)\right) \quad (3.2)$$

where $V_{\text{min,Total}}$ is the total of the minimum force request limits of the startability axle, cruise axle, and also the maximum force limits of the electric machines.

Once the force is limited, it is sent to the wheel coordination strategy block, and then the force request is distributed between the axles according to the throttle demand.

- **Coordination Strategies** The primary objective of the coordination strategies block is to allocate the torque to the available actuators, based on the different types of wheel coordination strategies.

3.3.4 Actuators

The actuator force requests coming out as output from the wheel coordination block are converted to wheel torque requests using the following formulas for the two electric machines:

$$T_{EM, stb} = F_{x, req, stb} \cdot r_{wh} \cdot gr_{stb} \cdot \left(0.99^{\frac{gr_{stb}}{3}}\right) \quad (3.3)$$

$$T_{EM, crs} = F_{x, req, crs} \cdot r_{wh} \cdot gr_{crs} \cdot \left(0.99^{\frac{gr_{crs}}{3}}\right) \quad (3.4)$$

where gr_{stb} and gr_{crs} represent the gear ratios for the startability and cruise axles, having values of 23 and 14, respectively. The power of each electric machine is limited to 300 kW, and r_{wh} denotes the radius of the wheel attached to the corresponding axle.

The friction brake actuators were modeled to accommodate the braking force request to the friction brakes coming from actuator coordination, and the braking torque can be formulated as :

$$T_{req, brk, i} = F_{x, req, brk, i} \cdot r_{w, i} \quad (3.5)$$

3.3.5 Vehicle Dynamics / VTM Model

The VTM Model, also known as the Volvo Transport Model, is a high-fidelity simulation environment with detailed vehicle models developed by Volvo GTT (Group Trucks Technology) in-house in order to study vehicle dynamics and assist in control development. It also assists in performing simulations of heavy vehicles such as rigid trucks, tractor-semitrailer, or other combinations. The model library consists of the tyre, suspension, brakes, chassis, etc. It can also be used in cases where the vehicle is in the early stages of R&D, which can be validated for real-life driving conditions.



Figure 3.7: VTM Animation of a 4x4 rigid truck

3.3.6 Sim Stopper

In the model, two logic blocks were implemented, which automatically terminate the simulation if any of the conditions are met. They are as follows:

- Rollover Stopper: When the rollover angle of the truck exceeds a value of 1 radian, then the simulation is terminated.
- Distance and Speed Stopper: In this logic, the distance travelled, and the actual velocity (longitudinal) are monitored, and the simulation is stopped if either of the conditions are met.
 - If the distance exceeds the required distance, indicating that the driving cycle has been completed.
 - On the other hand, if the $v_{x,\text{act}}$ (actual longitudinal velocity) remains below 0.80 m/s for a time period of 4 seconds, which would indicate that the vehicle is stalling at a very low speed (close to zero).

This is to ensure that all the algorithms have similar travel time and are affected by the integration of the simulation.

4

Control Strategies

This study focuses on the evaluation and comparison of three control strategies aimed at improving the energy efficiency of battery electric vehicles under typical driving conditions, such as country road and highway cruising. These strategies are designed to operate primarily during normal driving scenarios, where smooth and efficient torque coordination is crucial.

The first two strategies are formulated to minimize the instantaneous power losses across the electric actuators. These loss-minimization approaches are benchmarked against a third strategy, which instead prioritizes equal utilization of available tyre-road friction across all wheels. By distributing traction demands more evenly, the third approach aims to improve stability and extend component life.

All three strategies are implemented as closed-loop systems. They receive real-time feedback from actuator states and vehicle dynamics, as provided by the high-fidelity VTM Model. This enables the controllers to adapt to changing conditions and respond accurately to the vehicle's behavior throughout the driving cycle.

The three control allocation strategies that are being considered in the thesis are discussed below:

4.1 Power Loss Minimisation (PLM)

The Power Loss Minimization (PLM) algorithm is designed to minimize the instantaneous power losses across the vehicle's actuators, while simultaneously satisfying the longitudinal motion requirements. This is achieved under strict adherence to physical constraints imposed by the actuators and the tire contact patch.

This optimization-based control approach ensures that the propulsion demands are met efficiently, without violating the operational boundaries of the system components.

In this wheel torque coordination strategy, losses from the electrical drivetrain (losses generated due to the conversion of electrical energy to mechanical energy) which are lost as heat are minimised whilst the other losses like aerodynamic drag, rolling resistance, tyre slip are only calculated.

Control allocation refers to the process of solving a control problem in a system with more actuators than required or an over-actuated system and distributing the control inputs across several actuators to achieve a desired set of target outputs.

It is considered to be an over-actuated task if the vehicle equipped with electric machines is required to distribute torque to all the wheels separately. Thus, the implementation of a wheel torque control allocation strategy is an ideal solution to optimize wheel torque distribution across the wheels [2], [14–16].

The global longitudinal force request, $F_{x,\text{req}} = v_{\text{request}}$ is sent as an input to the coordination strategies block from the vehicle motion controller block. The optimal actuator coordination among the startability, cruise axles, and friction brakes is formulated as a control coordination problem to minimize power losses, as shown in the following equation [12]:

$$\mathbf{u}^* = \arg \min_{\mathbf{u}} \left(P_{\text{lossEM},\text{stb}} + P_{\text{lossEM},\text{crs}} + \sum_{i=1}^n P_{\text{lossbrk},i} \right) \quad (4.1)$$

where n = number of axles,

subject to:

$$\mathbf{B} \cdot \mathbf{u} = \mathbf{v}_{\text{request}}$$

$$\mathbf{u}_{\min} \leq \mathbf{u} \leq \mathbf{u}_{\max}$$

where,

$$\mathbf{u}_{\min} = [F_{x,\text{EM},\text{crs},\min} \quad F_{x,\text{EM},\text{stb},\min} \quad F_{x,\text{brk},1,\min} \quad F_{x,\text{brk},2,\min}]^T = \begin{bmatrix} F_{x,\text{EM},\text{crs},\min} \\ F_{x,\text{EM},\text{stb},\min} \\ F_{x,\text{brk},1,\min} \\ F_{x,\text{brk},2,\min} \end{bmatrix} \quad (4.2)$$

$$\mathbf{u}_{\max} = [F_{x,\text{EM},\text{crs},\max} \quad F_{x,\text{EM},\text{stb},\max} \quad F_{x,\text{brk},1,\max} \quad F_{x,\text{brk},2,\max}]^T = \begin{bmatrix} F_{x,\text{EM},\text{crs},\max} \\ F_{x,\text{EM},\text{stb},\max} \\ F_{x,\text{brk},1,\max} \\ F_{x,\text{brk},2,\max} \end{bmatrix} \quad (4.3)$$

$$-\mathbf{h} \leq \mathbf{G} \cdot \mathbf{u} \leq \mathbf{h} \quad (4.4)$$

From equation 4.1, $\mathbf{u}^*$ is the optimal requests - force request to the actuators. The total force required to achieve the desired vehicle motion is formulated as an equality constraint, ensuring that the sum of individual wheel forces matches the demanded longitudinal force. The permissible force range at each wheel, influenced by factors such as friction, vertical load, longitudinal and lateral forces or to summarize, a friction circle limit is modeled as an inequality constraint. Finally, the limitations of the actuators themselves such as their torque and power capacities, are enforced through box constraints.

Equations 4.2 and 4.3 represent the minimum and maximum capabilities of the actuators so that they operate within their limits and never exceed it.

To numerically solve the quadratic problem, the power losses $P_{\text{lossEM},\text{crs}}$ and $P_{\text{lossEM},\text{stb}}$, of the electric machines are expressed as second-order polynomials.

$$P_{\text{lossEM},\text{crs}} = a_{\text{EM},\text{crs}} \cdot T_{\text{EM},\text{crs}}^2 + b_{\text{EM},\text{crs}} \cdot T_{\text{EM},\text{crs}} + c_{\text{EM},\text{crs}} \quad (4.5)$$

$$P_{\text{lossEM},\text{stb}} = a_{\text{EM},\text{stb}} \cdot T_{\text{EM},\text{stb}}^2 + b_{\text{EM},\text{stb}} \cdot T_{\text{EM},\text{stb}} + c_{\text{EM},\text{stb}} \quad (4.6)$$

where, $a_{\text{EM},\text{crs}}$, $b_{\text{EM},\text{crs}}$, and $c_{\text{EM},\text{crs}}$ and $a_{\text{EM},\text{stb}}$, $b_{\text{EM},\text{stb}}$, and $c_{\text{EM},\text{stb}}$ are the curve fitting coefficients for the cruise and startability machines.

An important aspect to consider in the case of PLM is that the coefficient $c_{\text{EM},\text{stb}}$ or $c_{\text{EM},\text{crs}}$ are not used as this term is only valid in the case of idling losses at zero torque which will be discussed in the next strategy.

Also, the brakes losses are estimated and defined in section 2.5.

Equations 4.7 and 4.8 highlight the equations for finding the wheel force limits so that the forward motion of the vehicle stays within its limits while considering the lateral and vertical forces for each axle.

$$h = \sum_{i=1}^n \sqrt{(F_{x,\text{lim},i})^2 - (F_{y,\text{lim},i})^2} \quad (4.7)$$

where,

$$F_{x,\text{lim},i} = \mu \cdot F_{z,i} \quad (4.8)$$

and $F_{y,\text{lim},i}$ is the lateral force limit of the i -th axle.

The optimisation problem which has been discussed earlier in equations 4.1 - 4.8, is later reformulated as a minimisation problem which is shown as below:

$$\mathbf{u}^* = \arg \min_{\mathbf{u}} \left(\frac{1}{2} \mathbf{u}^T \mathbf{H} \mathbf{u} + \mathbf{g}^T \mathbf{u} \right) \quad (4.9)$$

subject to:

$$\mathbf{B} \cdot \mathbf{u} = \mathbf{v}_{\text{request}}$$

$$\mathbf{u}_{\min} \leq \mathbf{u} \leq \mathbf{u}_{\max}$$

$$\mathbf{G} \cdot \mathbf{u} \leq \mathbf{h}$$

4.1.1 4x4 Configuration

For a 4x4 rigid truck, the B matrix would be represented as:

$$\mathbf{B} = \begin{bmatrix} 1 & 1 & 1 & 1 \end{bmatrix} \quad (4.10)$$

The presence of 1 means that the actuators (both electric machines and all two friction brakes present on each axle) are being used to their full capacity .

The matrix $\mathbf{G}$ in equation 4.11 represents the availability of actuators on each axle and is composed of 0's and 1's for a 4x4 rigid truck, where a value of 0 indicates the absence of an actuator at a given position, and a value of 1 denotes its presence.

$$\mathbf{G} = \begin{bmatrix} -1 & 0 & -1 & 0 \\ 0 & -1 & 0 & -1 \\ 1 & 0 & 1 & 0 \\ 0 & 1 & 0 & 1 \end{bmatrix} \quad (4.11)$$

$$\mathbf{H} = 2 \cdot \begin{bmatrix} \frac{a_{\text{EM},\text{crs}} \cdot r_w^2}{gr_1^2} & 0 & 0 & 0 \\ 0 & \frac{a_{\text{EM},\text{stb}} \cdot r_w^2}{gr_2^2} & 0 & 0 \\ 0 & 0 & a_{\text{brk},1} & 0 \\ 0 & 0 & 0 & a_{\text{brk},2} \end{bmatrix} \quad (4.12)$$

where H is the hessian matrix

$$\mathbf{g}^T = \begin{bmatrix} \frac{b_{\text{EM},\text{crs}} \cdot v_{w,1}}{gr_c r_s} & \frac{b_{\text{EM},\text{stb}} \cdot v_{w,2}}{gr_s t_b} & -\nu_{\text{xact},1} & -\nu_{\text{xact},2} \end{bmatrix} \quad (4.13)$$

$$\mathbf{v}_{request} = F_{x,req}, \quad \mathbf{h} = \begin{bmatrix} F_{x,lim,1} \\ F_{x,lim,2} \\ F_{x,lim,1} \\ F_{x,lim,2} \end{bmatrix}, \quad \mathbf{u} = \begin{bmatrix} F_{x,req,EM,crs} \\ F_{x,reqEM,stb} \\ F_{x,req,brk,1} \\ F_{x,req,brk,2} \end{bmatrix} \quad (4.14)$$

In equations 4.12 and 4.13, longitudinal velocities at the axle level for a 4×4 rigid truck are represented by $v_{xact,1}$ and $v_{xact,2}$, while the braking components such as $a_{brk,1}$ and $a_{brk,2}$ are considered to be very small, on the order of 10^{-5} .

The term u represents the force allocation vector containing the F_x of the electric machines at the first and second axles ($F_{x,req,EM,crs}$, $F_{x,req,EM,stb}$), and the F_x of the friction brakes ($F_{x,req,brk,1}$, $F_{x,req,brk,2}$).

Later in the optimisation process, the force requests are converted to wheel torque requests using the following formulas for the two electric machines:

$$T_{EM,stb} = F_{x,req,stb} \cdot r_{wh} \cdot gr_{stb} \cdot \left(0.99^{\frac{gr_{stb}}{3}}\right) \quad (4.15)$$

$$T_{EM,crs} = F_{x,req,crs} \cdot r_{wh} \cdot gr_{crs} \cdot \left(0.99^{\frac{gr_{crs}}{3}}\right) \quad (4.16)$$

4.1.2 6x4 Configuration

For a 6x4 rigid truck, the matrices can be represented as below:

$$\mathbf{u}_{min} = \begin{bmatrix} F_{x,EM,stb,min} & F_{x,EM,crs,min} & F_{x,brk,1,min} & F_{x,brk,2,min} & F_{x,brk,3,min} \end{bmatrix}^T = \begin{bmatrix} F_{x,EM,stb,min} \\ F_{x,EM,crs,min} \\ F_{x,brk,1,min} \\ F_{x,brk,2,min} \\ F_{x,brk,3,min} \end{bmatrix} \quad (4.17)$$

$$\mathbf{u}_{max} = \begin{bmatrix} F_{x,EM,stb,max} & F_{x,EM,crs,max} & F_{x,brk,1,max} & F_{x,brk,2,max} & F_{x,brk,3,max} \end{bmatrix}^T = \begin{bmatrix} F_{x,EM,stb,max} \\ F_{x,EM,crs,max} \\ F_{x,brk,1,max} \\ F_{x,brk,2,max} \\ F_{x,brk,3,max} \end{bmatrix} \quad (4.18)$$

$$\mathbf{B} = \begin{bmatrix} 1 & 1 & 1 & 1 & 1 \end{bmatrix} \quad (4.19)$$

$$\mathbf{G} = \begin{bmatrix} -1 & 0 & 0 & -1 & 0 \\ 0 & -1 & 0 & 0 & -1 \\ 0 & 0 & 1 & 0 & 0 \\ 1 & 0 & 0 & 1 & 0 \\ 0 & 1 & 0 & 0 & 1 \end{bmatrix} \quad (4.20)$$

$$H = 2 \cdot \begin{bmatrix} \frac{a_{EM, stb} \cdot r_w^2}{gr_1^2} & 0 & 0 & 0 & 0 \\ 0 & \frac{a_{EM, crs} \cdot r_w^2}{gr_2^2} & 0 & 0 & 0 \\ 0 & 0 & a_{brk,1} & 0 & 0 \\ 0 & 0 & 0 & a_{brk,2} & 0 \\ 0 & 0 & 0 & 0 & a_{brk,3} \end{bmatrix} \quad (4.21)$$

$$g^T = \begin{bmatrix} \frac{b_{EM, stb} \cdot v_{w,1}}{gr_{stb}} & \frac{b_{EM, crs} \cdot v_{w,2}}{gr_{crs}} & -\nu_{xact,1} & -\nu_{xact,2} & -\nu_{xact,3} \end{bmatrix} \quad (4.22)$$

$$h = \begin{bmatrix} F_{x,lim,1} \\ F_{x,lim,2} \\ F_{x,lim,1} \\ F_{x,lim,2} \\ F_{x,lim,3} \end{bmatrix}, \quad u = \begin{bmatrix} F_{x,req,EM,stb} \\ F_{x,req,EM,crs} \\ F_{x,req,brk,1} \\ F_{x,req,brk,2} \\ F_{x,req,brk,3} \end{bmatrix} \quad (4.23)$$

