

**Design and Analysis of Energy-Efficient Cooperative Truck Platooning
Systems with Minimum Tailpipe Emissions**

by

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International

Abstract

Cooperative truck platooning can reduce energy consumption by leveraging aerodynamic drag reduction and coordinated motion; however, real-world highway operations are strongly affected by environmental disturbances (e.g., wind and road grade) and by lateral misalignment that can partially displace the follower from the leader’s wake. Moreover, regulatory constraints often require large time gaps (e.g., ≥ 2 sec), where wake benefits diminish and conventional platooning strategies yield limited and highly variable gains. In parallel, the rapid electrification of Class-8 trucks motivates a systematic comparison of platooning benefits across powertrain architectures, because motor assist, regenerative braking, and power-split flexibility introduce new mechanisms for energy optimization and emissions mitigation beyond conventional aerodynamics. This thesis addressed these research gaps by developing and validating disturbance-aware, powertrain-integrated platooning models and control strategies for Class-8 truck platoons, with the objectives of ensuring safety, improving energy savings beyond aerodynamics alone at regulation-compliant gaps, and mitigating tailpipe NO_x emissions through transient-aware powertrain coordination.

First, Canada’s first on-road cooperative truck platooning trials were analyzed to quantify how wind, road geometry, and lateral misalignment influence spacing regulation, fuel consumption, and NO_x emissions. Building on these trials, an integrated modelling framework was developed and validated using independent real-world datasets, unifying coupled longitudinal–lateral platoon dynamics, spacing- and offset-dependent aerodynamic interaction, and energy and emissions estimation. To

improve NO_x prediction fidelity under transient operation, a data-driven long short-term memory (LSTM)-based model was trained to capture transient NO_x dynamics. Validation results showed close agreement with measured trajectories and cumulative outcomes, including low errors in speed, spacing, and lateral states, and small deviations in cumulative fuel and tailpipe NO_x .

Second, hierarchical cooperative controllers were designed for large-spacing platooning under road-grade and wind disturbances. Numerical evaluations on Alberta Highway 2 demonstrated that meaningful benefits remained achievable at large time gaps, with fuel savings up to 6.6% and tailpipe NO_x reductions up to 23.2% under grade-varying conditions, and fuel savings up to 5.5% with tailpipe NO_x reductions up to 32% under windy conditions.

Third, an integrated longitudinal–lateral NMPC framework was developed to explicitly preserve wake alignment under crosswinds; compared with the baseline trials, fuel savings improved by up to 3.2% and tailpipe NO_x outcomes improved by up to 50.4% through reduced spacing errors, lateral offsets and suppressed torque transients.

Finally, the thesis extended platooning control to electrified powertrains (P2 hybrid, plug-in hybrid, and battery electric) via a two-level hierarchical MPC that coordinated platoon motion and optimized powertrain energy and emissions; simulations showed that electrification systematically amplified platooning benefits, with energy savings up to 12.4% and tailpipe NO_x reductions up to 40.9% for plug-in hybrid electric follower.

Overall, the results demonstrated that combining cooperative platoon coordination with powertrain-integrated optimization provides a practical pathway to safe, energy-efficient, and low-emission long-haul freight operations under realistic highway conditions.

Preface

This thesis is an original work by Luo Jiang. The research was conducted utilizing available computing systems and experimental equipment at the University of Alberta Energy Mechatronics Lab (EML) led by Prof. Mahdi Shahbakhti. The Cooperative Truck Platooning System (CTPS) trials were conducted in collaboration with Alberta Motor Transport Association, Transport Canada, Alberta Transportation, Pronto AI, Bison Transport, and Tantus.

Each chapter is fully or partially based on published or submitted papers in peer-reviewed journals and conference proceedings [1–12]. As the lead author, I made substantial contributions to all peer-reviewed publications referenced throughout this thesis. The specific contributions of co-authors to each paper are summarized below.

In Chapter 2 [1, 2], Dr. Javad Kheyrollahi contributed to platooning data analysis, and Dr. Charles (Bob) Koch provided valuable insights regarding the fuel consumption and NO_x emissions.

In Chapter 3 [7], Dr. Saeid Shahpouri contributed to the development of the long short-term memory (LSTM) model for tailpipe NO_x emissions prediction.

In Chapter 7 [6, 8], Sina Moghadasi contributed to the electrified truck modelling.

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List of Symbols

α	True wind angle [rad]
β	Apparent wind angle [rad]
$\ddot{x}$	Longitudinal acceleration of the vehicle [m/s ²]
$\ddot{y}$	Lateral acceleration of the vehicle [m/s ²]
Δt	Sampling time [s]
δ	Steering angle [rad]
δ_m	Rotational mass coefficient [-]
$\dot{\phi}$	Yaw rate [rad/s]
$\dot{m}_f$	Fuel mass flow rate (kg/s)
η_m	Electric motor efficiency [-]
η_t	Driveline efficiency [-]
μ	Rolling resistance coefficient [-]
ω_e	Engine angular speed [rad/s]
ω_m	Motor angular speed [rad/s]
ω_w	Wheel angular speed [rad/s]
ϕ	Yaw angle [rad]
ρ	Air density [kg/m ³]
σ_x	Longitudinal slip ratio [-]
τ_e	Engine torque response time constant [-]

τ_g	Time gap [s]
θ	Road angle [rad]
A	Frontal area of the vehicle [m ²]
A_s	Side projection areas of the vehicle [m ²]
C_σ	Longitudinal tire stiffness [N]
C_D	Air drag coefficient [-]
C_S	Aerodynamic side force coefficient [-]
d_0	Standstill safe distance [m]
d_s	Desired spacing [m]
e_l	Lateral offset [m]
e_s	Spacing error [m]
F_b	Braking force [N]
F_g	Gravitational force [N]
F_r	Rolling resistance [N]
F_t	Tractive force at the wheels [N]
F_{wy}	Lateral aerodynamic drag [N]
F_w	Aerodynamic drag [N]
g	Gravitational acceleration [m/s ²]
I	Yaw moment of inertia [kg·m ²]
I_b	Current [A]
i_0	Final drive ratio [-]
i_g	Gear ratio [-]
k	Current time step [-]

K_b	Proportional coefficient for braking [-]
L	Vehicle length [m]
l	Distance between wheel axle and CG of the vehicle [m]
m	Vehicle mass [kg]
N_c	Control horizon [-]
N_p	Prediction horizon [-]
P_b	Battery power [kW]
P_{elec}	Electric power [kW]
P_b	Brake chamber pressure [kPa]
Q_{nom}	Nominal battery capacity [Ah]
R_b	Battery resistance [Ω]
r_{eff}	Effective tire radius [m]
s	Scale factor [-]
T_b	Brake torque [N·m]
$T_{e,des}$	Desired engine torque [N·m]
T_e	Engine torque [N·m]
T_m	Motor torque [N·m]
v_a	Apparent wind velocity [m/s]
v_w	True wind velocity [m/s]
v_x	Longitudinal vehicle speed [m/s]
V_{oc}	Open circuit voltage [V]
v_{wh}	Vehicle wheel speed [m/s]
v_{wx}	Longitudinal wind speed [m/s]

v_{wy}	Lateral wind speed [m/s]
w	Tuning weight [-]
x	Longitudinal position of the vehicle [m]
y	Lateral position of the vehicle [m]

Abbreviations

ACC	Adaptive Cruise Control
BEV	Battery Electric Vehicle
BSFC	Brake-Specific Fuel Consumption
CACC	Cooperative Adaptive Cruise Control
CAN	Controller Area Network
CARB	California Air Resources Board
CDS	Collaborative Driving System
CFD	Computational Fluid Dynamics
CG	Center of Gravity
CO₂	Carbon Dioxide
CPU	Central Processing Unit
CTG	Constant Time Gap
CTPS	Cooperative Truck Platooning System
DAQ	Data Acquisition
DATP	Driver Assistive Truck Platooning

DEF	Diesel Exhaust Fluid
DOC	Diesel Oxidation Catalyst
DPF	Diesel Particulate Filter
ECCC	Environment and Climate Change Canada
ECU	Electronic Control Unit
EGR	Exhaust Gas Recirculation
EM	Electric Machine
EPA	Environmental Protection Agency
GHG	Greenhouse Gas
GPS	Global Positioning System
GRU	Gated Recurrent Unit
GVW	Gross Vehicle Weight
HEV	Hybrid Electric Vehicle
IARC	International Agency for Research on Cancer
ICE	Internal Combustion Engine
IMU	Inertial Measurement Unit
IVHS	Intelligent Vehicle/Highway System
LCC	Lane Centering Control
LCV	Long Combination Vehicle

LQR	Linear Quadratic Regulator
LRR	Low-Rolling-Resistanc
LSTM	Long Short-Term Memory
LTE	Long-Term Evolution
MAPE	Mean Absolute Percentage Error
MPC	Model Predictive Control
MY	Model Year
NACFE	North American Council for Freight Efficiency
NH₃	Ammonia
NMPC	Nonlinear Model Predictive Control
NO_x	Nitrogen Oxides
NREL	National Renewable Energy Laboratory
PATH	Partners for Advanced Transit and Highways
PHEV	Plug-in Hybrid Electric Vehicle
PID	Proportional–Integral–Derivative
PM	Particulate Matter
RAM	Random Access Memory
RNN	Recurrent Neural Network
RTK	Real-Time Kinematic

RWIS	Road Weather Information System
SAE	Society of Automotive Engineers
SCR	Selective Catalytic Reduction
SMC	Sliding-Mode Control
SOC	State of Charge
SQP	Sequential Quadratic Programming
V2I	Vehicle-to-Infrastructure
V2V	Vehicle-to-Vehicle
VCU	Vehicle Control Unit
VTG	Variable Time Gap
WHO	World Health Organization
ZEV	Zero-Emission Vehicle

PART I: Introduction and Literature Review

Chapter 1

Introduction and Background

Trucking is a crucial pillar of North America’s freight transportation system, where 56% of the U.S.-Canada trade and 73% of the U.S.-Mexico trade were transported by trucks in 2024 [13]. Domestically, trucks moved about 68% of freight shipments in the United States [14] and roughly 90% in Canada [15]. These figures underscore the truck sector’s essential role in economic activity and supply-chain resilience.