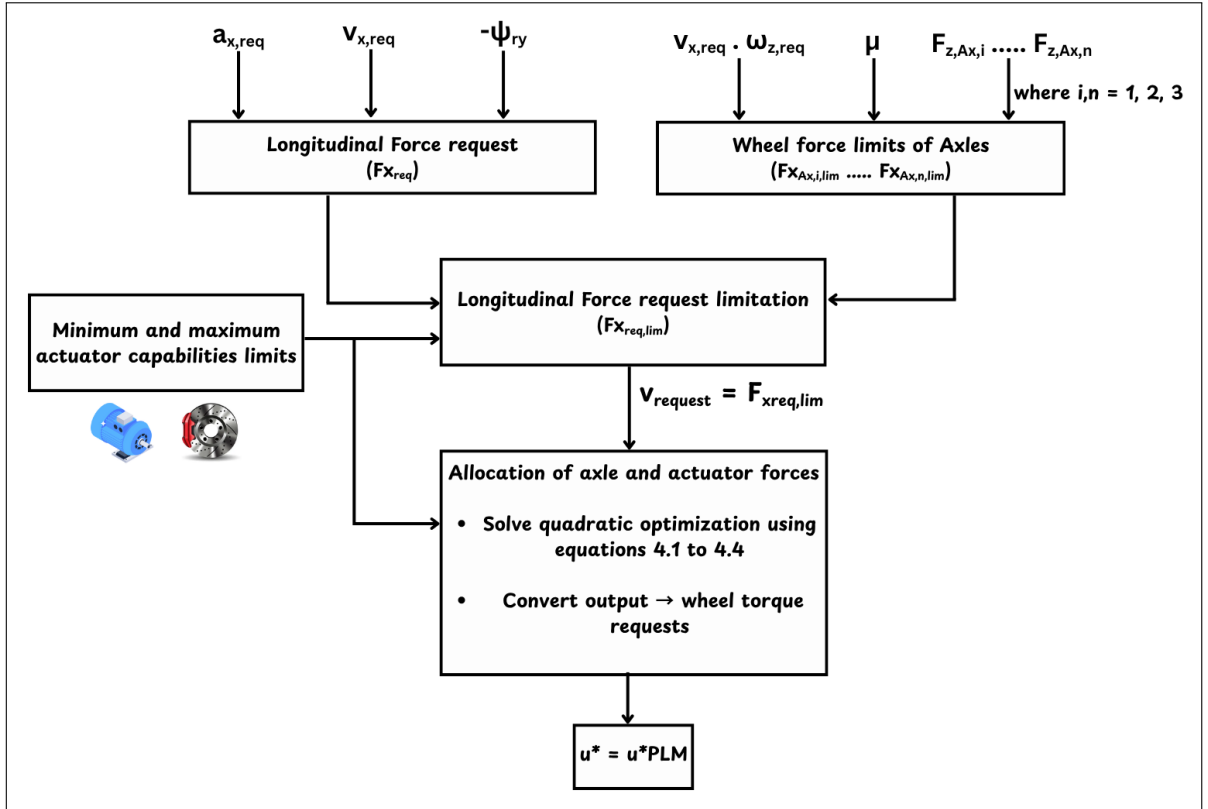


Figure 4.1: Overview of PLM Algorithm

Figure 4.1 illustrates the working flow of the PLM algorithm. Initially, the driver model receives inputs from the speed profile, such as the requested vehicle speed ($v_{x,req}$) and the road grade. These inputs are then passed to the F_x limiter block, where the longitudinal forces are constrained. The limited forces are subsequently distributed to the axles. As described earlier, the optimization problem is then solved, resulting in the required wheel torque requests.

To achieve higher efficiency and reduce power losses, a clutch mechanism needs to be introduced to disengage the induction motor (IM) when not in use, thereby minimizing idling losses. However, this functionality cannot be implemented within the current optimization framework. Therefore, the terms $c_{EM, stb}$ and $c_{EM, crs}$ are introduced in the subsequent strategy, which will be discussed later.

4.2 Power Loss Minimisation including Idle Losses (ILPLM)

The ILPLM strategy extends the baseline PLM algorithm by incorporating idle (zero-torque) losses into the torque allocation process / wheel torque coordination strategy. This enhancement enables more energy-efficient allocation, particularly during partial load conditions. The terms $c_{EM, stb}$ and $c_{EM, crs}$ are introduced in this optimisation method for the losses at zero torque.

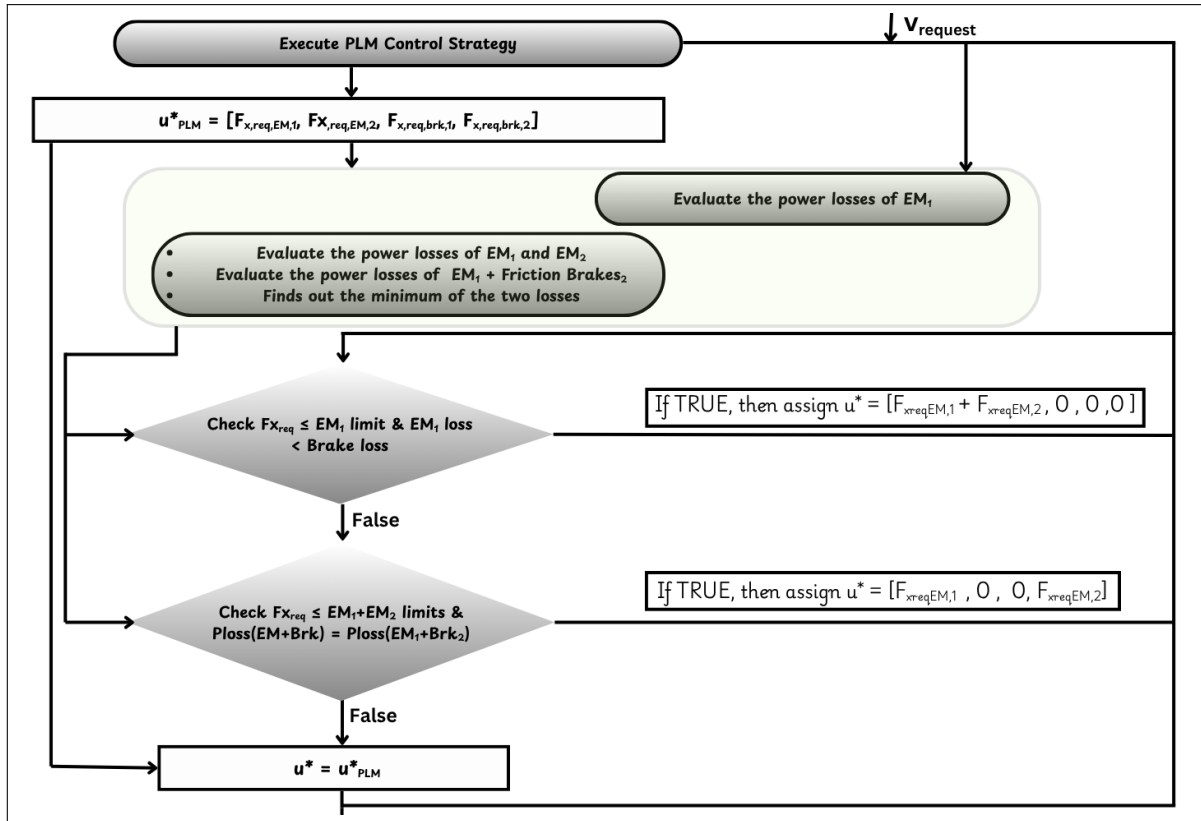


Figure 4.2: Overview of ILPLM Algorithm

Figure 4.2 represents the working of the ILPLM algorithm. Firstly, the PLM algorithm runs continuously to estimate the power losses of the EMs, using curve-fitted efficiency maps. The keywords EM_1 and EM_2 represent the PMSM and IM machines, while Friction Brakes₁ and Friction Brakes₂ indicate the friction brakes on the cruise and startability axes.

Secondly, to maximize the usage of the cruise axle, for each longitudinal force request, $F_{x, req}$ (longitudinal force request), the algorithm first checks if the requested demand can be fulfilled by the cruise axle alone.

The effective limit is dictated by choosing the lesser value or by taking the minimum between the force availability of PMSM and the friction circle limits of the particular axle.

If the requested force does not exceed the axle limits and remains within the bounds of the cruise axle, the ILPLM evaluates the power losses for the following two configurations:

- PMSM-only operation on the cruise axle.
- Combined operation of the PMSM on the cruise axle and friction braking on the startability axle.

The configuration that results in the lower total power loss is selected.

If $F_{x,\text{req}}$ exceeds the cruise axle capability, the algorithm evaluates whether the combined capacity of the cruise and startability axles is sufficient to distribute the requested torque at the wheels without violating the limits of EMs and friction circle. In this case, two additional combinations are compared:

- PMSM on the cruise axle with friction brakes on the startability axle.
- PMSM on the cruise axle with the IM on the startability axle.

The strategy then selects the actuator combination that results in the minimum total power loss. If the force request does not match any predefined efficient allocation, the task is passed to the original PLM algorithm for optimization based on the actuator constraints.

Through this multi-layered logic, the ILPLM strategy ensures that actuator selection is both feasible and energy-optimal, accounting for active and idle losses across all propulsion and braking components.

4.3 Equal Friction Utilisation (EFU)

The EFU strategy serves as the benchmark for evaluating the PLM and ILPLM strategies.

The EFU algorithm focuses on achieving balanced utilization of the available friction at the tyre and road interface of each wheel. This balance is based on the normal (vertical) load acting on each wheel at axle level, ensuring that the traction demand is distributed utilizing equal friction between the wheels [17].

For a two-axle vehicle (4x2 or 4x4) or even a three-axle vehicle (6x2 or 6x4), equal friction utilization is defined by the condition that the ratio of longitudinal force to available friction force is the same at each n th axle. Mathematically, this condition and the total force requirement are expressed as:

$$\frac{F_{x1}}{\mu F_{z1}} = \frac{F_{x2}}{\mu F_{z2}} = \dots = \frac{F_{xn}}{\mu F_{zn}} \quad (4.24)$$

where μ is the coefficient of friction between the road and tyre, $F_{x1}, F_{x2}, \dots, F_{xn}$ are the longitudinal forces on the 1st, 2nd and n th axles, while $F_{z1}, F_{z2}, \dots, F_{zn}$ are the normal loads on the 1st, 2nd and n th axles.

$$F_{x,\text{req}} = F_{x,\text{req}1} + F_{x,\text{req}2} + \dots + F_{x,\text{req}n} \quad (4.25)$$

where $F_{x,\text{req}1}, F_{x,\text{req}2}, \dots, F_{x,\text{req}n}$ are the longitudinal force requests from the throttle pedal by the driver at first, second, and n th axles respectively and $F_{x,\text{req}}$ is the total force request at the axle level.

The algorithm has been designed in such a way that when the actuators (electric machines or friction brakes) exceed the wheel force limits or the friction limits, then the remaining force limits are sent to the other axle. Therefore, this algorithm is not optimal for power loss.

In general, the force requests at the axle are first fulfilled by the electric machines (IM and PMSM), each of which is placed on each axle until limited (saturated) by their capability limits. The extra force was utilised by the friction brakes present on each axle.

In our study, we have used 4x4 and 6x4 configurations and we have different force distribution strategies.

4.3.1 4x4 Configuration

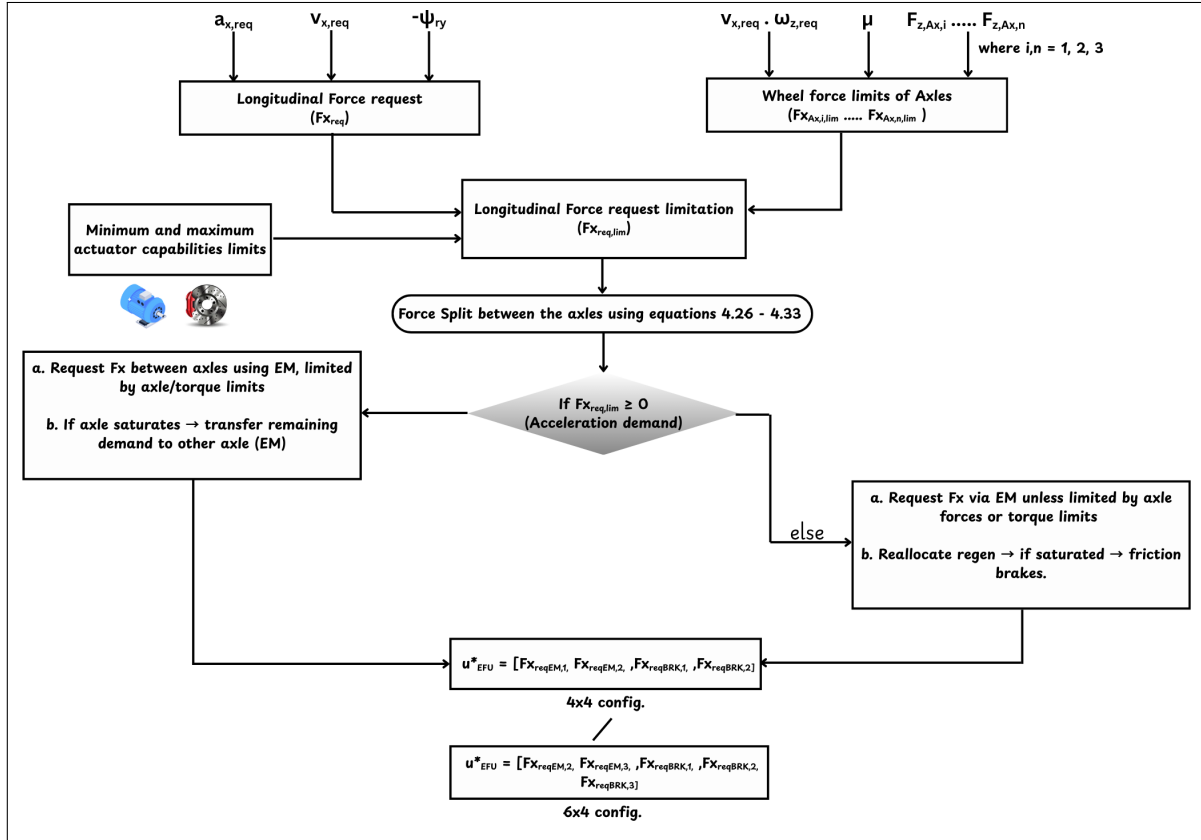


Figure 4.3: ECU algorithm for 4x4 / 6x4 configuration

Figure 4.3 highlights the ECU algorithm for a 4x4 / 6x4 configuration. It can be seen that longitudinal velocity and acceleration are the inputs to the driver model where we get the longitudinal force request as the output from the driver model but is fed as input to the force request limitation block along with other inputs like coefficient of friction, normal loads acting on each axle which gets limited by the wheel force limits. Furthermore, if we have an excess longitudinal force request, then the longitudinal force is limited based on the electric machine and axle wheel force limits which is expressed as below:

$$F_{x,tot,req,sat} = \min \left(\min(F_{x,tot,req}, V_{max,Total}), \mu (F_{z,Ax11} + F_{z,Ax12}) \right) \quad (4.26)$$

where $V_{max,Total}$ is the total of the maximum force request limits of the startability axle, cruise axle.

This equation holds if the force request is greater than 0, i.e., acceleration mode.

While if the force request is less than 0, i.e., braking mode, then the equation for saturated longitudinal force request is given by:

$$F_{x,\text{tot,req,sat}} = \max\left(\max(F_{x,\text{tot,req}}, V_{\min,\text{Total}}), -\mu(F_{z,\text{Ax11}} + F_{z,\text{Ax12}})\right) \quad (4.27)$$

where $V_{\min,\text{Total}}$ is the total of the minimum force request limits of the startability axle, cruise axle, and also the maximum regeneration force limits of the electric machines.