Currently, diesel engines dominate the commercial heavy-duty trucks, powering 97% of Class-8 trucks [3]. Correspondingly, heavy-duty trucks account for a substantial share of transportation energy use, on the order of 19% in the United States and approximately 30% in Canada [16]. At the same time, heavy-duty trucks are also the major source of greenhouse gas (GHG) emissions, contributing 25% of total transport emissions, and it is anticipated to see larger projected increases as freight demand grows in the future [17]. Without additional interventions, the truck sector is not on a trajectory consistent with net-zero emissions by 2050.

In response, regulators have tightened emissions requirements for heavy-duty vehicles and introduced zero-emission vehicle (ZEV) sales mandates. In the United States, the Environmental Protection Agency (EPA) Phase-3 standard targets a 40% reduction in tailpipe carbon dioxide (CO₂) emissions for model year (MY) 2032 relative to the Phase 2 MY 2027 levels [18]. In parallel, the regulatory framework retains the non-CO₂ GHG emission standards from Phase 2; for example, nitrogen oxides (NO_x)

are limited to the range of 0.035-0.05 g/bhp-hr depending on operating conditions [19]. At the state level, the California Air Resources Board (CARB) requires that all new medium- and heavy-duty trucks sold in California to be ZEVs by 2045 [20]. In Canada, Environment and Climate Change Canada (ECCC) regulates GHG reduction targets for model years 2021–2027 aligned with the U.S. EPA Phase-2 standards and plans post-2027 alignment with the latest U.S. requirements [21], while Canada targets 100% of new medium- and heavy-duty vehicle sales to be ZEVs by 2040 [22].

Nevertheless, given the scale of freight demand, the longevity of heavy-duty assets, and the multi-decade transition required for infrastructure and fleet turnover, near- and mid-term strategies that reduce energy consumption and emissions from the existing and newly sold combustion-dominant trucks remain indispensable. Accordingly, improving heavy-duty truck energy conversion efficiency while reducing tailpipe emissions continues to be a priority throughout the transition period.

1.1 Fuel Consumption and Tailpipe Emissions of Heavy-Duty Trucks

The early fuel efficiency benchmarking in Canada’s trucking industry survey reported that, in 1999, the annual fuel consumption among participating intercity fleets ranged from 33.2 to 52.9 L/100 km, with an average of about 39.5 L/100 km. The most fuel-efficient fleets were primarily operated in the relatively flat terrain of southern Ontario. Approximately 40% of fleets reported year-over-year improvements in fuel efficiency from 1997 to 1999, and seasonal effects were non-negligible, with fuel use typically increasing by 3–5 L/100 km in winter relative to summer. These early results highlighted both the inherently high energy demand of heavy-duty trucking and the substantial variability across fleets. They also motivated the adoption of standardized performance indicators (e.g., L/100 ton·km) to enable more consistent and comparable assessments of fleet fuel efficiency [23]. More recent benchmarking by Natural Resources Canada and the SmartWay program suggests continued im-

provement in average Class-8 fleet fuel consumption, declining from 45.3 L/100 km in 2012 to 38.4 L/100 km in 2016, corresponding to an average annual improvement of about 3.8%. In addition, freight-normalized metrics were reported to better reflect productivity and load factors. SmartWay data from 2014–2017 indicate that for-hire Canadian fleets operated at 3.34–3.88 L/100 ton·km, whereas private fleets used 4.27–5.48 L/100 ton·km over the same period; on average, for-hire carriers consumed roughly 30% less fuel per ton·km. This difference may partially reflect more cost-reduction practices among for-hire carriers, such as route optimization and back-hauling [24].

Telematics-derived datasets provide a complementary, spatially resolved view of contemporary on-road performance. Geotab’s Class 7–8 fuel-economy benchmark reports average heavy-truck fuel consumption by province and state, as summarized in Table 1.1 [25]. In Canada, provincial averages range from 40.5 to 52.2 L/100 km, yielding a national average of approximately 45 L/100 km; in the United States, state averages span 36.9 to 42 L/100 km with a national average of 38.6 L/100 km. Overall, the Geotab results highlight substantial regional variation, but also confirm that typical real-world operations still lag significantly behind best-practice fleets.

The latest fleet-level studies by the North American Council for Freight Efficiency (NACFE) report that the average fuel consumption of its long-haul study fleets reduced to 30.9 L/100 km in 2022 and 30.3 L/100 km in 2023 through the adoption of fuel-efficiency technologies and operational measures, corresponding to year-over-year improvements of 4.2% and 2.0%, respectively [26]. These results suggest that fleet efficiency continues to improve, although recent gains appear to be moderating relative to earlier periods.

Table 1.1: Average truck fuel consumption in Canada and the United States.

Country	Province/State	Avg fuel consumption	Country	Province/State	Avg fuel consumption
Canada	Alberta	49.5 (L/100 km)	USA	Minnesota	37.3 (L/100 km)
Canada	British Columbia	52.2 (L/100 km)	USA	Mississippi	37.1 (L/100 km)

Country	Province/State	Avg fuel consumption	Country	Province/State	Avg fuel consumption
Canada	Manitoba	42.5 (L/100 km)	USA	Missouri	38.1 (L/100 km)
Canada	New Brunswick	40.7 (L/100 km)	USA	Montana	37.1 (L/100 km)
Canada	Nova Scotia	40.5 (L/100 km)	USA	Nebraska	37.3 (L/100 km)
Canada	Ontario	44.8 (L/100 km)	USA	Nevada	38.4 (L/100 km)
Canada	Quebec	44.3 (L/100 km)	USA	New Hampshire	37.4 (L/100 km)
Canada	Saskatchewan	48.2 (L/100 km)	USA	New Jersey	38.9 (L/100 km)
USA	Alabama	39.4 (L/100 km)	USA	New Mexico	40.1 (L/100 km)
USA	Arizona	38.2 (L/100 km)	USA	New York	40.4 (L/100 km)
USA	Arkansas	37.5 (L/100 km)	USA	North Carolina	37.7 (L/100 km)
USA	California	40.2 (L/100 km)	USA	North Dakota	40.7 (L/100 km)
USA	Colorado	39.5 (L/100 km)	USA	Ohio	39.6 (L/100 km)
USA	Connecticut	37.7 (L/100 km)	USA	Oklahoma	39.2 (L/100 km)
USA	Delaware	37.0 (L/100 km)	USA	Oregon	37.6 (L/100 km)
USA	Florida	39.3 (L/100 km)	USA	Pennsylvania	40.1 (L/100 km)
USA	Georgia	38.5 (L/100 km)	USA	South Carolina	37.1 (L/100 km)
USA	Idaho	38.8 (L/100 km)	USA	South Dakota	38.1 (L/100 km)
USA	Illinois	38.8 (L/100 km)	USA	Tennessee	37.9 (L/100 km)
USA	Indiana	38.6 (L/100 km)	USA	Texas	42.0 (L/100 km)
USA	Iowa	38.6 (L/100 km)	USA	Utah	38.8 (L/100 km)
USA	Kansas	39.3 (L/100 km)	USA	Vermont	39.9 (L/100 km)
USA	Kentucky	38.0 (L/100 km)	USA	Virginia	38.0 (L/100 km)
USA	Louisiana	38.8 (L/100 km)	USA	Washington	40.8 (L/100 km)
USA	Maine	39.1 (L/100 km)	USA	West Virginia	38.2 (L/100 km)
USA	Maryland	37.4 (L/100 km)	USA	Wisconsin	37.0 (L/100 km)
USA	Massachusetts	36.9 (L/100 km)	USA	Wyoming	37.7 (L/100 km)
USA	Michigan	40.1 (L/100 km)			

Beyond its substantial demand for diesel fuel, the trucking sector imposes significant environmental and public-health burdens. Emissions from diesel-powered trucks contribute to both air-quality degradation and climate change. CO₂ is a dominant anthropogenic greenhouse gas, and for diesel combustion it is closely coupled to fuel consumption through the carbon content of the fuel. In addition, diesel exhaust has been classified as carcinogenic to humans by the International Agency for Research on Cancer (IARC), a specialized agency of the World Health Organization (WHO) [27]. Among the emitted pollutants, NO_x are of particular concern because they contribute to the formation of ground-level ozone and secondary particulate matter, thereby adversely affecting air quality and public health.