For a 4x4 configuration, the force request at both axles can be represented as:

$$F_{x_{\text{req}},1} = \frac{F_{z,1}}{F_{z,1} + F_{z,2}} \cdot F_{x_{\text{req}}} \quad (4.28)$$

$$F_{x_{\text{req}},2} = F_{x_{\text{req}}} - F_{x_{\text{req}},1} \quad (4.29)$$

As shown in Fig 4.3, it can be seen that the saturated longitudinal forces are split per axle on which the EMs are mounted following certain set of conditions in acceleration and braking mode and the outputs of the EMs and friction or foundation brakes are channelized through u_{EFU}^*

4.3.2 6x4 Configuration

As seen in Figure 4.3, the inputs for the 6x4 configuration are similar to those of the 4x4. The normal loads, longitudinal forces for the third axle have been introduced. Since the front axle is not propelled, then there is no longitudinal force request at the front axle is zero. The rear two axles are only equipped with EMs hence only those two axles will have $F_{x,\text{req}}$. The saturated longitudinal force requests for a 6x4 rigid truck can be expressed as:

$$F_{x,\text{tot,req,sat}} = \min\left(\min(F_{x,\text{tot,req}}, V_{\max,\text{Total}}), \mu(F_{z,\text{Ax11}} + F_{z,\text{Ax12}} + F_{z,\text{Ax13}})\right) \quad (4.30)$$

$$F_{x,\text{tot,req,sat}} = \max\left(\max(F_{x,\text{tot,req}}, V_{\min,\text{Total}}), -\mu(F_{z,\text{Ax11}} + F_{z,\text{Ax12}} + F_{z,\text{Ax13}})\right) \quad (4.31)$$

For splitting $F_{x,\text{tot,req,sat}}$ into the respective axle forces during acceleration mode, the force is first given to the one of the axles then the overflow of the demanded force goes to the other axle.

During regeneration or braking mode, the EM axles are prioritized. However, if both rear axle EM brakes (rear axle group) are saturated, then the surplus braking force goes to the foundation or friction brakes in the order of 1st-2nd-3rd axle.

The force request at both axles can be represented as:

$$F_{x_{\text{req}},2} = \frac{F_{z,2}}{F_{z,2} + F_{z,3}} \cdot F_{x_{\text{req}}} \quad (4.32)$$

$$F_{x_{\text{req}},3} = \frac{F_{z,3}}{F_{z,2} + F_{z,3}} \cdot F_{x_{\text{req}}} \quad (4.33)$$

5

Results

This section reflects on the results that were obtained by simulating all three coordination strategies - EFU, PLM and ILPLM on different roads ranging from country roads to a highway road with the different values of coefficient of friction from 0.80 to 0.40. The results show the comparison between the strategies and how the strategies distribute the forces among the actuators as discussed in Section 4. The goal is not only to minimise the energy losses, find safety metrics but also to bring about the most efficient coordination strategy.

Table 5.1: Configuration table for different roads with different coefficients of friction

Road	Config	Algorithm	Coefficient of Friction (μ)				
			0.80	0.70	0.60	0.50	0.40
Road E20	4x4	EFU	x	x	x	x	x
		PLM	x	x	x	x	x
		ILPLM	x	x	x	x	x
	6x4	EFU	x	x	x	x	x
		PLM	x	x	x	x	x
		ILPLM	x	x	x	x	x
Road R1758	4x4	EFU	x	x	x	x	x
		PLM	x	x	x	x	x
		ILPLM	x	x	x	x	x
	6x4	EFU	x	x	x	x	x
		PLM	x	x	x	x	x
		ILPLM	x	x	x	x	x
Road R180	4x4	EFU	x	x	x	x	x
		PLM	x	x	x	x	x
		ILPLM	x	x	x	x	x
	6x4	EFU	x	x	x	x	x
		PLM	x	x	x	x	x
		ILPLM	x	x	x	x	x

Table (5.1) shows that the 4x4 rigid truck and the 6x4 rigid truck were working under these conditions and completed the simulation. However, to compare the different strategies the case with 0.40 coefficient of friction is only presented and discussed in the results section.

The results are further divided into three parts, model validation, and the two metrics, lateral stability and energy savings.

5.1 Model Validation

Prior to a detailed analysis of the coordination strategies, the model is validated to check if the proposed coordination strategies are working efficiently and are allocating forces in the expected manner. The strategies were compared using the following : a) simulation time for the travelled distances b) lateral deviation of the vehicle from its expected path c) yaw rate and steering wheel angle over the distance travelled d) Longitudinal force request to the coordination strategies e) wheel force allocation of the strategies for a steady state turning case which represents how the forces are being allocated on powered and non-powered axles.

5.1.1 Distance vs Time plots for 4x4 and 6x4 configuration

Figure 5.1 and Figure 5.2 represent the distance vs. time plots for 4x4 and 6x4 trucks, where it can be seen that the time taken to complete the entire driving cycle is almost the same and differs by 1 second. The different coordination strategies overlap with each other without any deviation, which is considered to be negligible. These small variations are due to the different coordination strategies and road profiles, which in turn alter vehicle motion slightly.

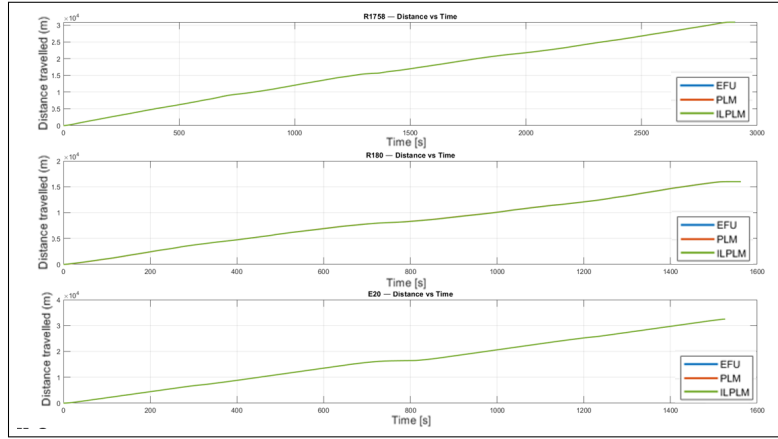


Figure 5.1: Distance vs Time plots for 4x4 configuration

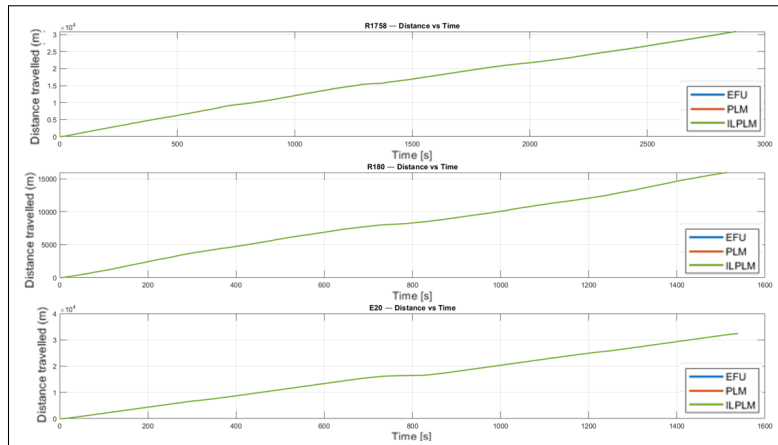


Figure 5.2: Distance vs Time plots for 6x4 configuration

5.1.2 Longitudinal force request to the coordination strategies vs Distance plots for 4x4 and 6x4 configuration

Figure 5.3 and Figure 5.4 represent the relation between the saturated longitudinal force request (F_{x,tot_req_sat}) and the longitudinal resistive forces ($F_{resistance}$) for the different strategies that have been simulated for the different roads. For the vehicle to move forward, it must overcome all the resistance or resistive forces that have been discussed earlier. For instance, if the vehicle has to climb uphill, (F_{x,tot_req_sat}) which is represented by blue solid curve, which in turn is limited by machine limits or friction circle limits, should be higher than the resistive forces ($F_{resistance}$) combined such as rolling resistance, grade resistance, air drag resistance and is represented by green dotted line to propel the vehicle forward.

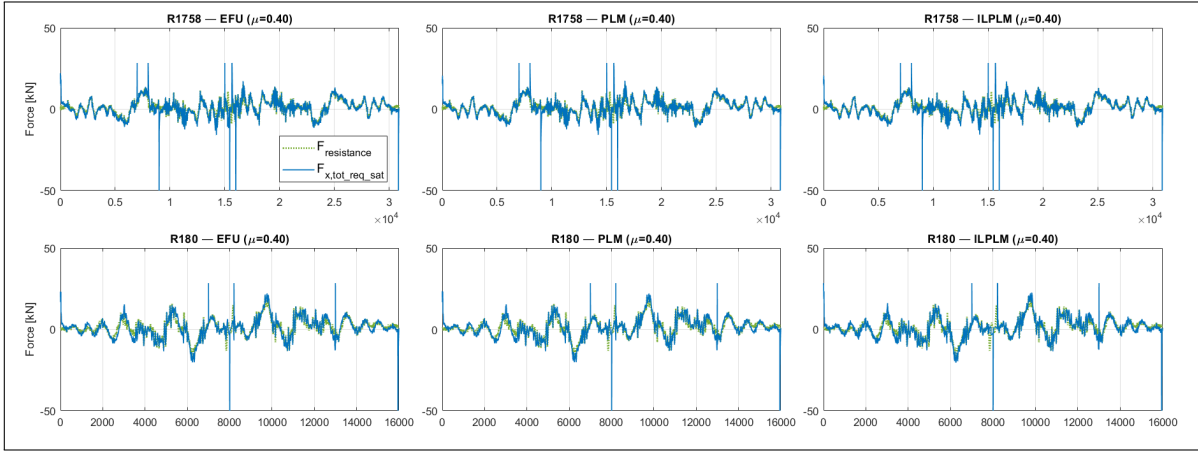


Figure 5.3: Force vs Distance plots for 4x4 configuration

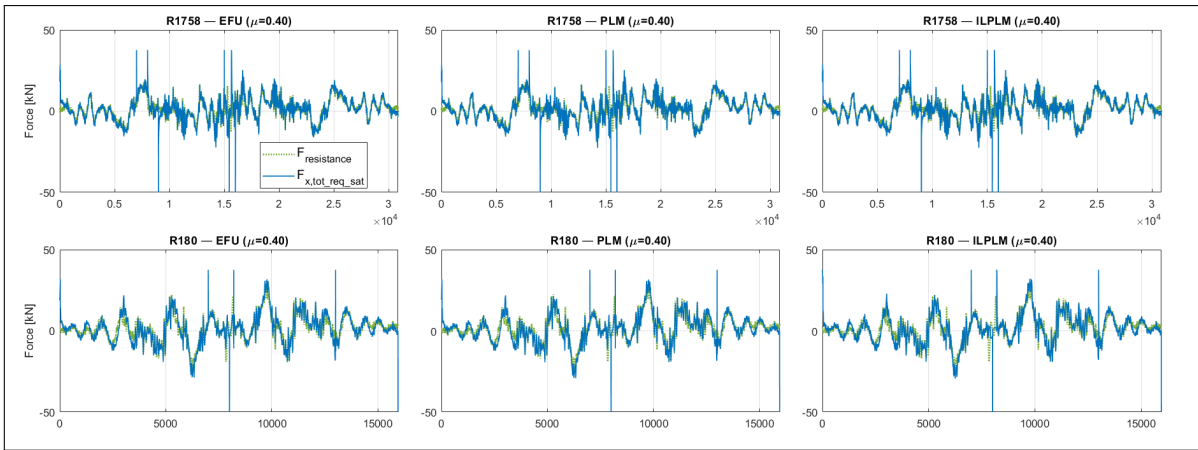


Figure 5.4: Force vs Distance plots for 6x4 configuration

5.1.3 Yawrate and SWA vs Distance plots for 4x4 and 6x4 configuration

In this work, no yaw rate stability algorithm is used; all the allocator forces are directed to the axles, which are later split equally at the wheels by the electric machines and friction brakes. Figure 5.5 represents the yaw rate of the vehicle as a function of distance traveled for all three coordination

strategies. It can be seen that the yaw rate varies from - 0.3 to + 0.3 rad/s. All strategies have identical values as expected.

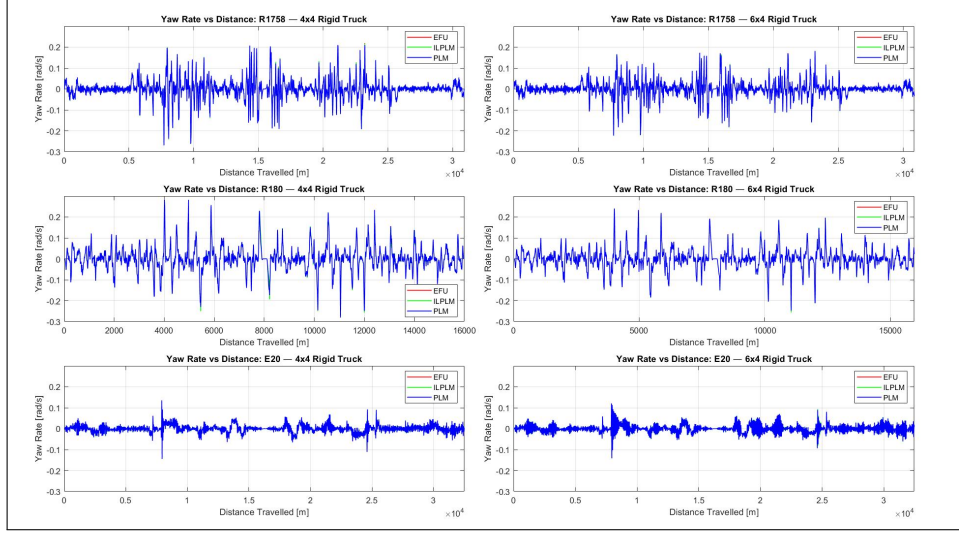


Figure 5.5: Yawrate vs Distance plots for 4x4 and 6x4 configuration

Figure 5.6 represents the steering wheel angle (SWA) for all three strategies. SWA varies from ± 200 degrees at the steering wheel for all strategies of both trucks for all roads. However, the driver model needs more improvement so that it does not counter-steer unnecessarily.