Fleet benchmarking further illustrates both progress and remaining gaps. The SmartWay program reported that, from 2014 to 2016, the Class-8 truck carrier partners reduced CO₂ intensity in both distance-normalized and freight-normalized terms, decreasing from 1023 to 953 g/km and from 122 to 69 g/ton·km, respectively [24]. Over the same period, SmartWay also reported reductions in NO_x emissions across truck body types in 2016 relative to 2015; for example, dry-van operations decreased from 0.262 to 0.248 g/ton·km [24]. Nevertheless, these levels may still fall short of applicable regulatory targets (EPA’s Phase 1 Standards) for model years 2014-2016 Class-8 trucks [28], indicating that further reductions remain necessary. Moreover, the U.S. EPA Phase 2 and Phase 3 standards introduce progressively more stringent CO₂ requirements for heavy-duty vehicles over MY 2021–2027 and MY 2027–2032, respectively, raising the bar for both manufacturers and fleet operators and implying that additional technology and operational measures will be required to achieve compliance. Collectively, these observations reinforce that reducing fuel consumption and tailpipe emissions from heavy-duty trucks is both an economic and societal imperative: lower fuel use decreases operating costs and CO₂ emissions, while reductions in NO_x deliver immediate air-quality and public-health benefits.

1.1.1 Fuel Saving Strategies for Heavy-Duty Trucks

Fuel consumption of heavy-duty trucks is governed by road-load forces and powertrain conversion efficiency. At typical highway speeds, the dominant contributors to tractive energy demand are aerodynamic drag and tire rolling resistance, while drivetrain and engine efficiencies determine how effectively this demand is converted into vehicle motion. Accordingly, a range of fuel-saving strategies has been adopted in modern heavy-duty trucks, including aerodynamic devices, long combination vehicles, platooning technologies, etc.

Tractor and Trailer Aerodynamic Devices

Aerodynamic drag accounts for the largest fraction of truck energy losses at highway speeds. Modern tractor designs have therefore become increasingly streamlined as manufacturers refine key geometric features such as the front profile, windshield angle, hood slope, bumpers, and mirrors, and deploy add-on devices including roof fairings and fuel tank side fairings, as shown in Figure 1.1. These developments have reduced the average tractor drag coefficient from approximately 0.80 to 0.65 (about a 19% reduction) compared with more traditional tractor styling [29]. While improvements to tractor aerodynamics can yield substantial fuel savings, further reductions are often achieved through trailer-focused devices (e.g., trailer fairings in Figure 1.1), which have been reported to provide additional fuel-economy gains of up to approximately 9% or more under representative highway conditions [30].

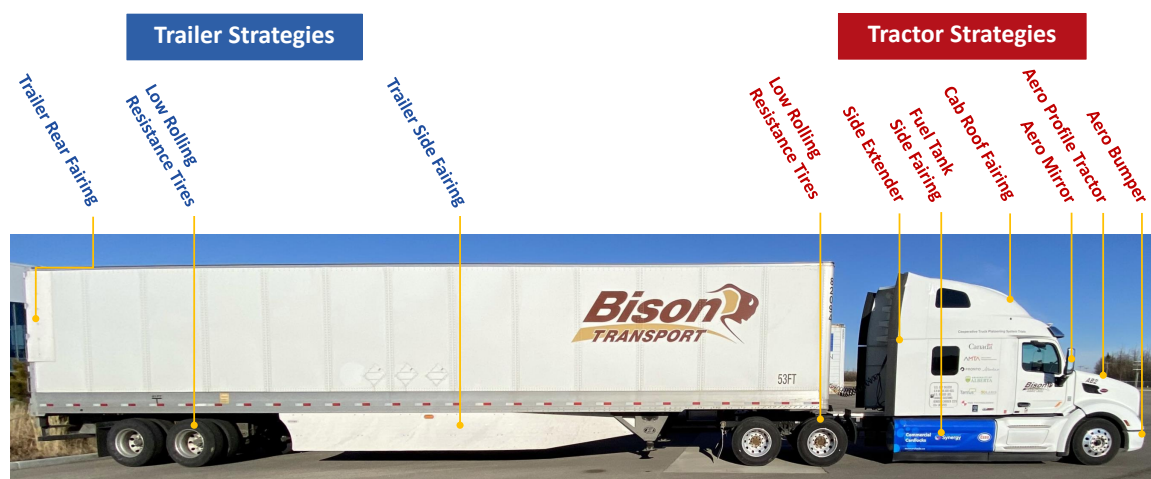


Figure 1.1: Fuel saving strategies on a MY 2019 Peterbilt Class-8 tractor-trailer truck.

Several retrofit and factory-option aerodynamic devices are widely used on tractor-trailer combinations:

- **Roof fairing:** an integrated air deflector mounted on the cab roof to guide flow over the trailer front.
- **Side extender fairings:** devices that mitigate the aerodynamic penalty asso-

ciated with the tractor–trailer gap; reported studies indicate fuel-consumption reductions of approximately 2–3% [31].

- **Aerodynamic bumpers and fuel-tank side fairings:** components that reduce drag-generating protrusions and underbody turbulence.
- **Trailer side fairings (side skirts):** panels installed along the lower trailer edges (typically between axles) to reduce underbody flow and turbulence; studies report fuel-economy improvements of about 3–7% [32].
- **Trailer rear fairings (boat tails):** devices mounted at the trailer rear to reduce base drag; reported fuel-economy improvements are typically on the order of 3–5% [33].

Overall, a Class-8 tractor–trailer equipped with a suite of aerodynamic devices can realize an average fuel-economy improvement of approximately 11% relative to a representative long-haul baseline configuration [29].

Long Combination Vehicles

Tire rolling resistance represents a major component of the tractive energy demand for heavy-duty trucks, particularly under high gross weights. Beyond the adoption of low-rolling-resistance (LRR) tires, long combination vehicles (LCVs) have been deployed as an operational strategy to improve freight efficiency by increasing payload per tractor unit. As illustrated in Figure 1.2, LCV configurations can reduce the number of tractors required to move a given amount of freight, thereby lowering total powertrain losses and auxiliary fuel consumption. In addition, the relatively close spacing between trailers can reduce the aerodynamic drag of the overall combination, yielding further fuel savings.

Consistent with these mechanisms, test data indicate that, compared to two single-trailer vehicles operating independently, LCVs can achieve fuel savings of up to 28%



Figure 1.2: A long combination vehicle consists of one tractor and two trailers. Image generated with Gemini 3 (Google, 2026) based on an original image by Luo Jiang.

[34]. Despite these benefits, LCV deployment is constrained by regulatory and operational limitations. LCVs are typically restricted to designated routes, are subject to speed and load limits, and may be prohibited from operating under adverse weather conditions. Moreover, some jurisdictions impose large minimum separation requirements between LCVs and surrounding traffic, with mandated separations reported in the range of 150–1000 m [35]. Due to these stringent conditions and associated infrastructure and safety considerations, LCV operations remain limited to specific corridors and jurisdictions.

Truck Platooning

Truck platooning enables multiple heavy-duty trucks to travel cooperatively as a wirelessly connected convoy and has emerged as a promising approach to further reduce energy consumption beyond existing fuel-saving technologies (e.g., aerodynamic retrofits and LCV operations). As shown in Figure 1.3, the follower truck in a platoon experiences reduced aerodynamic drag by traveling in the lead truck’s slipstream, which directly contributes to fuel savings. In addition to aerodynamic benefits, trucks in a platoon can coordinate with each other through vehicle-to-vehicle (V2V) com-

munication to optimize speed profiles, minimizing unnecessary speed fluctuations and further reducing fuel consumption [11].



Figure 1.3: Demonstration of a two-truck platoon. Photo courtesy of Alberta Motor Transport Association (AMTA).

Experimental tests and field demonstrations have shown that platooning can reduce aerodynamic drag across multiple trucks in the platoon, although the magnitude of benefit depends strongly on the platoon configuration. In general, the largest drag reductions occur for vehicles located between the first and last positions in longer platoons, while in a two-truck platoon the follower typically receives the dominant aerodynamic benefit. The tests and demonstrations literature reviewed in this thesis indicates fuel consumption reductions in the range of 5%–21%, with outcomes varying as a function of inter-vehicle spacing, vehicle speed, vehicle number within the platoon, and vehicle mass. Furthermore, CO_2 emissions are directly coupled to fuel consumption; approximately 2.7 kg of CO_2 is emitted per liter of diesel fuel combusted [36]. Consequently, fuel savings achieved through platooning translate directly into reductions in CO_2 emissions.

These characteristics make truck platooning a particularly attractive pathway for achieving scalable efficiency gains under real-world freight operations. Motivated by this potential, a central focus of this thesis is the analysis and design of energy-efficient

cooperative truck platooning systems.