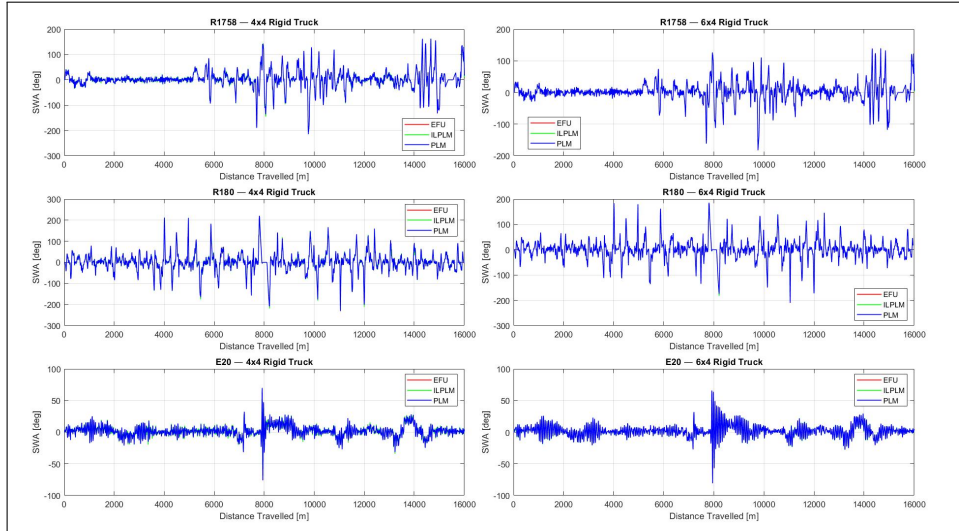


Figure 5.6: Steering angle vs Distance plots for 4x4 and 6x4 configuration

5.1.4 Wheel torque allocation during steady state turning

5.1.4.1 4x4 truck configuration

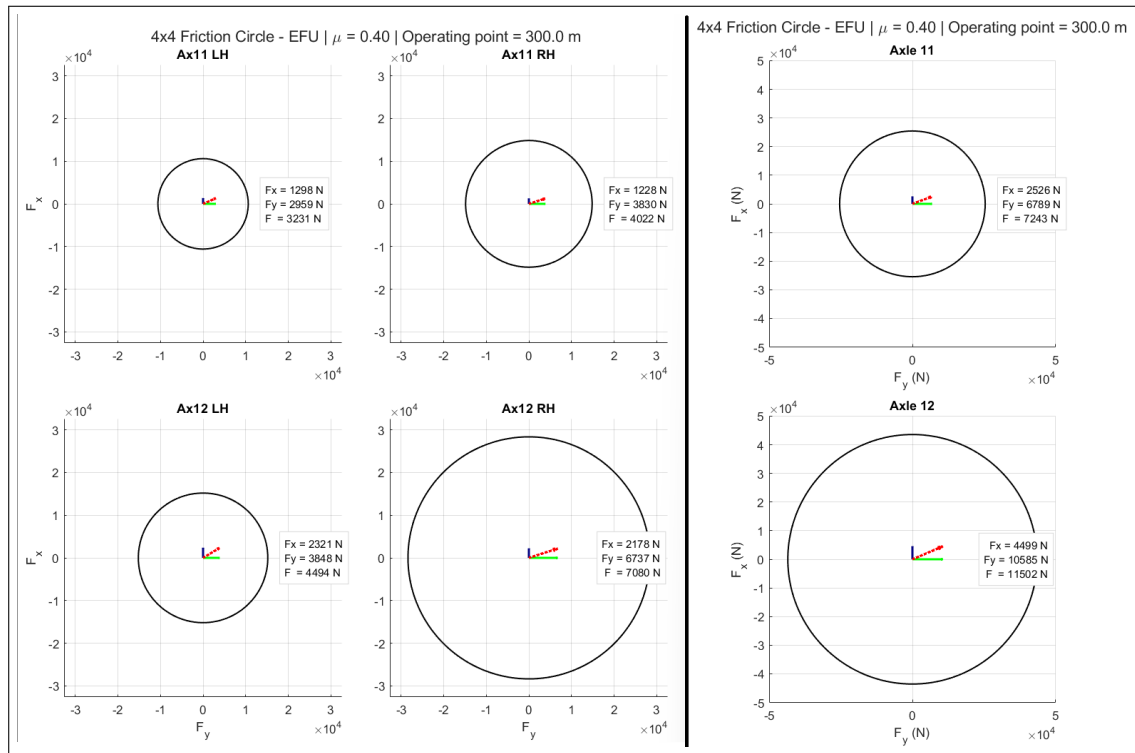
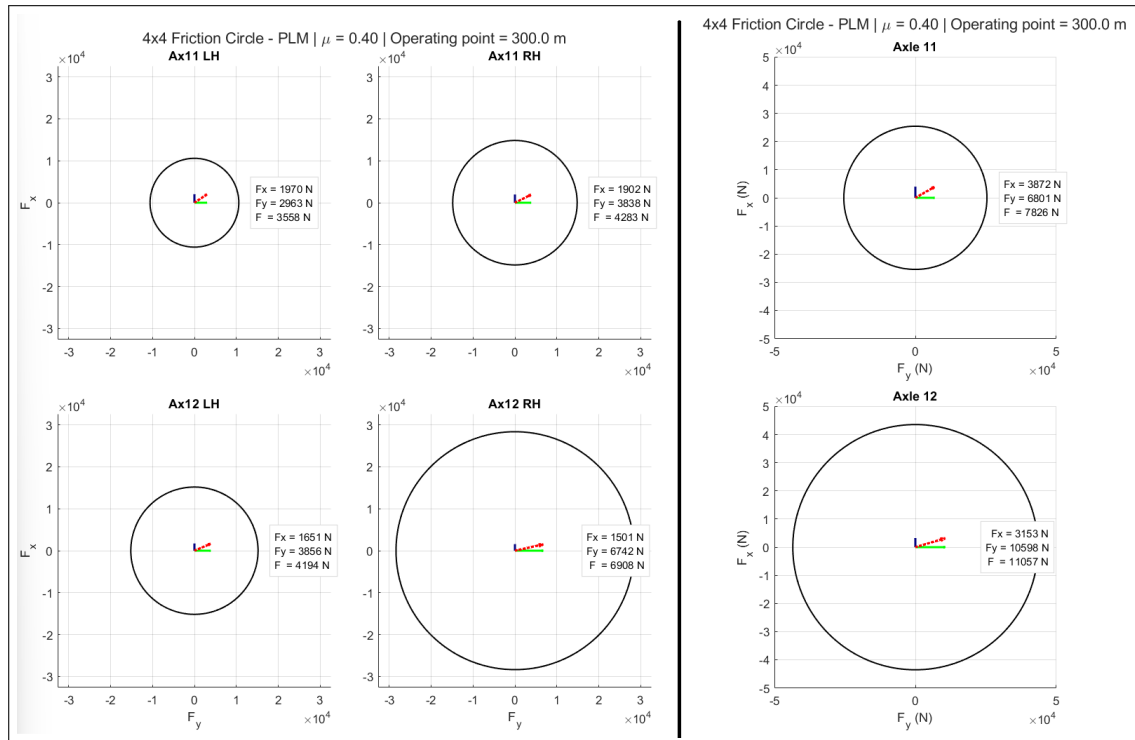
The goal of this test is to validate if the coordination strategies allocate wheel forces in the expected manner which is based on the formulations presented. For this test, a circular road is considered for a short distance of 500 m with a slope of 0.04 radians and a constant radius of curvature of 200 m. The vehicle is driving with a constant longitudinal speed of 14.14 m/s on coefficient of friction of 0.40.

Figures 5.7, 5.8 and 5.9 indicate friction circle plots for different configurations EFU, PLM and ILPLM respectively for 4x4 rigid truck configuration for all the wheels. The limits of the friction circle are denoted by $\mu \cdot Fz_i$, F_x and F_y indicates the longitudinal force and lateral force of a wheel and F is the resultant force.

Friction circle also known as traction circle represents the tire force limits a wheel can generate before losing its grip. These friction circle are plotted at an operating point of 300 m where it has reached a steady state cornering condition.

The radius of the circle is different for the wheels because the vertical/normal load F_z is different for each wheel. In Figure 5.7, 5.8 and 5.9, the friction circles of right wheels of front and rear axle are bigger than those of left wheels which is because the vehicle is turning to the left which causes load transfer between the wheels causing the outer wheels to have more vertical load than the inner wheel which is left wheel. It is clearly visible from the figure that the tires are operating within its grip capacity and there is still unused friction potential which indicates that the vehicle is stable and not sliding.

In Figure 5.9 which are the friction circles of ILPLM algorithm, the rear wheels have negative F_x because of the switching condition in ILPLM. So, in this case only the electric machine of front axle is switched on and electric machine on rear axle is switched off. So, the negative F_x is only due to braking.

Figure 5.7: Friction circle (4x4 rigid, EFU, $\mu = 0.40$)Figure 5.8: Friction circle (4x4 rigid, PLM, $\mu = 0.40$)

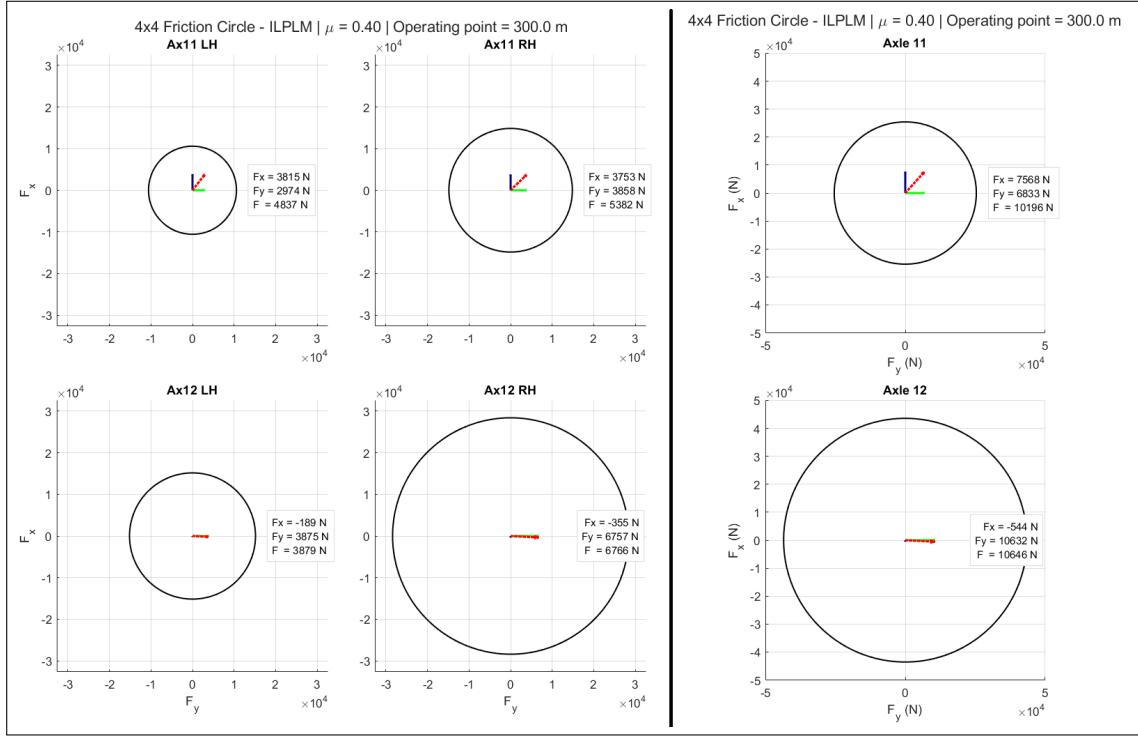


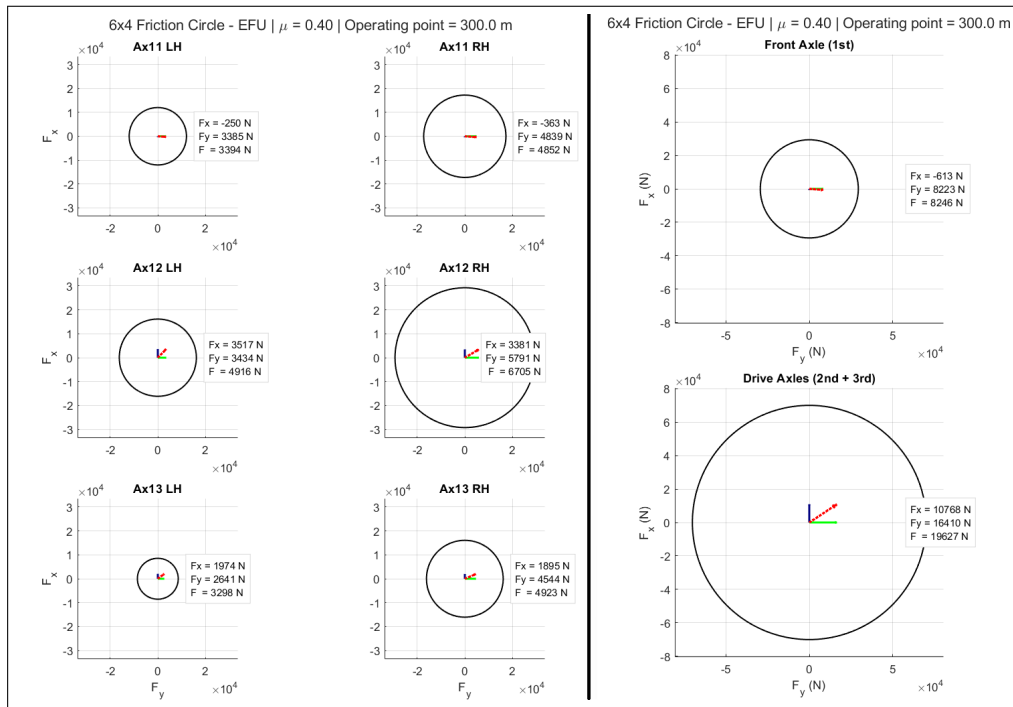
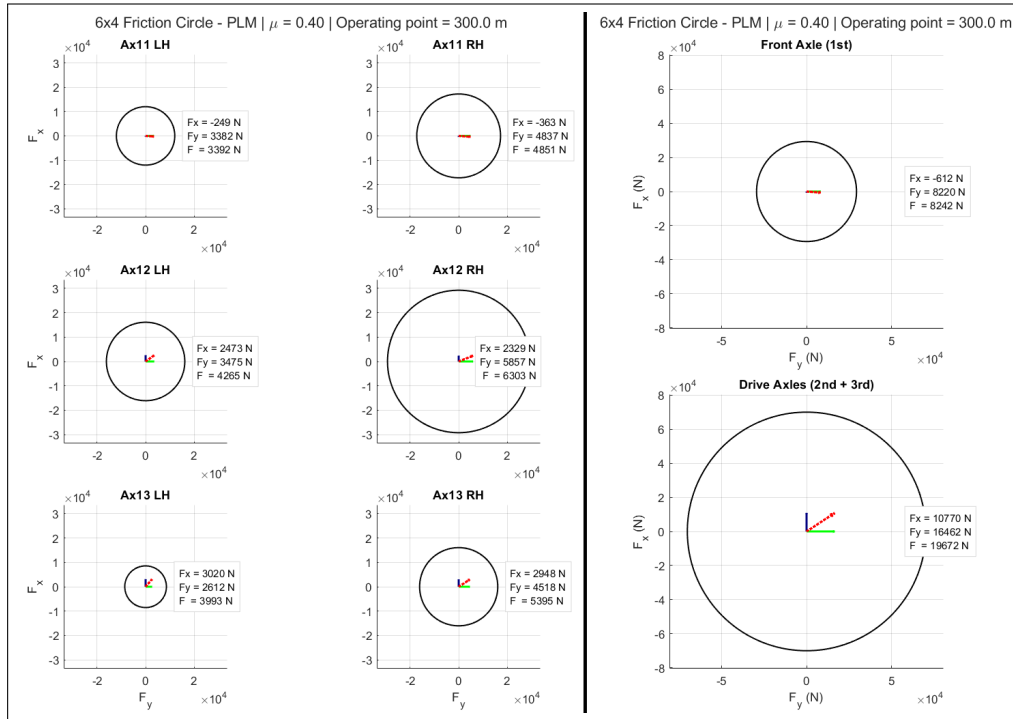
Figure 5.9: Friction circle (4x4 rigid, ILPLM, $\mu = 0.40$)

5.1.4.2 6x4 truck configuration

The same road has been used for 6x4 configuration as the 4x4 configuration. Figures 5.10, 5.11 and 5.12 indicates friction circle plots for different configurations EFU, PLM and ILPLM respectively for 6x4 rigid truck configuration for all the wheels.

It is clearly visible from the figures that the tires are operating within its grip capacity and there is still unused friction potential which indicates that the vehicle is stable and not sliding.

The wheels on the first axle of all the configurations have negative F_x because the first axle is not propelled and there is only braking on it. Similarly, in case of ILPLM configuration, the electric machine on first axle of rear axle group is switched off and second axle of rear axle group is switched on.

Figure 5.10: Friction circle (6x4 rigid, EFU, $\mu = 0.40$)Figure 5.11: Friction circle (6x4 rigid, PLM, $\mu = 0.40$)

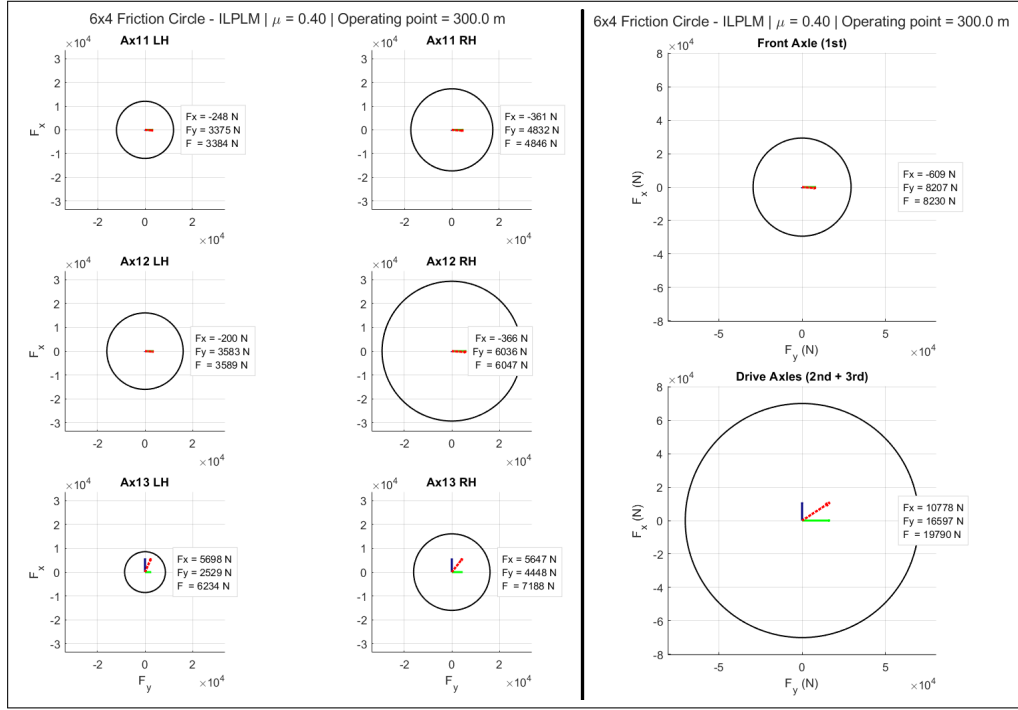


Figure 5.12: Friction circle (6x4 rigid, ILPLM, $\mu = 0.40$)

5.1.5 Lateral deviation vs Distance travelled for 6x4 rigid truck configuration

Figure 5.13 represents the lateral deviation of the vehicle's trajectory for the intended path and has been simulated on all the roads for all three coordination strategies. The driver model is slightly aggressive and causes a deviation of approximately ± 1 m for the country roads R1758 and R180 as they have tighter curves with more steering input, and ± 0.2 m for the straight highway road E20 with less steering corrections.

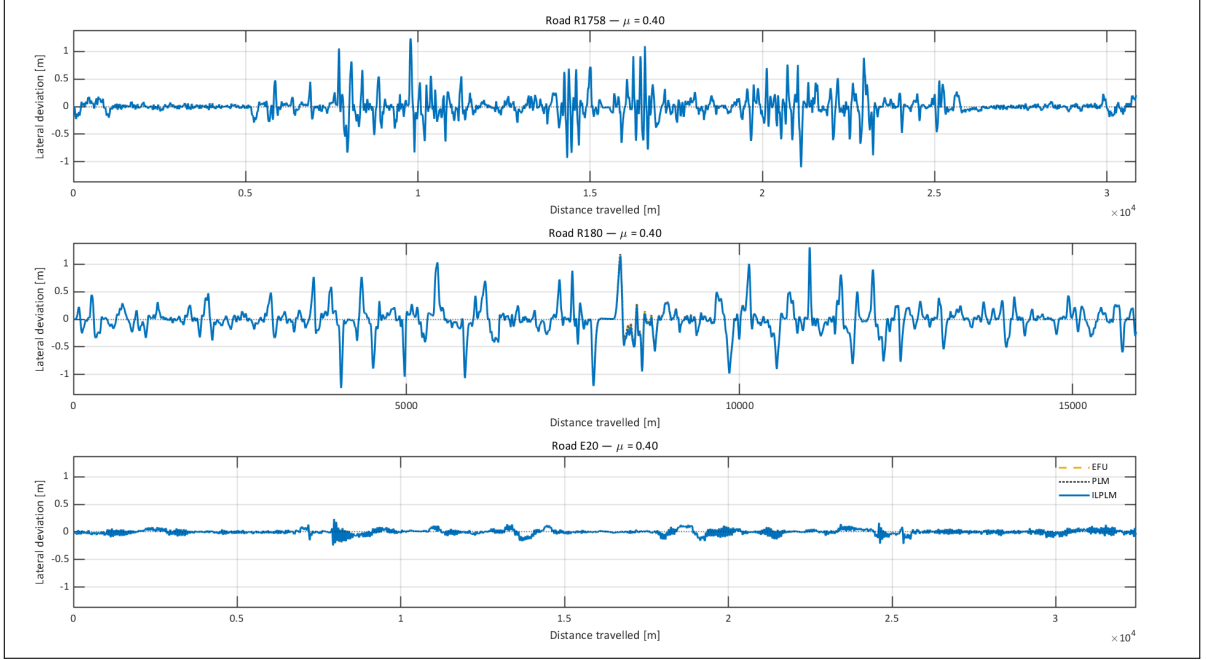


Figure 5.13: Lateral Deviation (6x4 rigid, $\mu = 0.40$)

5.2 Lateral stability

5.2.1 Lateral stability for 4x4 rigid truck configuration

Consider a friction circle with a maximum limit of μF_z as shown in Figure 5.14. The green dot indicates the lateral force F_y that comes from the VTM simulation model. The critical limit F_y is given by $F_{y,lim}$ and is indicated by a red dot.

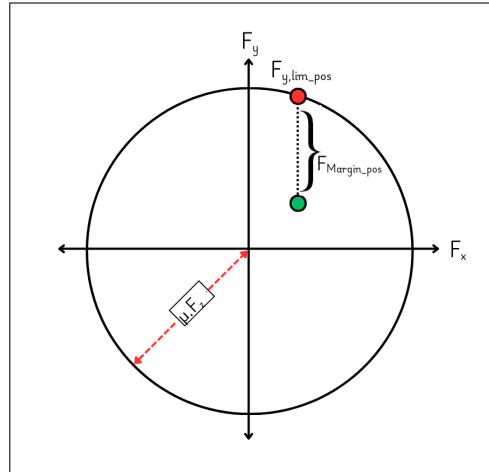


Figure 5.14: Friction Circle with positive F_{margin} limits for a wheel

where,

$$F_{y\text{lim,pos}} = \sqrt{(\mu F_z)^2 - F_x^2} \quad (5.1)$$

$$F_{\text{margin,pos}} = F_{y\text{lim,pos}} - F_y \quad (5.2)$$

The $F_{\text{margin,pos}}$ is the difference between the limit of the critical lateral force and the lateral force of the model, which is given by equation (5.2).

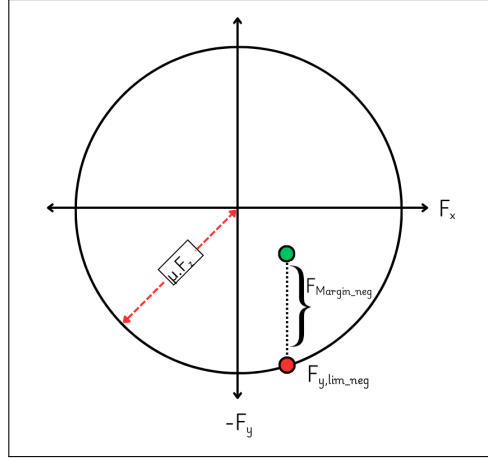


Figure 5.15: Friction Circle with negative F_{margin} limits for a wheel

The lateral force in the opposite direction is denoted by $F_{y,neg}$ and the critical limit as $F_{y\text{lim,neg}}$. Similarly, $F_{\text{margin,neg}}$ is given by equation (5.4).

$$F_{y\text{lim,neg}} = -\sqrt{(\mu F_z)^2 - F_x^2} \quad (5.3)$$

$$F_{\text{margin,neg}} = -(F_{y\text{lim}} - F_y) \quad (5.4)$$

The same concept has been applied and can be represented in lateral force vs distance plots for a 4x4 truck Figure 5.16, which is simulated for the R1758 Bollebygd-Töllsjö road. The four subplots represent different wheels showing F_y+ and F_y- , which come from the VTM simulation model.

Lateral force limits of the wheels are calculated using equations (5.1) and (5.3). Similarly, there are positive and negative lateral margins for each wheel, depending on the left or right turn taken by the truck.

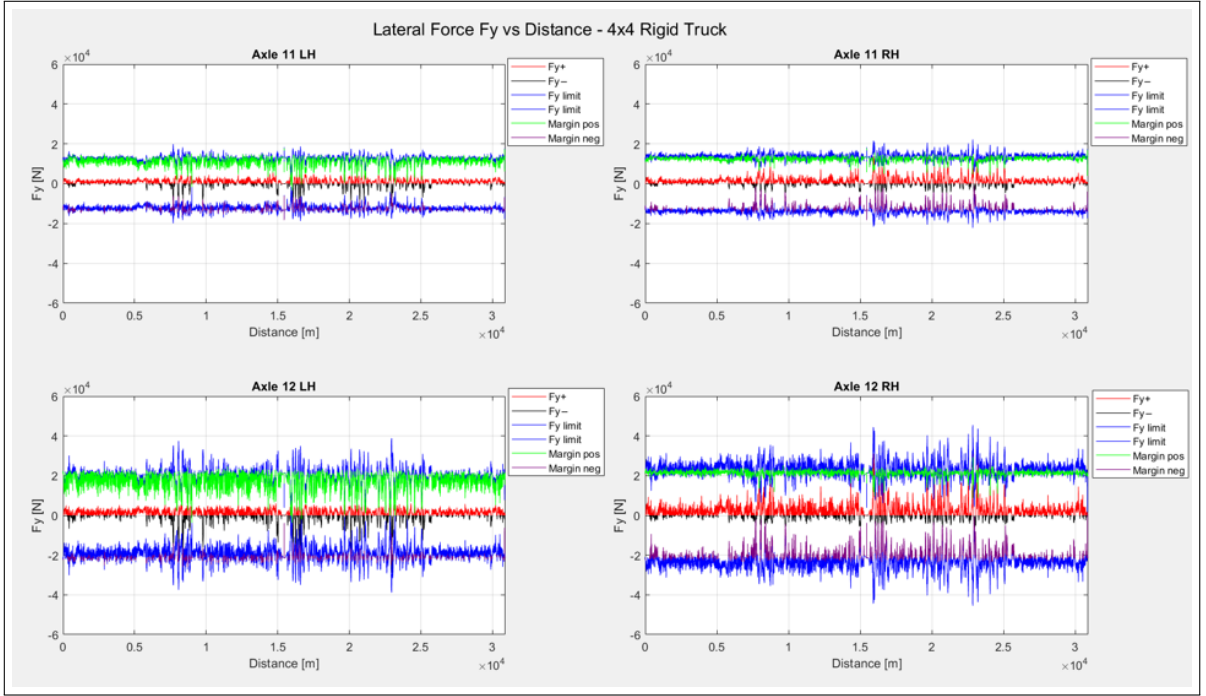


Figure 5.16: Lateral force vs Distance (ILPLM, 4x4 rigid, $\mu = 0.40$, $R1758$)

The lateral forces and lateral margins from Figure 5.16 are evaluated at the wheel level. Equations (5.5) and (5.6) calculate the positive and negative lateral margins of the front axle.

From equation (5.7), we obtain a single margin on the front axle where the minimum of the positive front axle margin and the absolute value of the negative front axle margin is taken.

$$F_{\text{margin,pos,front}} = F_{\text{margin,pos,Ax11LH}} + F_{\text{margin,pos,Ax11RH}} \quad (5.5)$$

$$F_{\text{margin,neg,front}} = F_{\text{margin,neg,Ax11LH}} + F_{\text{margin,neg,Ax11RH}} \quad (5.6)$$

$$F_{\text{margin,front}} = \min(F_{\text{margin,pos,front}}, |F_{\text{margin,neg,front}}|) \quad (5.7)$$

The margin of the rear axle is computed the same way as the front axle margin and is given by equation (5.10).

$$F_{\text{margin,pos,rear}} = F_{\text{margin,pos,Ax12LH}} + F_{\text{margin,pos,Ax12RH}} \quad (5.8)$$

$$F_{\text{margin,neg,rear}} = F_{\text{margin,neg,Ax12LH}} + F_{\text{margin,neg,Ax12RH}} \quad (5.9)$$

$$F_{\text{margin,rear}} = \min(F_{\text{margin,pos,rear}}, |F_{\text{margin,neg,rear}}|) \quad (5.10)$$

To obtain the maximum amount of grip utilized, we normalize (with respect to normal or vertical load) the equations (5.7) and (5.10), from which we obtain Index 1 and Index 2, which is then formulated in equations (5.11) and (5.12), whereas Index 3 is the minimum of Index 1 and Index 2.

$$\text{Index 1} = \frac{F_{\text{margin,front}}}{\mu F_z} \quad (5.11)$$

$$\text{Index 2} = \frac{F_{\text{margin,rear}}}{\mu F_z} \quad (5.12)$$

$$\text{Index 3} = \min(\text{Index 1}, \text{Index 2}) \quad (5.13)$$

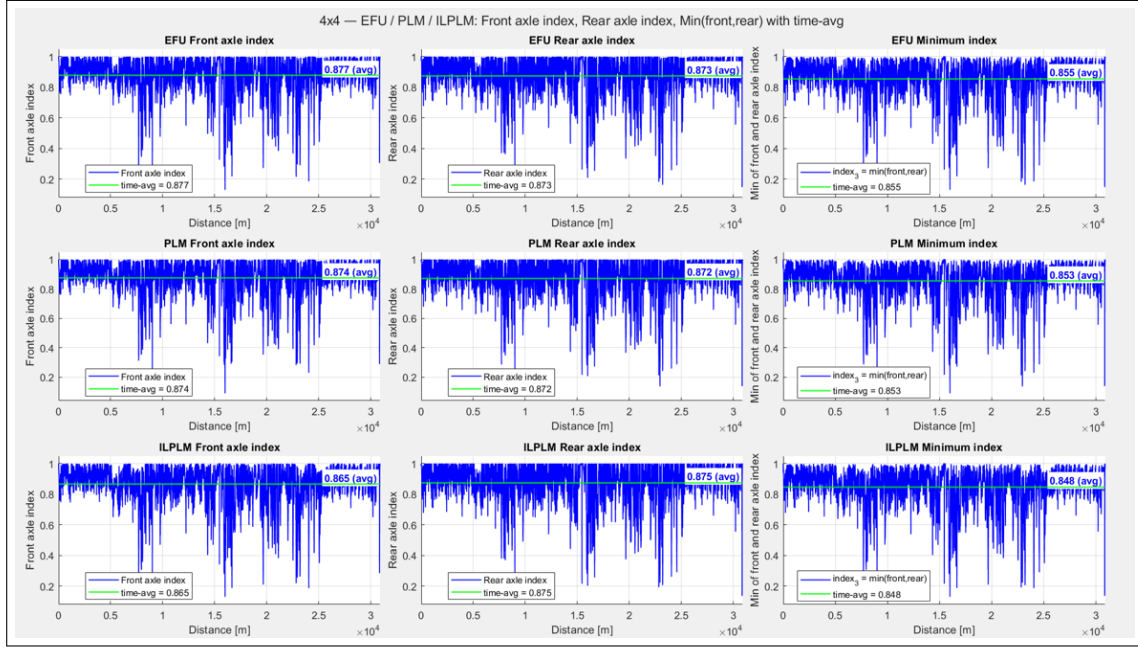


Figure 5.17: Comparison of normalised lateral force vs Distance (4x4 rigid, $\mu = 0.40$, R1758)

Figure 5.17 shows the indices plotted over the distance travelled for all the coordination strategies for the R1758 Bollebygd-Töllsjö road. The closer the index is to 1, the more stable and safe the truck is. If the index is closer to 0, it leads to instability as it would mean the wheels are slipping, which in turn would make the truck unsafe for driving.

To validate the lateral margin plots, a time-weighted average has been used in this study. The reason for choosing this method instead of using a simple arithmetic mean is that the simulation uses non-uniform time steps, as a variable step solver (ode45) has been used for this work. The solver adjusts the size of the time step automatically, which results in unequal time intervals between the data points. On the other hand, a time-weighted average makes sure that the values of the data points contribute to the average proportionally to the time duration they persist, which results in a more accurate measure.

It is evident from the minimum index plots in the figure that the EFU is more stable, as the time average of EFU is higher than PLM, and the least stable is ILPLM. The order of stability according to the time average is EFU, PLM and ILPLM, as expected.

But the time average is not an appropriate metric to measure the stability of a truck since real driving considers many other factors, including turns, road camber, bumps, braking, and load transfer. It can also be seen from Figure 5.17, there are some low-value spikes where the index is closer to 0 due to short periods of low stability, which can happen due to various reasons, for example, tire slip that gets

smoothened out by time average. Therefore, the time average is not helpful to determine which coordination strategy is more stable than the other. So, a different metric is proposed to measure the stability.

Another approach, which involves using a percentile-based sorting, was used to evaluate and compare the distribution of the index 3 (minimum index) across the three coordination strategies.

Firstly, all the valid or real values of the index 3 are collected from the simulated result and are sorted in ascending order, which brings forth a clear understanding of how the index values are distributed across the coordination strategy.