1.1.2 NO_x Reduction Strategies for Heavy-Duty Trucks

A variety of NO_x reduction techniques ranging from in-cylinder to aftertreatment techniques have been commercially implemented on modern diesel trucks to comply with increasingly stringent emissions regulations. In-cylinder approaches primarily aim to suppress NO_x formation by modifying the combustion environment, whereas aftertreatment systems convert NO_x in the exhaust stream before it is released to the atmosphere. However, real-world NO_x outcomes depend not only on component-level deNO_x capability but also on transient engine operation, and the control strategies that govern vehicle speed and load. In this regard, truck platooning introduces an additional, system-level opportunity to further reduce NO_x through coordinated motion and power-demand management, complementing existing deNO_x techniques.

Engine-In Cylinder Techniques

As shown in Figure 1.4, in a Class-8 diesel truck engine, exhaust gas recirculation (EGR) is used to reduce engine-out NO_x emissions by recirculating a portion of the exhaust gas back into the intake. The recirculated exhaust gas is largely inert and dilutes the intake charge, which lowers the oxygen concentration in the combustion chamber and absorbs heat from the combustion process. This in turn reduces the peak combustion temperature, and the cooler in-cylinder conditions achieved with EGR result in significantly less NO_x being produced. Modern heavy-duty diesel engines employ cooled EGR systems, in which the exhaust gas passes through a heat exchanger (i.e., EGR cooler) before re-entering the intake; this cooling further lowers the gas temperature and thereby enhances the NO_x reduction effectiveness of the EGR process. A recent study conducted by Jin et al. [37] reported that applying a high EGR rate (30% of the intake flow) on a heavy-duty diesel engine dropped NO_x emissions from 8.25 g/kWh down to 5.54 g/kWh, corresponding to an approximately 33%

reduction in engine-out NO_x . However, higher EGR rates can introduce important trade-offs, including increased pumping losses, potential reductions in combustion efficiency and power output, and elevated particulate-matter (PM) emissions, which may adversely affect fuel consumption and engine durability [38]. Consequently, EGR is often optimized in combination with other measures.

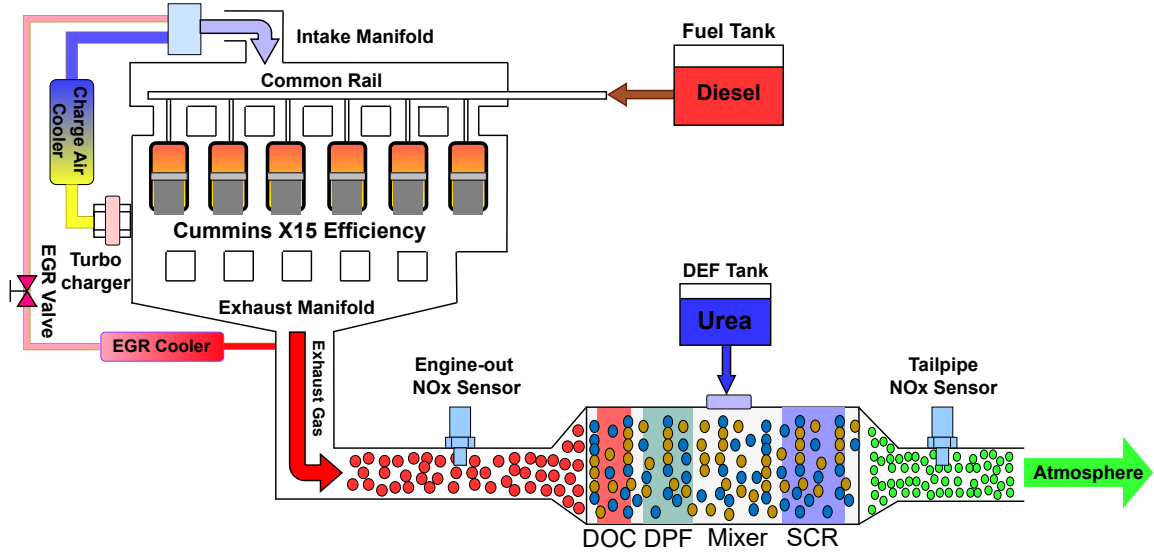
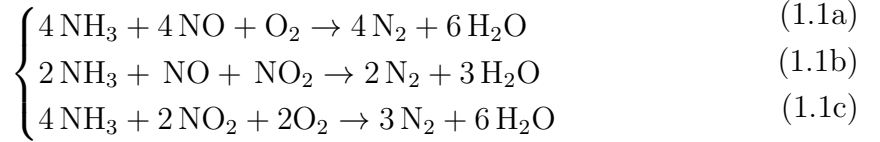


Figure 1.4: Diagram of NO_x emissions reduction techniques for Class-8 diesel truck engine. EGR: exhaust gas recirculation; DOC: diesel oxidation catalyst; DPF: diesel particulate filter; SCR: selective catalytic reduction; DEF: diesel exhaust fluid. It is noted that the diagram is schematic and does not represent the actual physical layout of the engine; it is included only for illustration of the system’s working principle.

Aftertreatment Systems

Another effective and widely adopted approach for reducing NO_x emissions from diesel-powered heavy-duty trucks is exhaust aftertreatment. Selective catalytic reduction (SCR) is the primary system used to achieve high NO_x conversion under a wide range of operating conditions. SCR employs urea-based diesel exhaust fluid (DEF) as a reductant source; after injection into the exhaust stream upstream of the catalyst, DEF undergoes thermal decomposition and hydrolysis to form ammonia (NH_3), which serves as the active reducing agent. The operating principle of SCR is illustrated in Figure 1.4. Specifically, DEF is injected upstream of a mixing section

to promote atomization and uniform distribution. The injected urea then vaporizes and decomposes, generating NH_3 , which is subsequently adsorbed on the SCR catalyst surface and reacts with NO_x . Through the de NO_x reactions in (1.1a)–(1.1c), NO_x is reduced primarily to nitrogen and water [7]. Reported studies indicate that SCR systems using ammonia derived from injected urea can achieve NO_x conversion efficiencies exceeding 90% under favorable operating conditions [39, 40].



Platooning Strategies

The advent of truck platooning provides a system-level opportunity to further reduce NO_x emissions by complementing established in-cylinder and aftertreatment de NO_x techniques. Specifically, platoon connectivity enables vehicles to share motion and operating information in real time, which can be exploited within predictive control frameworks to proactively shape future power demand and exhaust conditions in emissions-favorable directions.

Simulation studies provide evidence of this potential. Pramanik et al. [41] designed a platoon controller using a multi-agent framework, and simulation results for a three-truck platoon reported a 13% reduction in engine-out NO_x emissions. Ma and Wang [42] developed one of the first dedicated aftertreatment controllers for diesel vehicle platooning by explicitly incorporating platoon powertrain dynamics. Their controller leveraged predictive information of exhaust conditions at the SCR inlet to regulate ammonia coverage and improve NO_x conversion efficiency, achieving a 13% reduction in tailpipe NO_x compared with a pre-calibrated map-based strategy.

Despite these promising results, relatively limited effort has been devoted to tailpipe NO_x reduction at the platoon-coordination level. This gap motivates the focus of this thesis on platoon-coordination level strategies to further reduce tailpipe

NO_x under real-world operating conditions.

1.2 Truck Platooning Fundamentals and Evidence

Truck platooning offers a promising pathway for enhancing energy savings and reducing exhaust emissions under real-world freight operations. As a result, truck platooning has attracted considerable and growing attention from both academia and industry. This section summarizes key experimental evidence from prior tests and trials and highlights major factors influencing fuel consumption and NO_x emissions.

1.2.1 Prior Truck Platooning Tests and Trials

A wide range of closed-track tests and on-road trials have been conducted worldwide to explore truck platooning benefits. Table 1.2 summarizes the major prior truck platooning activities. The European Commission’s PROMOTE CHAUFFEUR project initiated truck platooning exploration from 1996 to 2003 with two phases (I and II). Demonstrations on test tracks and in mixed traffic showed fuel savings of up to 21% [43, 44]. The California Partners for Advanced Transit and Highways (PATH) program embarked upon its research on truck platooning in 2000. Two identical Freightliner tractors towing 53-foot trailers, linked by an electronic control system with only an automated longitudinal control function, were tested on an unused airfield runway in Crows Landing, California. Results indicated an average fuel saving ranging from 5% to 12% for the follower truck [45]. In 2008, Japan launched the five-year project Energy ITS, which investigated the energy consumption of heterogeneous truck platoons composed of three heavy trucks and a light truck. The tests on a closed track and along an expressway before public use demonstrated fuel savings of up to 18% [46, 47]. Comparable fuel-saving trends have also been reported in other closed-track experiments [34, 48]. Furthermore, a Cummins-led truck platooning project was launched to assess its performance under real-world driving conditions, and it was found that on-road platooning resulted in 3.6% lower fuel savings of the follower

truck compared to test-track conditions [49]. Similarly, an on-road trial conducted by TNO reported average fuel savings of about 4–6% [50]. Overall, the field-test literature confirms the energy-saving potential of truck platooning, while also indicating that realized benefits depend strongly on operating conditions.

Table 1.2: Major prior truck platooning trials.