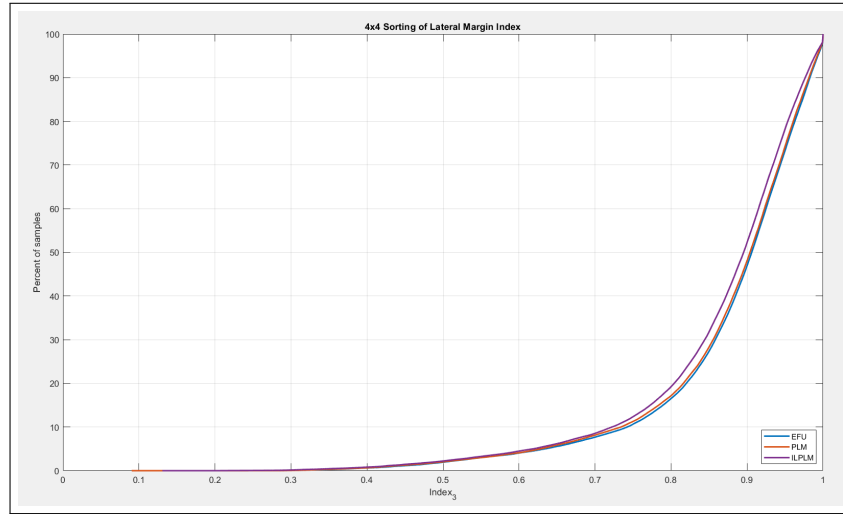


Figure 5.18: Percent of samples versus index (4x4 rigid, $\mu = 0.40$, R1758)

The sorted index values are plotted against their corresponding percentiles (percentage of samples), and the three curves represent the three coordination strategies as seen in Figure 5.18. From the same graph, it can be seen that the blue curve, which represents EFU, lies farthest to the right, depicting that a higher number of the samples have larger index 3 values as compared to the other strategies. This also indicates that a larger index of the lateral margin is being maintained more consistently.

Basically, the more the curve is towards the right in the plot, indicated by EFU, makes it the most stable strategy resulting in more samples with higher values of the index followed by PLM.

On the other hand, the more the curve is towards the left, represented by ILPLM, is highlighted as the most unstable strategy, as the index value is more towards 0 (after which it will start slipping).

The same plots of percent of samples versus index have been done for road R180 Hällered-Alingsås and road E20 Gothenburg-Alingsås, and are attached in Appendix 1.

5.2.2 Lateral stability for 6x4 rigid truck configuration

This part represents the lateral stability results for a 6x4 rigid truck. The formulation for this configuration is derived from Section 5.2.1, and the amendments were made only for the additional axle. The lateral forces and lateral margins from Figure 5.19 are evaluated at the wheel level, which have been derived from equations 5.14-5.22.

As discussed above in the previous section, equations 5.5, 5.6, and 5.7 calculate the positive and negative lateral margins, and finds the minimum of the front axle, same way for a 4x4 rigid truck.

The front axle margin and the rear axle margin of the rear axle group are computed as:

$$F_{\text{margin,pos,front}} = F_{\text{margin,pos,Ax11LH}} + F_{\text{margin,pos,Ax11RH}} \quad (5.14)$$

$$F_{\text{margin,neg,front}} = F_{\text{margin,neg,Ax11LH}} + F_{\text{margin,neg,Ax11RH}} \quad (5.15)$$

$$F_{\text{margin,front}} = \min(F_{\text{margin,pos,front}}, |F_{\text{margin,neg,front}}|) \quad (5.16)$$

$$F_{\text{margin,pos,rear,1}} = F_{\text{margin,pos,Ax12LH}} + F_{\text{margin,pos,Ax12RH}} \quad (5.17)$$

$$F_{\text{margin,neg,rear,1}} = F_{\text{margin,neg,Ax12LH}} + F_{\text{margin,neg,Ax12RH}} \quad (5.18)$$

$$F_{\text{margin,rear,1}} = \min(F_{\text{margin,pos,rear,1}}, |F_{\text{margin,neg,rear,1}}|) \quad (5.19)$$

$$F_{\text{margin,pos,rear,2}} = F_{\text{margin,pos,Ax13LH}} + F_{\text{margin,pos,Ax13RH}} \quad (5.20)$$

$$F_{\text{margin,neg,rear,2}} = F_{\text{margin,neg,Ax13LH}} + F_{\text{margin,neg,Ax13RH}} \quad (5.21)$$

$$F_{\text{margin,rear,2}} = \min(F_{\text{margin,pos,rear,2}}, |F_{\text{margin,neg,rear,2}}|) \quad (5.22)$$

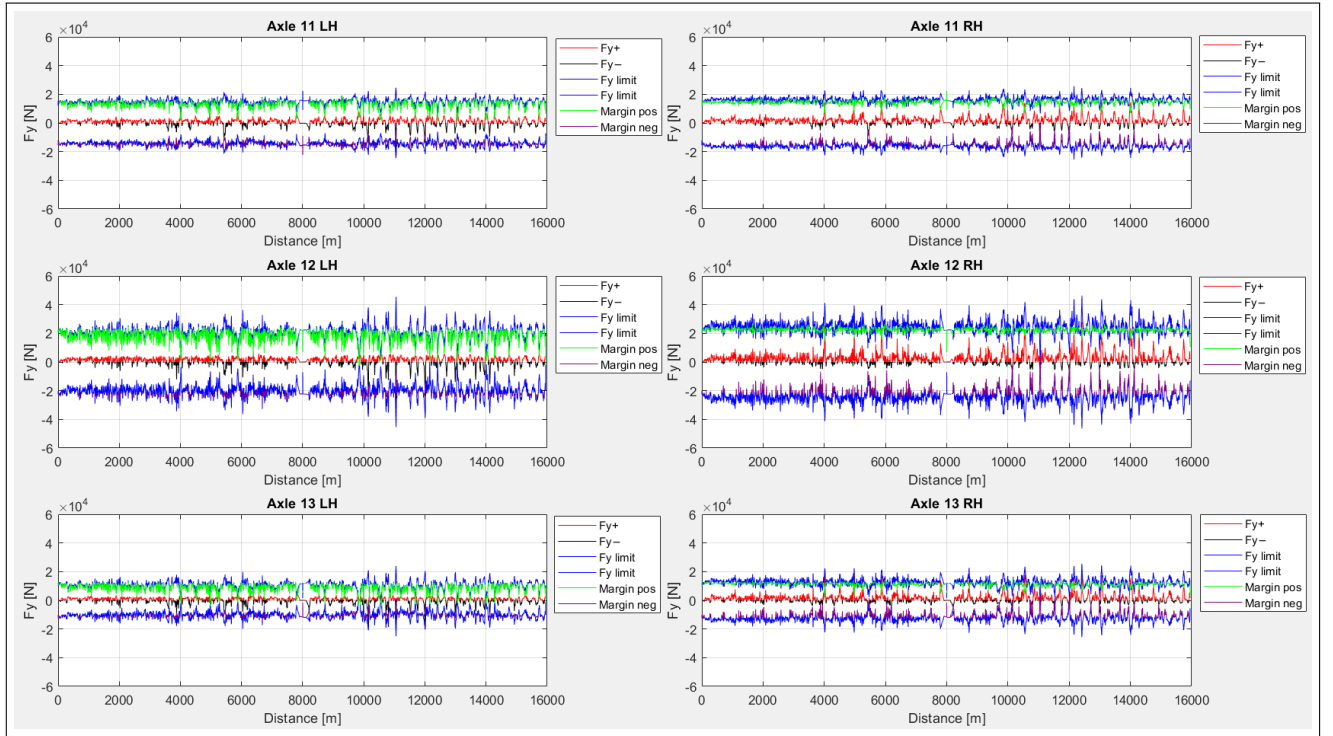


Figure 5.19: Lateral force vs Distance (ILPLM, 6x4 rigid, $\mu = 0.40$, $R180$)

Index 1, Index 2, and Index 3 have been calculated in the same way as mentioned in equations 5.11 and 5.12, whereas Index 4 is the minimum of Index 1 and the sum of Index 2 and Index 3.

$$\text{Index 1} = \frac{F_{\text{margin, front}}}{\mu F_{z, \text{front}}} \quad (5.23)$$

$$\text{Index 2} = \frac{F_{\text{margin, rear, 1}}}{\mu F_{z, \text{rear, 1}}} \quad (5.24)$$

$$\text{Index 3} = \frac{F_{\text{margin, rear, 2}}}{\mu F_{z, \text{rear, 2}}} \quad (5.25)$$

$$\text{Index 4} = \min \left(\text{Index 1}, \sum (\text{Index 2}, \text{Index 3}) \right) \quad (5.26)$$

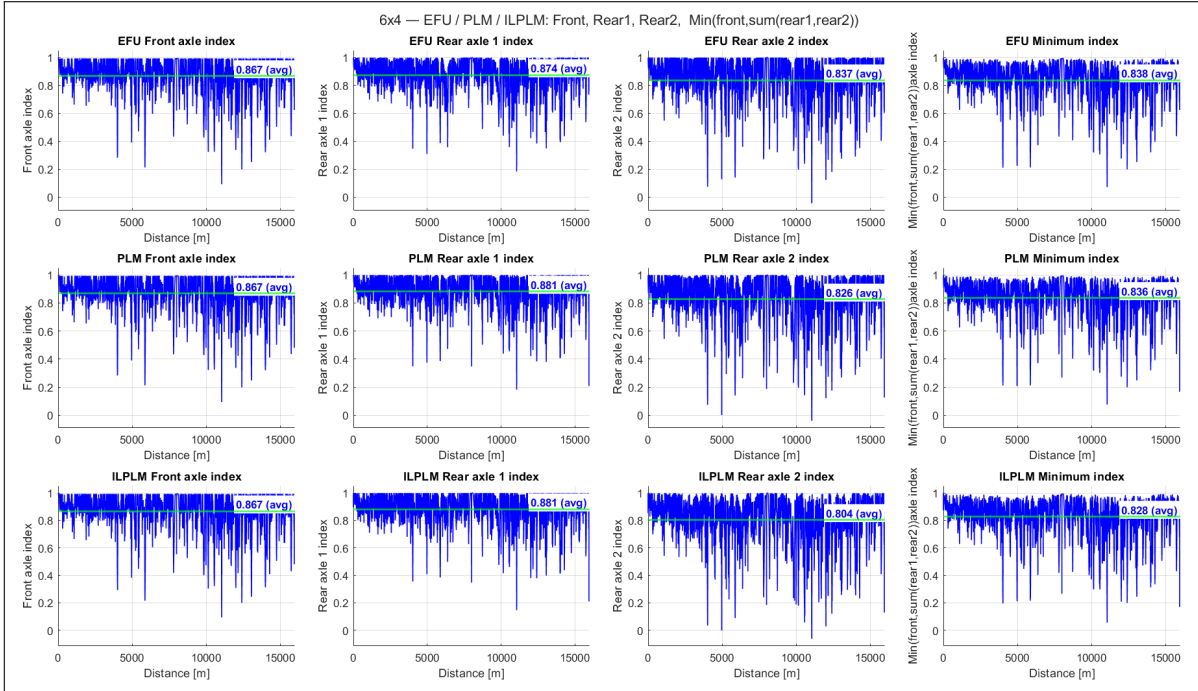


Figure 5.20: Comparison of normalised lateral force vs Distance (ILPLM, 6x4 rigid, $\mu = 0.40$, R180)

Figure 5.20 shows the indices plotted over the distance travelled for all the three coordination strategies for the R180 Hällered to Alingsås road.

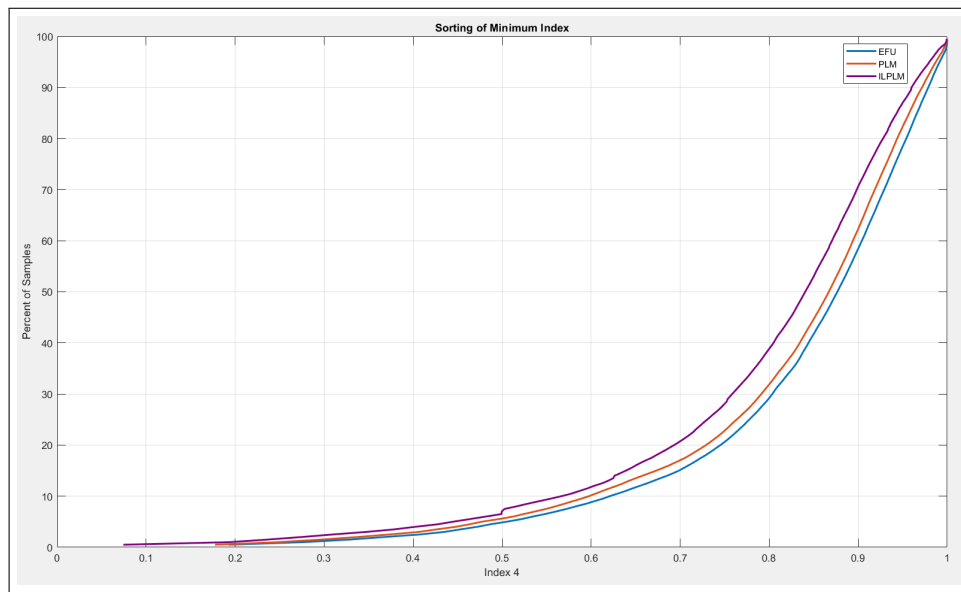


Figure 5.21: Percent of samples vs index (6x4 rigid, $\mu = 0.40$, $R180$)

Figure 5.21 represents the sorting plot for the 6x4 configuration, which uses the same technique from section 5.2.1. In this plot, the trend tends to be the same as expected, making EFU the safest and ILPLM the most unsafe.

The same plots of percent of samples versus index have been done for road R180 Hällered-Alingsås and road E20 Gothenburg-Alingsås, and are attached in Appendix 1.

5.3 Energy Savings

To assess how efficient each coordination strategy is, the energy consumption needs to be evaluated. Figures 5.22 and 5.23 highlight the energy consumption in different types of power loss that have been simulated on the R1758 road for 4x4 and 6x4 rigid trucks. EFU and PLM strategies produce similar losses in the drivetrain while converting from electrical to mechanical energy. ILPLM strategy has proven to deliver the least drivetrain losses as one axle gets electrically disconnected, making it the most efficient one.

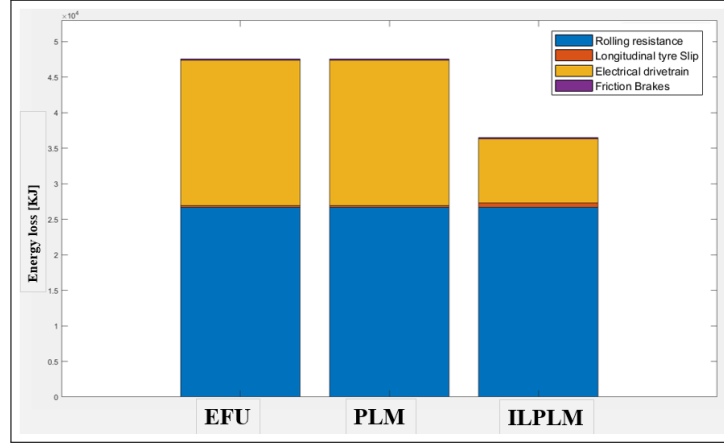


Figure 5.22: Energy plot (4x4 rigid, $\mu = 0.40$, R1758)

However, the losses generated from friction brakes and the drivetrain are only minimised in the case of PLM and ILPLM. In contrast, rolling resistance and longitudinal slip losses occur due to the torque split across the wheels as a result of different kinds of wheel torque coordination strategies. In addition, longitudinal tire slip losses are greater in the case of ILPLM compared to other strategies as the load on the axles gets higher, which are not electrically disconnected.

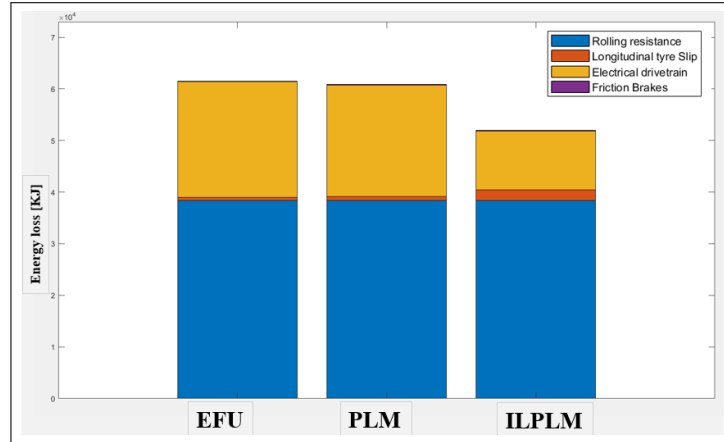


Figure 5.23: Energy plot (6x4 rigid, $\mu = 0.40$, R1758)

Friction brakes are nearly the same for all strategies unless met by a sudden deceleration due to a sudden U-turn or any other scenario, and can also be seen in Figures 3.2 to 3.4. The same energy plots have been done for road R180 Hällered-Alingsås and road E20 Gothenburg-Alingsås, and are attached in Appendix 1.