Project/ Organization	Time Gap (sec)	Speed (km/h)	Max Lateral Offset* (m)	Fuel Saving ** (%)	NO_x Reduction *** (%)	# Truck
CHAUFFEUR	0.3 - 0.96	60, 80	-	10 - 21	-	2, 3
PATH	0.12 - 0.4	80, 89	0.5	5 - 12	-	2
Energy ITS	0.21 - 0.45	80	-	8 - 18	-	3, 4
SARTRE	0.2 - 0.6	90	-	8 - 16	-	2, 3
NREL	0.2 - 0.8	89, 105, 113	-	2.8 - 9.7	-25 to -14	2, 3
Peloton	0.38	103	-	10	-	2
Cummins	0.6	105	-	2.6 - 4.3	-	2, 3
EDDI	0.7	80	-	3 - 4	-	2
DATP	0.31 - 1.58	105	0.6	4.5 - 7	-6 to 13	2, 3
NRC	0.6 - 1.5	89, 105	1.3	5.2 - 7.8	-	2
ENSEMBLE	1.4 - 2	85	-	5 - 7	-	2, 3, 4
TNO	1.49 - 2.25	80	-	4 - 6	-	2
WABCO	0.42 - 3	85	-	8.3 - 11.7	-	2
CACC	0.14 - 3	105	-	8 - 13	-	2
Scania	1 - 5	90	-	1.3 - 7.1	-	2, 3

* Lateral offset refers to the lateral deviation of a follower truck from perfect alignment with the center line of its preceding truck.

** The fuel saving here is for follower trucks in a platoon.

*** The NO_x reduction here is for follower trucks in a platoon. NO_x reduction with a minus sign indicates an increase in NO_x emissions for follower trucks.

Beyond fuel savings, several studies have examined NO_x emissions due to their adverse impacts on air quality and public health. Track tests conducted by the National Renewable Energy Laboratory (NREL) reported increases in engine-out NO_x emissions for the follower truck in the range of 14%–25% [51]. In contrast, Driver Assistive Truck Platooning (DATP) tests reported engine-out NO_x changes spanning both reductions and increases (approximately -6% to 13%) [52]. Similarly, the ENSEMBLE project reported both increases (0.25 g/km) and decreases (0.21 g/km) in tailpipe NO_x when comparing two-truck platooning on a test track with solo driving [53]. These findings indicate that NO_x impacts are not consistently beneficial in the

field tests, which differs from the more uniformly favorable trends often reported in simulation studies. A key reason is that real-world platooning introduces additional disturbances that can induce fluctuations in the engine torque demand.

1.2.2 Factors Affecting Platooning Fuel Savings

Fuel savings in truck platooning depend on multiple interacting factors, including vehicle mass, tractor-trailer configuration, platoon size, and operating conditions. Lammert et al. investigated the effect of truck weight on fuel consumption during platooning, and the test results indicated that the gross vehicle weight (GVW) of 65,000 lbs had extra 2.5% fuel saving compared to a GVW of 80,000 lbs at the speed of 105 km/h with 15 m following distance [54]. A similar trend was reported in the project of Energy ITS, where loaded trucks in a platoon experienced a 5% reduction in fuel saving compared to the trucks with empty trailers at the speed of 80 km/h with a spacing of 10 m [47]. This is because the heavier truck requires higher engine torque to overcome additional rolling resistance and gravitational forces. In addition, increased load makes acceleration and deceleration events become more energy-intensive, reducing the effectiveness of coordinated driving. These findings highlight the critical role of vehicle load in affecting resistant forces and powertrain response, underscoring the importance of integrating powertrain-aware coordination into the design of platoon control systems to maximize fuel conversion efficiency under varying load conditions. The configuration of the tractor-trailers also affects platoon fuel savings. For example, trucks equipped with aerodynamic devices of side skirts and boat tails achieved a 2% increase in fuel savings compared to those using standard trailers [55]. The majority of the truck platooning trials to date have considered two-truck platoons, although platoons with three or more trucks were demonstrated in a few projects. It was found that the platoon size more than three trucks achieved greater net fuel savings than a two-truck platoon, but the fuel saving differences diminished at larger separation distances [56].

Beyond vehicle- and formation-level factors, operating conditions such as inter-vehicle time gaps, road grades, varying winds and lateral alignments have significant impacts on fuel savings for truck platooning in real-world driving conditions.

Time Gaps

A major current focus of fuel savings for truck platooning lies in the time gaps (separation distances) among trucks. When trucks travel together within a certain distance, the follower trucks experience reduced aerodynamic drag, which contributes to fuel savings. As the separation distances decrease, the fuel savings of platooning trucks typically increase [57]. In the truck platooning tests conducted by McAuliffe et al., a fuel saving of 13% was observed at a separation distance of 4 m (0.14-sec time gap at 105 km/h), compared to a 4.5% fuel saving at the larger separation distance of 87 m (3-sec time gap at 105 km/h) [34]. This trend is further supported by additional experimental evidence [58–62]. Thus, many researchers have focused on maximizing fuel savings by reducing aerodynamic drag with small time gaps. It is noted that most of the prior truck platooning tests were conducted in closed tracks/runways, or on public roads under escort. While smaller time gaps between trucks can lead to greater fuel savings, they also present challenges to platooning safety, particularly in public traffic and commercial operations.

Computational fluid dynamics (CFD) and wind tunnel studies demonstrated that the follower truck in a two-truck platoon can still experience reduced aerodynamic drag at separations up to 152 m (equivalent to 5 sec at speed of 110 km/h) [34, 63]. Furthermore, recent field studies have explicitly evaluated fuel saving benefits of truck platooning at larger time gaps. WABCO researchers conducted a series of on-track tests to evaluate the fuel economy improvements of a two-truck platoon with time gaps ranging from 0.42 to 3 sec. Within the European legal minimum time gap of 2 sec, the follower truck achieved significant fuel savings of 10% to 11.7%. The variation in fuel savings across this range was only 1.7%, suggesting that large-spacing platooning

only lose limited fuel benefits. Furthermore, when the time gap was extended to 3 sec, fuel savings decreased slightly but still remained substantial at 8.3% [48]. Similarly, a three-truck platoon was tested at the speed of 105 km/h on a closed track at Transport Canada’s Motor Vehicle Test Centre with time gaps varying from 0.14 to 3 sec. The maximum fuel savings for the follower truck was recorded at approximately 13% with a 0.4-sec time gap, while a measurable fuel saving of 8% was still observed at the largest time gap of 3 sec. These findings confirm that large-spacing truck platoons can still gain considerable fuel-saving benefits, while alleviating the safety concerns associated with close-spacing configurations.

Road Grades

The majority of the prior research and experimental evaluations of fuel-efficient platooning have been conducted under near-ideal roadway conditions (i.e., flat roads). In practice, however, road grade can substantially affect platoon energy efficiency by altering the required tractive effort and the relative importance of aerodynamic versus gravitational components of road load. Experimental evidence indicates that fuel consumption increases on road sections with high road grades ($\pm 2\%$) [64]. Consistent with this observation, Taylor et al. [65] reported that platoon fuel savings (relative to a single truck) decreased from 6.9% on flat ground to 4.9% when operating over a section with grades varying from -3% to 1% . Similarly, Ward et al. [66] found that platooning vehicles achieved fuel savings of 3.9%–6.5% over small gradients, whereas no fuel savings were observed over steep terrain. In more adverse cases, Turri et al. [67] reported an increase of approximately 4% in the follower vehicle’s fuel consumption relative to solo driving.

This degradation primarily stems from the lack of coordination between the lead and follower trucks on roads with varying slopes, which leads to non-optimal deceleration and acceleration behaviors. Varying road grades affect power demand and braking intensity, with uphill segments increasing engine load and downhill segments

potentially leading to greater reliance on braking. As a result, the follower truck experiences significant speed fluctuations, thereby increasing fuel use. Droege et al. [64] proposed a long-horizon predictive cruise control strategy to create a fuel-optimal speed profile for the lead truck by using the look-ahead road grade data, and the follower truck then tried to maintain the same speed profiles as the lead truck’s to keep a desired platoon gap. Their simulation results indicated an average fuel savings of 13.9% for the platoon trucks on the selected hilly terrain of Interstate 69 highway with the spacing error of more than 11 m. However, in a truck platoon at relatively large spacing, it is not optimal for the follower truck to track the same speed profiles of the lead truck, which would induce additional transients, thereby increasing the follower truck’s fuel consumption.

Varying winds

Winds represent a major and inherently uncertain disturbance in real-world operations and can significantly affect the fuel efficiency of truck platooning. Wind forces can be decomposed into longitudinal and lateral components relative to the truck’s driving direction. Longitudinal winds consist of headwinds (blowing against the truck’s direction of travel) and tailwinds (blowing in the direction of travel), while lateral winds include crosswinds. Crosswinds can introduce lateral forces [68] and yaw moments [69], potentially destabilizing the platoon. From an energy perspective, crosswinds can also reduce platoon fuel savings by inducing lateral offsets [70] and increasing drag coefficient [71]. Wind-tunnel and track-based studies further indicate that platoon aerodynamic interactions remain beneficial under windy conditions, but the magnitude of benefit depends on wind speed and direction; for example, an 8% reduction in wind-averaged drag coefficient was identified in [72].

Longitudinal winds, including headwinds and tailwinds, significantly affect aerodynamic drag by directly increasing or decreasing the airspeed, further influencing truck fuel consumption [73]. In addition to shifting the mean drag level, wind disturbances

can induce speed and torque fluctuations within the platoon, which can increase fuel consumption as vehicles attempt to maintain a desired speed and spacing in the presence of unsteady aerodynamic loads [2]. Consequently, enhancing fuel savings for truck platooning under varying wind conditions remains an ongoing challenge.

Lateral Offsets

Lateral offsets shift the follower truck partially out of the lead vehicle’s wake and attenuate the drag-reduction benefit [74]. As summarized in Table 1.2, experiments reporting non-negligible lateral offsets typically exhibit reduced fuel savings relative to well-aligned platoons. For example, two Class-8 trucks equipped with Cooperative Adaptive Cruise Control (CACC) platooning systems were tested on highways around Montreal, Canada with the time gaps ranging from 0.63 to 3.7 sec. These systems provided longitudinal spacing control, while steering was manually performed by the drivers, which is classified as SAE Level 1 automation [75]. The test results showed that the average lateral offsets at small time gaps (0.63–0.86 sec) were close to 0.1 m; however, significantly greater deviations were observed at larger spacings, with average offsets reaching up to 0.7 m due to the manual steering by the drivers [76]. A corresponding drop in fuel savings for the follower truck was noted.

The impact of lateral misalignment has also been quantified under controlled conditions. In a closed-track experiment with a two-truck platoon, intentional lateral offsets of 0.65 m and 1.3 m were introduced to isolate their aerodynamic effect. The results demonstrated a measurable degradation in fuel-saving performance as the offset increased; at the maximum offset, where the follower truck partially straddled the adjacent lane, the absolute fuel savings were reduced by approximately 3%–4% relative to the baseline aligned condition [59]. These findings indicate that maintaining lateral alignment is important for preserving aerodynamic benefits and motivate the development of platooning control systems that explicitly regulate lateral positioning in addition to longitudinal spacing.

1.2.3 Factors Affecting Platooning NO_x Emissions

The engine-out NO_x penalties observed in the NREL project (Table 1.2) were primarily associated with rapid engine torque transients. In diesel engines, aggressive torque changes can drive temporary mismatches in air-path and EGR control, yielding excursions in in-cylinder oxygen availability and combustion temperature; these transient departures from nominal conditions can elevate NO_x formation [77]. Motivated by this mechanism, researchers from Auburn University [78] implemented a truck platooning control system to suppress the variations in engine torque rates, leading to reduced engine-out NO_x emissions. However, their results also indicated that the follower truck could still produce higher engine-out NO_x than the baseline when time gaps were smaller than 0.5 sec. This outcome is consistent with the control requirements at close spacings, where the follower must respond quickly to maintain the desired safe spacing, inducing large torque excursions and amplifying transient NO_x formation.

Engine-out NO_x emissions are fed into the aftertreatment system for further reduction, and tailpipe NO_x emissions therefore depend on both upstream engine-out production and downstream conversion efficiency. Platoon connectivity enables vehicles to share driving and operating information in real time, facilitating predictive control strategies that can shape exhaust conditions and improve aftertreatment performance. Ma and Wang [42] developed an integrated powertrain and aftertreatment control method for platooning by explicitly incorporating platoon powertrain dynamics. Their approach leveraged predicted exhaust conditions at the SCR inlet and optimized the ammonia coverage ratio to enhance NO_x conversion under varying operating conditions. Compared with conventional aftertreatment control strategies that rely on pre-calibrated maps to compute dosing inputs, the proposed method exploited connectivity-enabled prediction of powertrain and exhaust states, and a 13% reduction in tailpipe NO_x emissions was reported relative to the map-based baseline.

1.2.4 Platooning with Electrified Trucks

To support the goal of net-zero emissions by 2050, electric trucks as an important class of zero-emission vehicles are increasingly being introduced to the market. Nevertheless, the widespread adoption of fully battery-electric trucks for long-haul freight remains constrained by limited onboard energy capacity, long charging times, and insufficient charging infrastructure [79]. In the meantime, hybrid electric trucks are being actively developed and deployed as a practical transitional pathway, delivering reductions in energy consumption and GHG emissions through partial electrification while alleviating the range and refueling constraints [80]. Table 1.3 summarizes representative electrified Class-8 heavy-duty truck models.

Table 1.3: Representative electrified Class-8 heavy-duty truck models.

Company/Model	Powertrain*	Energy Source	Energy Consumption
Hylion 6X4HE	HEV	Battery (8 kWh) + Diesel	15.7 - 21.4 L/100km
Great Wall Hi4-G	HEV	Battery (12.9 kWh) + Diesel	19.9 - 29.7 L/100km
ReVolt Motors	PHEV	Battery (210 kWh) + Diesel	19.6 L/100km
Edison Motors	PHEV	Battery (280 kWh) + Diesel	12 L/100km
Peterbilt 579EV	BEV	Battery (500 kWh)	155 kWh/100km
Scania 45 R	BEV	Battery (624 kWh)	178 kWh/100km
Freightliner eCascadia	BEV	Battery (438 kWh)	118 kWh/100km
Volvo VNR Electric	BEV	Battery (565 kWh)	128 kWh/100km
Mercedes-Benz eActros	BEV	Battery (621 kWh)	111 kWh/100km
MAN eTGX	BEV	Battery (480 kWh)	96 kWh/100km
Lion8	BEV	Battery (630 kWh)	142 kWh/100km
Tesla Semi	BEV	Battery (900 kWh)	125 kWh/100km

*HEV: hybrid electric vehicle; PHEV: plug-in hybrid electric vehicle; BEV: battery electric vehicle.

Research on electrified truck platooning remains simulation-based in literature. Pramanik and Anwar [41] developed a predictive energy management framework for hybrid-electric truck platooning using an agent-based multi-objective optimization approach. The simulation results reported that a three-truck platoon achieved an

average of 10% fuel savings and 13% reduction in engine-out NO_x emissions compared with a non-platoon baseline. These benefits arise because hybrid electric trucks, equipped with dual power sources (i.e., the internal combustion engine (ICE) and electric machine), support bi-directional power flow: the electric motor provides traction assistance to reduce engine load during propulsion and also recovers kinetic energy via regenerative braking during deceleration. Consequently, hybridization not only shifts engine operating points toward more efficient regions, but also increases operational flexibility such as engine-off coasting when traction demand is low while maintaining drivability through electric torque support. Furthermore, Hu and Bauer [81] investigated battery-electric truck platooning and argued that electric truck platoons can realize larger efficiency gains than ICE platoons because electric drivetrains exhibit a broad high-efficiency operating plateau, so reductions in aerodynamic drag translate more directly into net energy savings. Their simulations at constant gap of 10 m showed that electric platooning provided approximately 20% more energy savings compared with the conventional ICE truck platooning. Moreover, recent simulation studies indicated that mixed-powertrain platoons (e.g., combinations of electric and diesel trucks) may erode the overall platooning gains relative to homogeneous platoons [82]. In particular, the different powertrain types can influence the followers' energy outcome through the impact on speed transients and the resulting traction/braking demands. For example, computational tests showed that the same battery-electric follower consumed 4.8% less energy when following a BEV leader than when following a conventional leader [83].

1.3 Platooning Control Scheme

1.3.1 Platooning Control Methods

Control methods are fundamental to truck platooning because they enable operational safety and coordinated motion while unlocking potential benefits such as fuel

savings and emissions reduction. In general, platooning control system can be divided into longitudinal control and lateral control. The longitudinal control aims to maintain a coordinated movement and optimal spacing between vehicles. It involves the management of speed, acceleration, and braking of the vehicles within a platoon. The objective of lateral control is to control the lateral motion of the platoon in a case of transitional maneuvers such as lane changes, platoon formation (joining), and platoon dissolution (splitting). Most practical platooning architectures adopt a hierarchical structure consisting of a strategy layer and a control layer. The strategy layer determines high-level motion commands (e.g., desired acceleration and steering angle) based on objectives and constraints, whereas the control layer tracks these commands by actuating steering, braking, and throttle to execute platooning maneuvers accurately.

A broad range of platooning control techniques has been proposed in the literature. Memon et al. [84] developed a proportional–integral–derivative (PID) feedback controller and evaluated its performance under sudden acceleration and braking. Their results indicated that the PID controller could be stable once steady-state behavior is achieved, but was inadequate for scenarios with strong transients and nonlinearities. Peng et al. [85] presented a platoon control strategy based on sliding-mode control (SMC) theory and reported improved platoon formation time and stability. Li et al. [86] proposed a distributed H_∞ control method for heterogeneous vehicular platoons to balance robust stability, disturbance attenuation, and string stability. Hu et al. [87] presented a distributed controller to optimize platoon fuel consumption by combining switching feedback control with economic model predictive control; their results showed that string stability can be ensured while achieving up to 6.8% fuel savings. In [88], an optimal fuzzy adaptive robust control scheme was proposed to address uncertainties that are nonlinear and potentially fast time-varying. Zhu et al. [89] developed a composite controller consisting of a linear quadratic regulator (LQR) component and a SMC component to guarantee platoon consensus under external

disturbances. Data-driven methods have also been explored, often in combination with model-based designs to meet safety and performance requirements. For example, Guo et al. [90] proposed a neural-network-based distributed adaptive approach combined with SMC for vehicle-following platoons in the presence of input saturation, unknown unmodeled nonlinear dynamics, and external disturbances. Table 1.4 summarizes representative platooning control methods reviewed in the literature.