5.4 Influence of disconnecting Startability Axle for 4x4 and 6x4 rigid truck configuration

Figures 5.24 and 5.25 represent the power loss of the two axles equipped with electric machines for the 4x4 and 6x4 rigid trucks. In the case of ILPLM, the startability axle or the IM is electrically disconnected, and the load increases on the other axle as the utilisation increases, because of which the other EM axle produces increased power losses, as represented by Ax11 in Figure 5.24 and also by Ax13 in Figure 5.25.

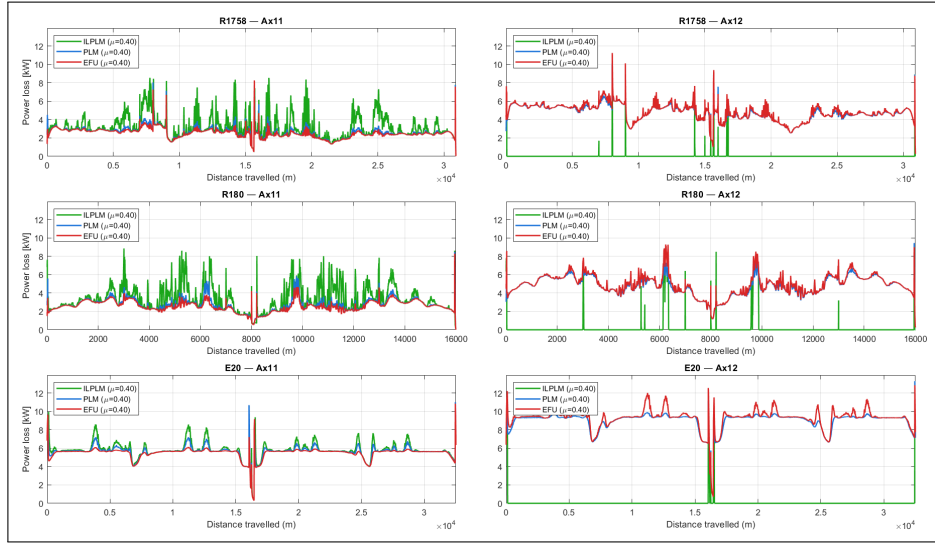


Figure 5.24: Axle Switching (4x4 rigid, $\mu = 0.40$)

On the other hand, the IM switches on for a minimal period of time and turns off in order to fulfil the requested force and can be depicted by Ax12 in Figures 5.24 and 5.25. On top of that, there are certain instances where the frequency of switching on and off was higher, which generates higher oscillations in the drivetrain.

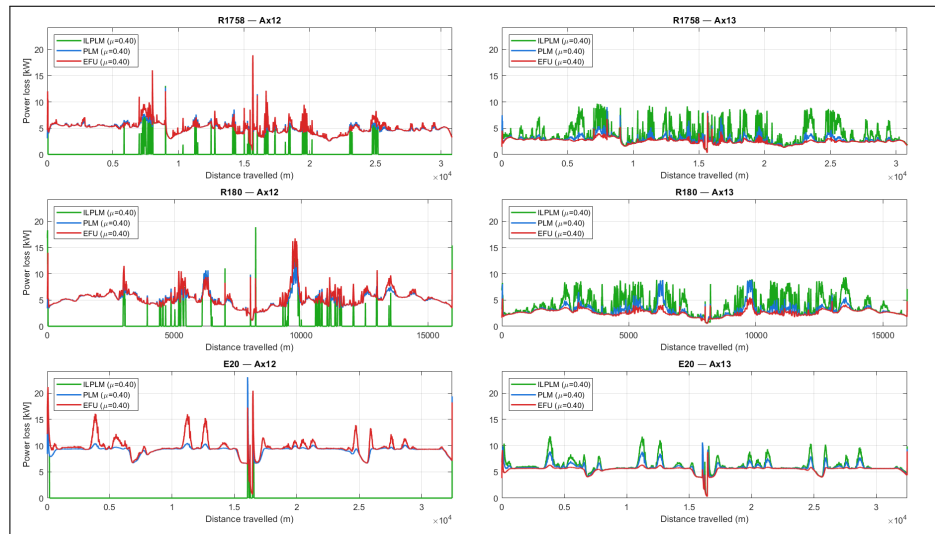


Figure 5.25: Axle Switching (6x4 rigid, $\mu = 0.40$)

6

Conclusion

For one to conclude, the main bargain comes in whether to choose the most efficient strategy or the safest strategy. Thus, the optimal solution is to find a solution that is energy efficient as well as safe.

Figures 6.1 and 6.2 highlight the trade-off between the lateral stability and normalised total energy consumed for the three different wheel torque coordination strategies. The comparison is established based on the value of the 20th percentile (or 20% of the total number of samples) corresponding to the lateral index margin (index 3) for a 4x4 truck.

The normalised total energy consumed is defined as the sum of the total energy losses (rolling resistance, longitudinal tyre slip, drivetrain, friction brakes) normalised with the product of the distance travelled by the vehicle during the simulation, gross weight of the vehicle, and acceleration due to gravity:

$$\text{Normalised Total Energy Consumed} = \frac{\text{Total Consumed Energy}}{\text{Distance travelled} \times GW_{\text{truck}} \times g} \quad (6.1)$$

where, Normalised total energy consumption: (dimensionless), Total energy consumed: [J or N · m], Distance travelled: [m], GW_{truck} : Gross weight of the truck [kg], g : Gravitational constant [m/s^2]

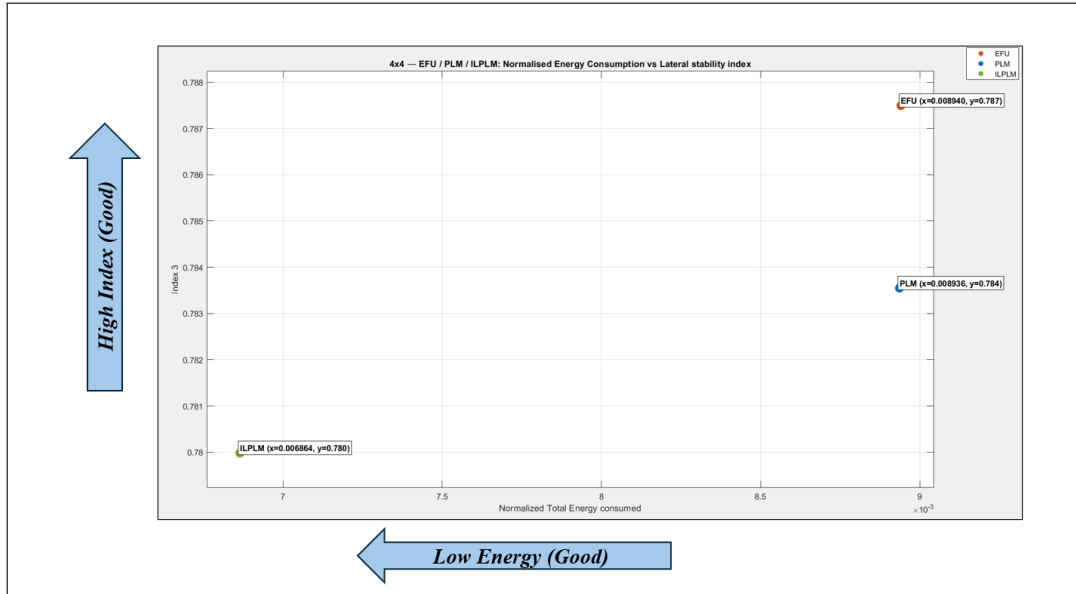


Figure 6.1: Trade off plot(4x4 rigid $\mu = 0.40$)

Figure 6.2 highlights the case for a 6x4 rigid truck and uses the same principle as used above. The only modification done is by replacing the index 3 with index 4 here.

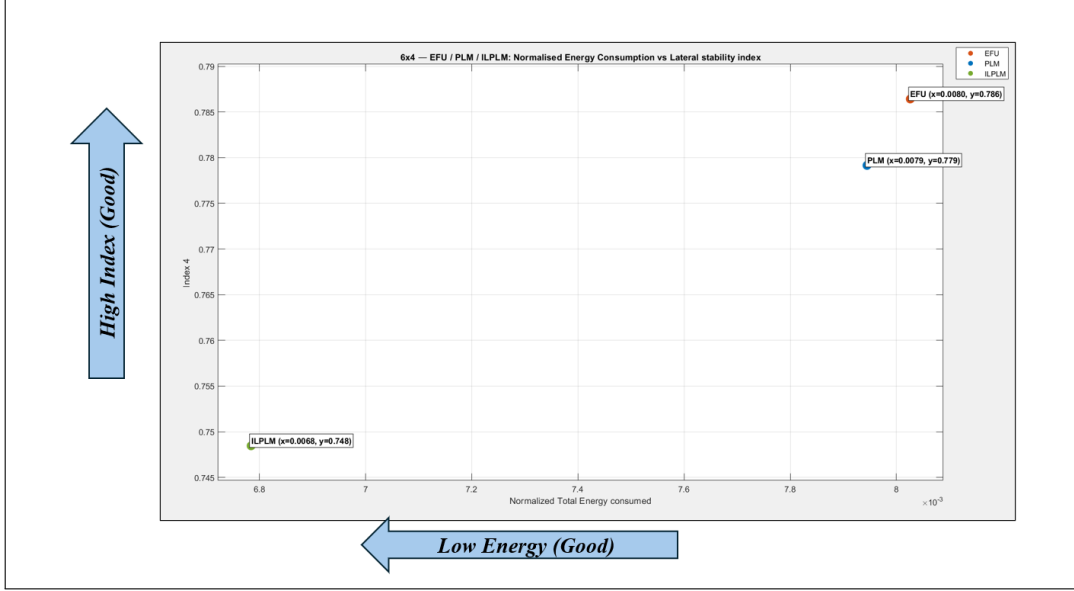


Figure 6.2: Trade off plot(6x4 rigid $\mu = 0.40$)

To comprehend the trade-off plots, as lower energy is consumed (enhanced efficiency), the lateral margin index tends to decrease slightly. Thus, EFU exhibits high stability (higher value of lateral index) with a higher value of normalised total energy consumed, followed by PLM. While ILPLM depicts higher energy savings, but with less lateral margin index.

Thus, to conclude the research, the following objectives have been fulfilled:

- Proposed metrics for vehicle stability and energy savings for different truck configurations with different axle positions for comparing all the wheel torque coordination strategies.
- The energy savings have been evaluated for the wheel torque coordination strategies for different kinds of configurations, which have proven to be the best in ILPLM, followed by PLM and EFU for both 4x4 and 6x4 rigid trucks.
- In terms of lateral stability, EFU is proven to be more stable, followed by PLM and ILPLM in case of both 4x4 and 6x4 rigid trucks.

7

Future Work

- Improving the Driver model to reduce aggressive driving behaviour.
- To make a longitudinal wheel slip controller for lower friction roads.
- Investigation of City Driving Scenario.
- Study on Construction Site Driving Scenario.

Bibliography

- [1] Zhang, Ronghui Li, Kening Yu, Fan He, Zhaocheng Yu, Zhi. (2017). Novel electronic braking system design for EVS based on constrained nonlinear hierarchical control. *International Journal of Automotive Technology*. 18. 707-718. 10.1007/s12239-017-0070-0.
- [2] Janardhanan, S. (2023). On power loss minimisation for heavy vehicles with axle-wise and modular electrical propulsion and friction braking. Thesis for the Degree of Licentiate of Engineering in Machine and Vehicle Systems, Chalmers University of Technology. <http://dx.doi.org/10.13140/RG.2.2.20661.76008>
- [3] Verbruggen, F. J. R., Rangarajan, V., & Hofman, T. (2019). Powertrain design optimization for a battery electric heavy-duty truck. *2019 American Control Conference (ACC)*, 1488–1493. IEEE. doi:10.23919/ACC.2019.8814712.
- [4] Morozov, A., Humphries, K., Zou, T., Rahman, T., & Angeles, J. (2018). Design, analysis, and optimization of a multi-speed powertrain for class-7 electric trucks. *SAE International Journal of Alternative Powertrains*, 7(1), 27–42.
- [5] Verbruggen, F., Salazar, M., Pavone, M., & Hofman, T. (2020). Joint design and control of electric vehicle propulsion systems. *2020 European Control Conference (ECC)*, 1725–1731. IEEE. doi:10.23919/ECC51009.2020.9143733.
- [6] Janardhanan, S., Henderson, L., Jonasson, M., et al. (2025). Simulation-based assessment of wheel torque allocation strategies on heavy vehicles with drivetrains on multiple axles. *Transportation Engineering*, 20. <http://dx.doi.org/10.1016/j.treng.2025.100322>
- [7] Kobayashi, T., Katsuyama, E., Sugiura, H., Hattori, Y., Ono, E., Yamamoto, M. (2020). Theoretical analysis of tyre slip power dissipation mechanism using brush model. *Vehicle System Dynamics*, 58(8), 1242–1256. <http://dx.doi.org/10.1080/00423114.2019.1612926>
- [8] Jacobson, B. (2023). Vehicle Dynamics - Compendium for Course MMF062. Chalmers University of Technology.
- [9] Un-Noor, F., Padmanaban, S., Mihet-Popa, L., Mollah, M. N., Hossain, E. (2017). A comprehensive study of key electric vehicle (EV) components, technologies, challenges, impacts, and future direction of development. *Energies*, 10(8). <http://dx.doi.org/10.3390/en10081217>
- [10] Pellegrino, G., Vagati, A., Boazzo, B., Guglielmi, P. (2012). Comparison of induction and PM synchronous motor drives for EV application including design examples. *IEEE Transactions on Industry Applications*, 48(6), 2322–2332. <http://dx.doi.org/10.1109/TIA.2012.2227092>

- [11] F. N. (2024). Handling analysis and control development of commercial trucks with Volvo transport models. [Online]. Available: <https://se.mathworks.com/videos/handling-analysis-and-control-development-of-commercial-trucks-with-volvo-transport-models-1622035211192.html>
- [12] Janardhanan, S., Henderson, L., Jonasson, M., Jacobson, B., Gelso, E. R. (2025). Simulation-based assessment of wheel torque allocation strategies on heavy vehicles with drivetrains on multiple axles. *Transportation Engineering*, 20, 100322. ISSN 2666-691X. <https://doi.org/10.1016/j.treng.2025.100322>
- [13] Laine, L., Andreasson, J. (2007). Control allocation based electronic stability control system for a conventional road vehicle. In *Proceedings of the 2007 IEEE Intelligent Transportation Systems Conference*, pp. 514–521. <http://dx.doi.org/10.1109/ITSC.2007.4357632>
- [14] Janardhanan, S., Laine, L., Jonasson, M., Jacobson, B., Gelso, E. (2022). *Motion and power coordination of electric propulsion and braking distributed on multiple axles on heavy vehicles.* Submitted to IEEE VPPC 2022, Merced, USA, Nov 1–4, 2022, pp. 1–8.
- [15] Pennycott, A., De Novellis, L., Sabbatini, A., Gruber, P., Sorniotti, A. (2014). *Reducing the motor power losses of a four-wheel drive, fully electric vehicle via wheel torque allocation.* *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, 228(7), 830–839.
- [16] Persson, J., Åkesson, J. (2024). *On torque vectoring to improve steering predictability while minimising power loss in heavy electric vehicles using model predictive control.* Master's Thesis in Systems, Control and Mechatronics, Department of Mechanics and Maritime Sciences, Chalmers University of Technology.
- [17] U. Erdinc, M. Jonasson, M. S. K., L. Laine, B. Jacobson, and J. Fredriksson, “Experimental Validation of Yaw Stability Control Strategies for Articulated Vehicle Combinations,” in *Proceedings of the 2024 IEEE Intelligent Vehicles Symposium (IV)*, Jeju Island, Korea, 2024, pp. 1476–1483.

A

Appendix 1

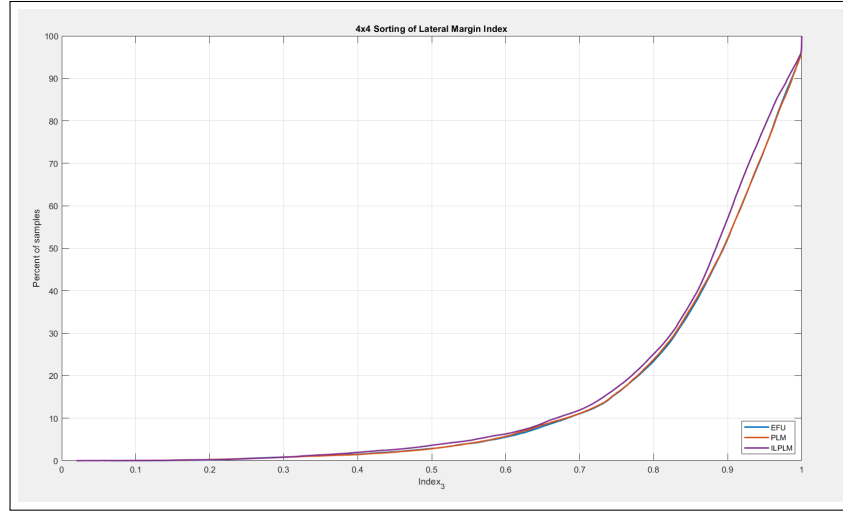


Figure A.1: Percent of samples versus index (4x4 rigid, $\mu = 0.40$, R180)

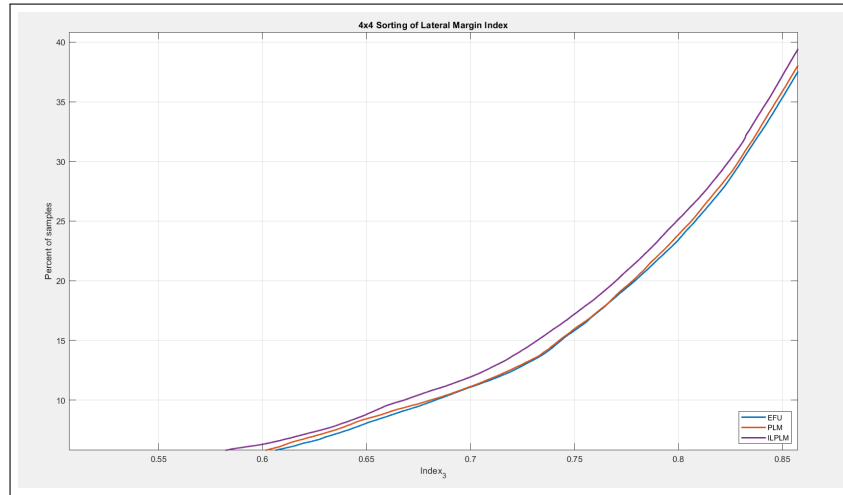


Figure A.2: Percent of samples versus index, Zoomed view (4x4 rigid, $\mu = 0.40$, R180)

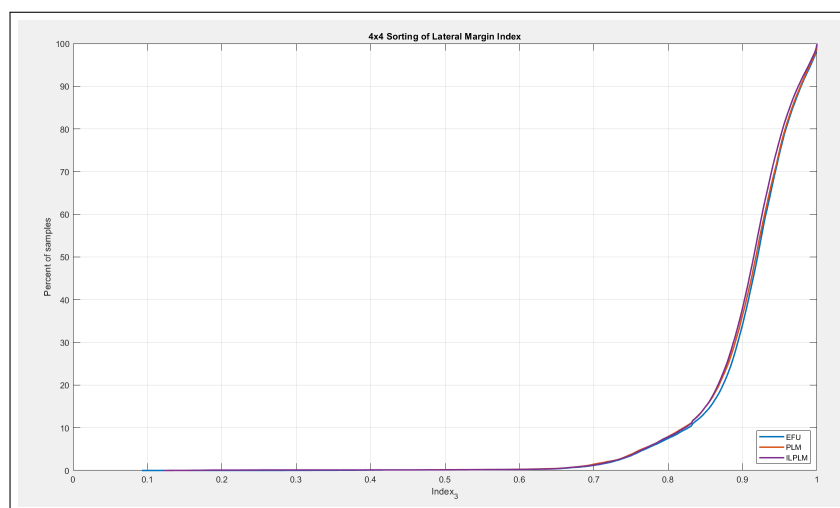


Figure A.3: Percent of samples versus index (4x4 rigid, $\mu = 0.40$, E20)

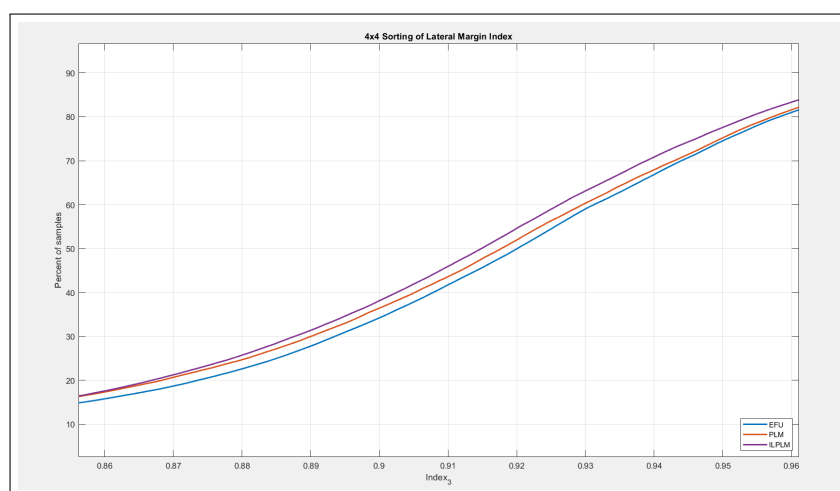


Figure A.4: Percent of samples versus index, Zoomed view (4x4 rigid, $\mu = 0.40$, E20)

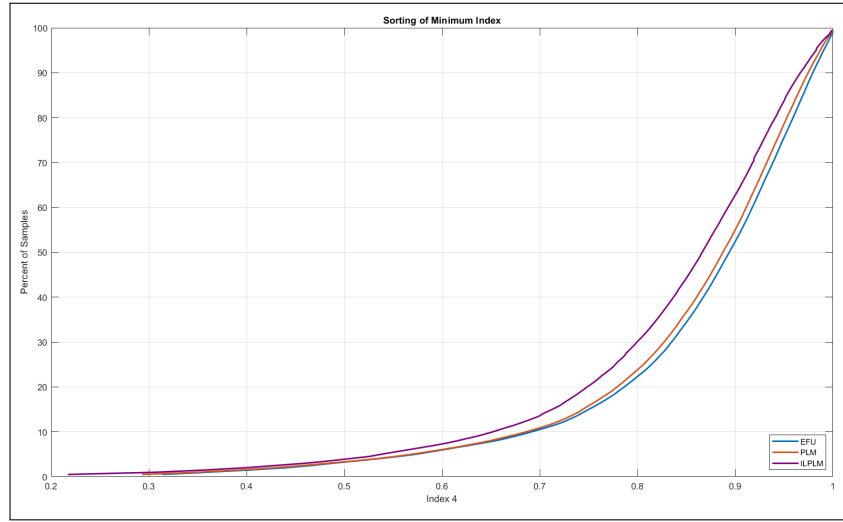


Figure A.5: Percent of samples versus index (6x4 rigid, $\mu = 0.40$, R1758)

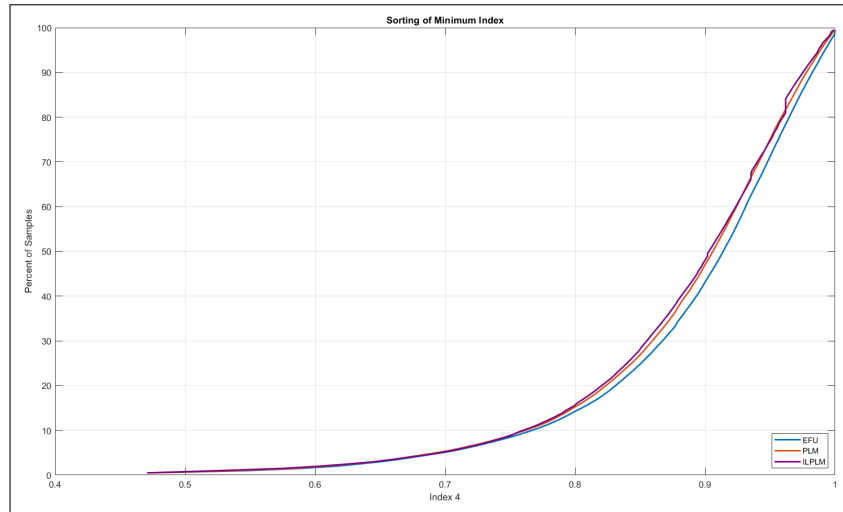


Figure A.6: Percent of samples versus index (6x4 rigid, $\mu = 0.40$, E20)

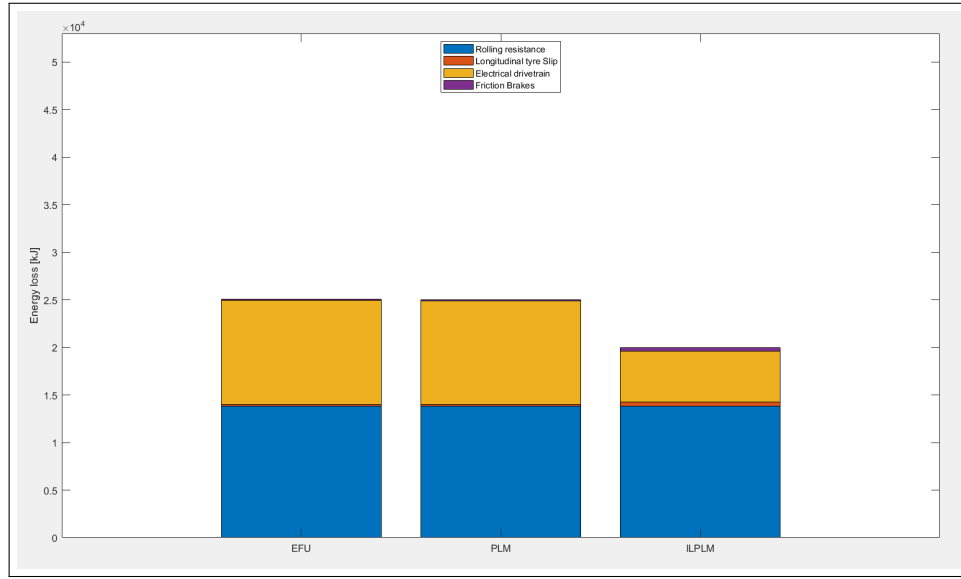


Figure A.7: Energy plot (4x4 rigid, $\mu = 0.40$, $R180$)

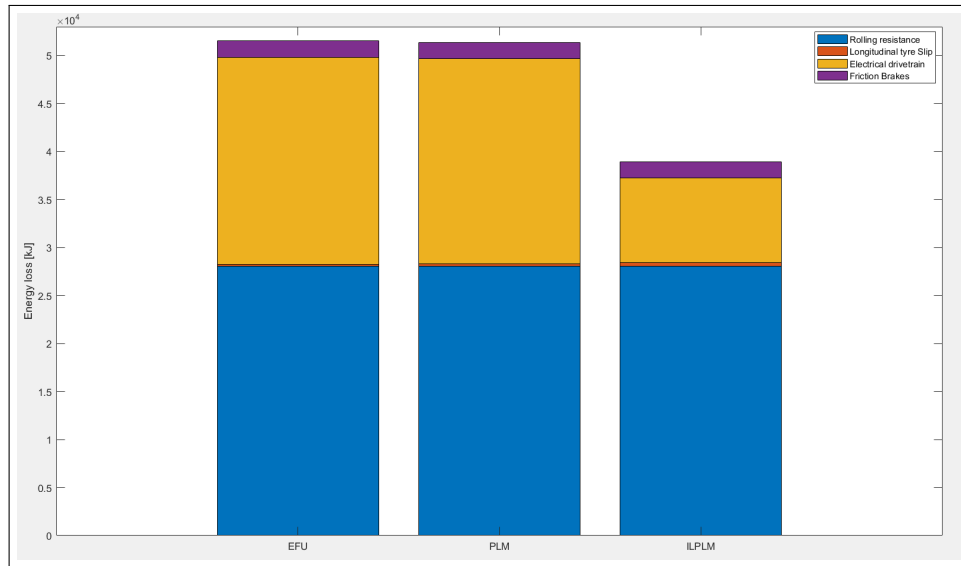


Figure A.8: Energy plot (4x4 rigid, $\mu = 0.40$, $E20$)

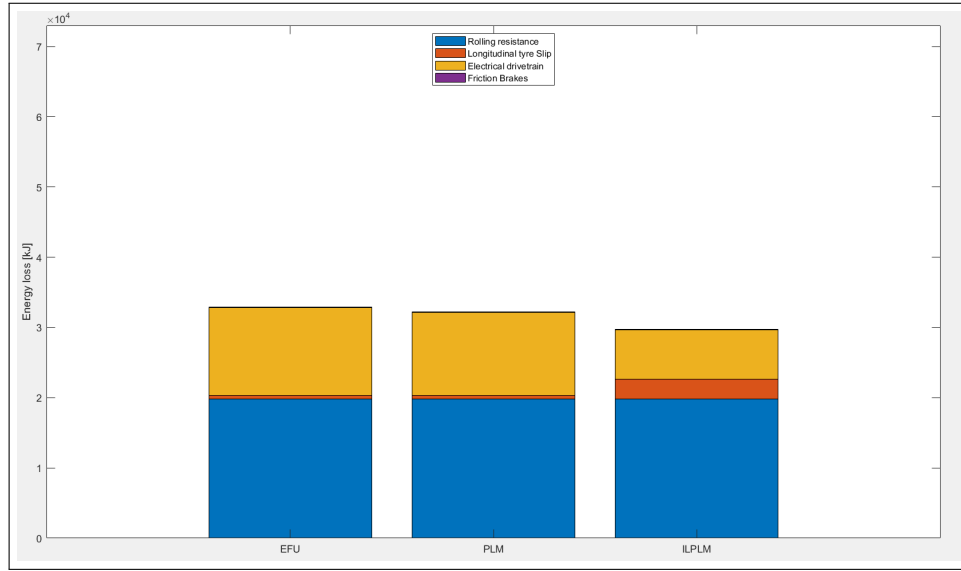


Figure A.9: Energy plot (6x4 rigid, $\mu = 0.40$, R180)

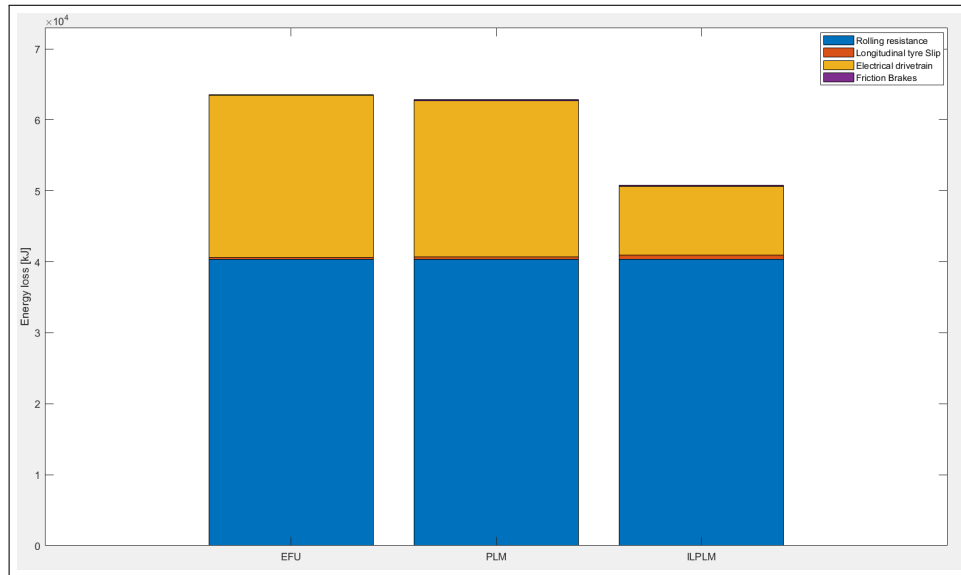


Figure A.10: Energy plot (6x4 rigid, $\mu = 0.40$, E20)

DEPARTMENT OF MECHANICS AND MARITIME SCIENCES

CHALMERS UNIVERSITY OF TECHNOLOGY

Gothenburg, Sweden 2025

www.chalmers.se



CHALMERS
UNIVERSITY OF TECHNOLOGY