Overall, PID-based controllers are attractive due to simplicity and ease of implementation; however, they are often insufficient when strong disturbances, nonlinear dynamics, and explicit constraints must be handled. Learning-based controllers can provide strong approximation capability for complex nonlinearities, but their performance depends critically on data coverage, validation, and robustness provisions, particularly for safety-critical deployment. Model predictive control (MPC) has been widely adopted in the platooning field because it naturally accommodates multi-input multi-output dynamics, explicitly handles state and actuator constraints, and supports systematic trade-offs among safety, energy, and emissions objectives [121].

1.3.2 Platooning Control Architecture

Hierarchical control architectures are widely used in cooperative vehicle platooning because they decompose the overall task into layers with clear responsibilities, improving implementability, scalability, and safety. A representative example is the Intelligent Vehicle/Highway System (IVHS) architecture proposed by Varaiya [122], which consists of five layers: network, link, coordination, regulation, and physical, as shown in Figure 1.5. The network layer and the link layer are referred to as the road-side system for vehicle cooperation. The former mainly assigns appropriate routes to each vehicle, while the latter primarily regulates the maximum platoon size allowed to enter the road system and the optimal speed according to traffic conditions. The coordination, regulation, and physical layers constitute the on-board vehicle system. The coordination layer receives commands from the link layer and enables cooperation

Table 1.4: A summary of different control methods for vehicle platoon systems.

Control Method	Advantage	Limitation	Vehicle Model
PID [84, 91–94]	• It is efficient and robust against some common uncertainties such as plant uncertainties • Easy to implement	• The performance of PID controller reduces in the presents of non-linearities and large disturbance	Model free
SMC [85, 95–100]	• Fast response under system uncertainties and external disturbances • Suitable for nonlinear systems • Provides constant and small displacement errors when the speed of the plant increases	• Requires prior knowledge of uncertainty and disturbance bounds • Control law is sensitive to path curvature variations • Controller sampling rate effects on the performance of the controller	Kinematic or dynamic vehicle model required
H_∞ [86, 101–105]	• Achieves stabilization with guaranteed performance	• A high level of mathematical understanding is needed and it is only applicable for linear systems	Accurate kinematic or dynamic vehicle model required
MPC [42, 87, 106–111]	• Consider the whole vehicle model in the control problem • Inclusion of system constraints in the design procedure • Robust against plant parameter variations	• Computational requirements of online optimization making this controller unsuitable for high-speed • Larger tuning parameter set compared to PID controller • The performance is sensitive to the accuracy of the prediction model	Accurate kinematic or dynamic vehicle model required
Fuzzy logic [88, 112–115]	• Provides smooth response • Robustness to sudden changes in the environment	• Stability is not guaranteed • Formal stability analysis needed to achieve a systematic controller tuning • A major number of variables can lead to uncontrollable rules	Model free
LQR control [89, 116, 117]	• Robust against measurement noise • Offline gain optimization results in a simple online controller	• It is not easy to design the controller in the presents of high uncertainties	Accurate kinematic or dynamic vehicle model required
Learning-based control [90, 118–120]	• More accurate results can be obtained through sufficient training • Low computational cost after the model is trained	• A large amount of real-world data is necessary to train the controller • Not scalable	Model free

among vehicles within the platoon through information and communication protocols to ensure driving safety. The regulation layer is responsible for lane keeping, speed control, and following maneuvers. The physical layer concerns the vehicle’s actual motion; the controller at this layer actuates the steering wheel, throttle, and braking system. It also includes corresponding sensors to perceive vehicle speed, acceleration,

and engine-state information, and transmits these data to the regulation layer.

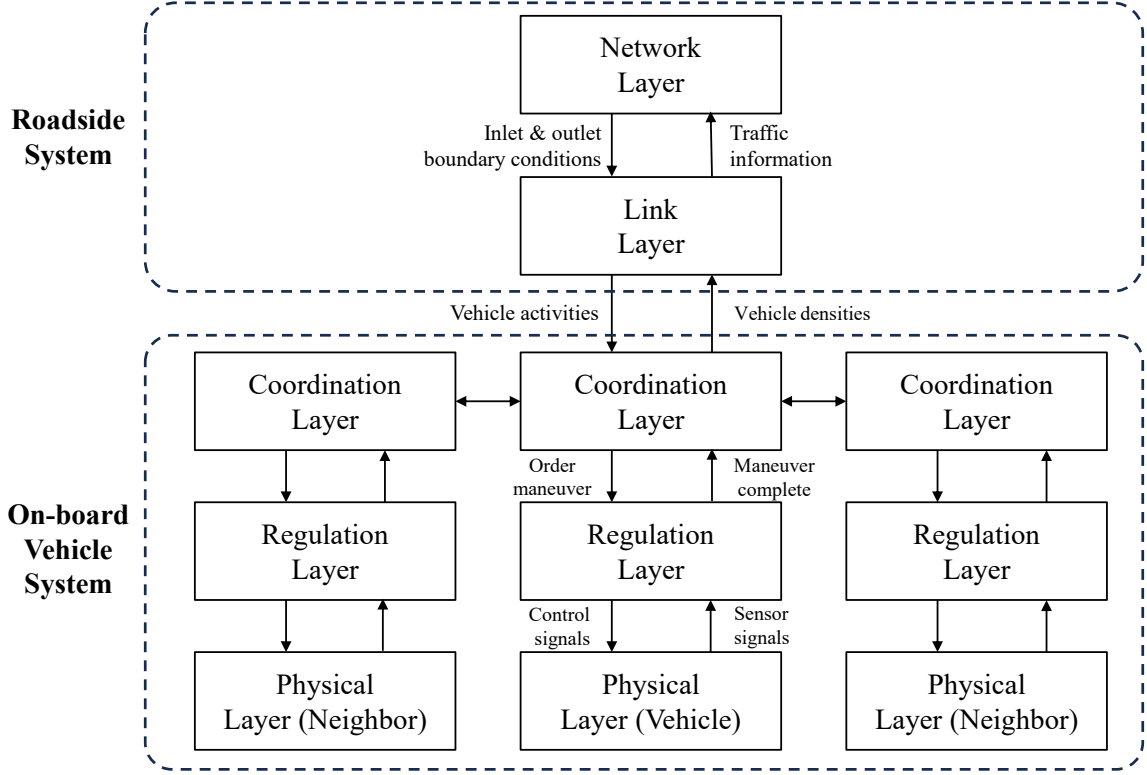


Figure 1.5: The five-layer intelligent vehicle/highway system control architecture.

Tsugawa et al. [123] proposed a cooperative driving system control framework with a similar separation of responsibilities, organized into three layers: traffic control, vehicle management, and vehicle control, as shown in Figure 1.6. This framework organizes vehicles into platoons for management and uses sensor systems and V2V communication technologies to enable smooth platoon merging and lane changes. The traffic control layer is part of the roadside infrastructure and applies to all vehicles. This layer comprises multiple distributed controllers; each controller independently determines the commands to be sent to neighboring vehicles and forwards them to the vehicle management layer. The vehicle management layer, located in each vehicle, receives motion commands from the traffic control layer via vehicle-to-infrastructure (V2I) communication, and also obtains neighboring vehicles' driving information through V2V communication while receiving sensor data from the vehicle

control layer. The vehicle control layer primarily senses the state ahead of the vehicle and roadway conditions, and sends this information to the vehicle management layer; meanwhile, it receives commands from the upper layer and controls the vehicle actuators accordingly.

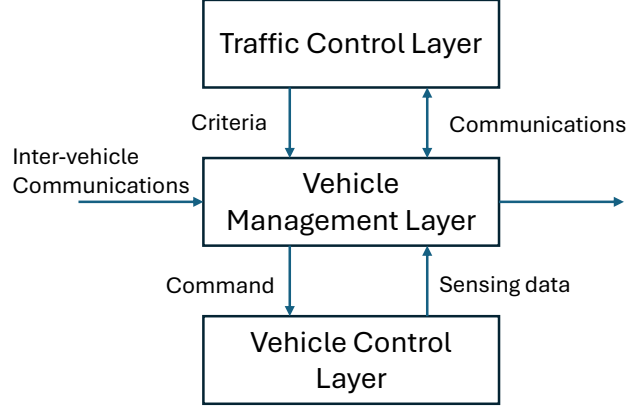


Figure 1.6: The three-layer cooperative driving system architecture.

Building on these concepts, Hallé and Chaib-draa [124] proposed the a collaborative driving system (CDS) framework, consisting of a traffic control layer, a management layer, and a guidance layer. The CDS framework treats the platoon as an autonomous agent supported by cooperative adaptive cruise control and explicit coordination logic for join/leave events, with an emphasis on maintaining intra-platoon stability during transitional maneuvers.

The hierarchical control architecture can reduce the system’s dependence on a centralized control unit, and it is widely adopted to the platooning control system for safety enhancement [125–127], energy management [128–130], and emissions mitigation [42, 131, 132].

1.4 Problem Identification and Proposed Solutions

The trucking sector remains predominantly reliant on ICE powertrains, and this reliance is expected to persist for the foreseeable future, particularly for heavy-duty

applications that continue to be dominated by diesel engines. Consequently, the trucks consume large quantities of diesel fuel and emit harmful pollutants, including NO_x , which pose significant risks to air quality and public health. Against this backdrop, truck platooning has been proposed as a complementary approach to improve energy efficiency and reduce emissions on top of existing vehicle-level technologies and operational measures. Building on the literature reviewed in the preceding sections, Figure 1.7 summarizes the main strategies investigated to date for energy savings and tailpipe NO_x reduction in heavy-duty truck platooning, and it also positions the new approaches developed in this thesis within this landscape.

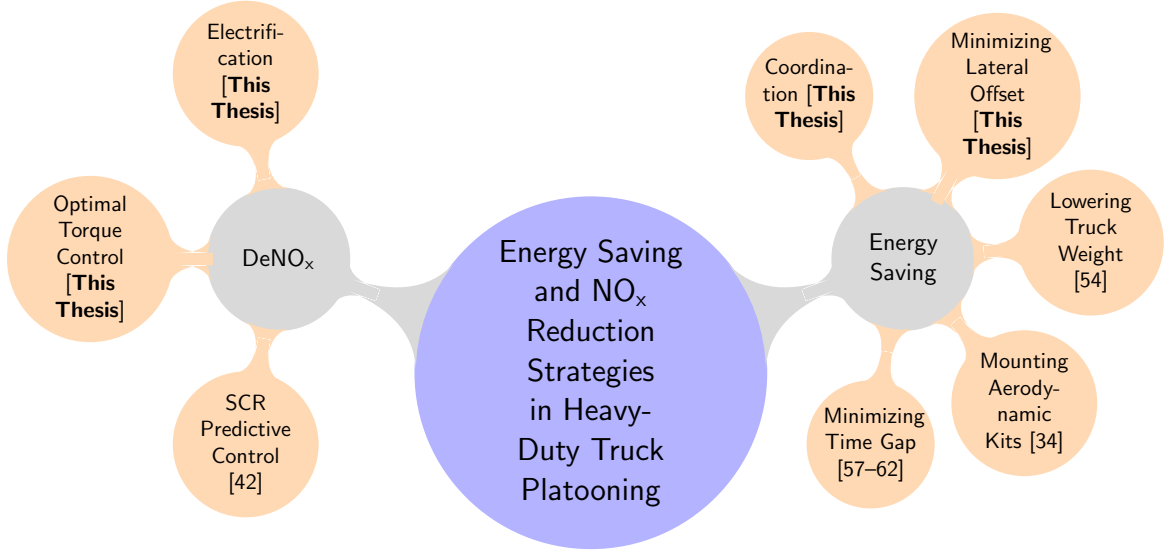


Figure 1.7: Strategies for reducing energy consumption and tailpipe NO_x emissions in heavy-duty truck platooning.

Prior studies have examined the energy and emissions impacts of platooning across a range of experimental platforms, and have employed diverse control methodologies to realize potential benefits. However, the review also indicates that several critical gaps remain and must be addressed to enable robust, regulation-relevant benefits and to accelerate deployment in real-world commercial operations. The research gaps motivating this thesis are grouped into three themes:

- **Regulation-compliant platoon coordination:** Design a platoon-level con-

trol system that enables operation at regulation-compliant time gaps (i.e., ≥ 2 sec) while delivering measurable fuel savings and tailpipe NO_x reduction through coordinated motion planning and powertrain management.

- **Integrated longitudinal–lateral control:** Develop integrated control strategies that minimize lateral offsets and maintain aerodynamic alignment, thereby preserving expected platooning benefits under real-world driving conditions.
- **Electrified platooning assessment and optimization:** Conduct a comprehensive assessment of electrification levels (e.g., conventional, hybrid, and electric configurations) and quantify their impacts on energy optimization and emissions reduction for platooning operation.

1.4.1 Regulation-Compliant Platoon Coordination

Most prior truck platooning studies have emphasized close-spacing operation—typically at time gaps below 2 sec—to maximize aerodynamic benefits and, in some cases, to improve emissions outcomes. However, commercial operation at such small time gaps is often restricted or prohibited due to safety regulations [133–138]. For example, a minimum 2 sec time gap is required in Europe and Western Australia, whereas other Australian jurisdictions (e.g., New South Wales and Victoria) require at least 3 sec. In addition, certain jurisdictions in Canada and the United States impose minimum time gaps as large as 4 sec. Consequently, many close-spacing platooning configurations reported in the literature are not directly deployable on public roads because they do not satisfy local regulatory constraints. Moreover, there remains a limited body of research on how to optimize large-spacing platoons for both fuel savings and tailpipe NO_x reduction under realistic operating conditions, particularly on hilly roads and in the presence of wind disturbances.

To address these gaps, this thesis proposes the following solutions:

- 1) Conduct a first-of-its-kind cooperative truck platooning trial on Canadian public roads under permitted time gaps to verify the feasibility and safety of platooning

under commercial operations. The resulting dataset is used to identify practical limitations and to inform design requirements for adapting cooperative platooning technology to public-road freight transportation.

2) Develop a platooning system model that captures varying road gradients and aerodynamic interactions between trucks under windy conditions. Collected real-world trucking data are used to validate the proposed model and ensure fidelity under regulation-relevant operating scenarios.

3) Design an innovative hierarchical control architecture at the platoon-coordination level that achieves fuel savings by smoothing speed profiles and reduces tailpipe NO_x by optimizing engine torque response, while maintaining safety under regulation-compliant time gaps (e.g., ≥ 2 sec).

1.4.2 Integrated Longitudinal and Lateral Control

Prior studies span a broad range of automation configurations, with many efforts focusing primarily on longitudinal control to realize platooning benefits. Other investigations incorporate lateral control mainly to ensure that vehicles remain safely within a lane, particularly in dedicated-lane scenarios. In real-world operations, however, steering variability and crosswinds frequently induce lateral offsets that misalign platooning vehicles. Such misalignment can shift a follower partially out of the leader’s wake, attenuate aerodynamic drag reduction, and consequently reduce achievable fuel savings. These observations motivate the inclusion of an explicit lateral-alignment objective—beyond conventional lane keeping—to preserve aerodynamic benefits and maintain robust performance under disturbances. Despite its practical importance, the role of lateral alignment in platooning performance remains under-addressed in the existing literature.

To address these gaps, this thesis proposes a coordinated control mechanism that integrates longitudinal and lateral control modules. The longitudinal module maintains a prescribed inter-vehicle spacing while compensating for wind-induced distur-

bances, thereby ensuring safety and helping preserve aerodynamic benefits. It further leverages V2V connectivity to enhance coordination and avoid aggressive engine transients, which can contribute to lower tailpipe NO_x emissions. In parallel, the lateral control module regulates the lateral motion of trucks to ensure lane keeping while explicitly minimizing lateral offsets that degrade fuel savings under crosswind conditions. Through the shared longitudinal velocity coupling, these two control modules interact dynamically.

1.4.3 Electrified Platooning Assessment and Optimization

Despite encouraging simulation evidence of electrified truck platooning in the literature, several research gaps remain. First, there is no single control framework in literature to encompass trucks ranging from conventional diesel to hybrid electric and battery electric vehicles. Second, most prior studies ignore tailpipe NO_x emissions for the analysis of truck platooning. Third, most prior studies do not consider real-world drive cycle, including road grade, wind variations. Fourth, the truck platooning models used in most prior studies are not experimentally validated. Finally, although electrified platooning has been discussed for different powertrains, no prior studies have systematically investigated and compared the performance of conventional, hybrid, and battery-electric configurations for homogeneous truck platoons.

To address the identified research gaps, this thesis proposes a two-level hierarchical comprehensive control architecture for optimizing energy consumption and minimizing tailpipe NO_x emissions in electrified truck platooning. The upper-level controller employs model predictive control (MPC) to ensure safe platooning and optimize the speed trajectories of electrified trucks within the platoon. The lower-level controller applies MPC-based powertrain energy management to determine optimal torque allocation with the objective of minimizing energy consumption while reducing tailpipe NO_x emissions. In addition, this thesis conducts a comprehensive assessment of electrification levels (including conventional, hybrid and electric configurations) and quan-

tifies their impacts on energy optimization and emissions performance in platooning, thereby providing a consistent basis for comparing achievable benefits across power-train technologies.

1.5 Thesis Outline and Contributions

1.5.1 Thesis Outline

This thesis is organized into five parts spanning eight chapters. The overall structure and the relationships among the parts are schematically illustrated in Figure 1.8.

The main parts and chapters of this thesis are summarized as follows:

- **Part I: Introduction and Literature Review**

- **Chapter 1** introduces the research context and motivation, reviews the relevant literature, identifies key research gaps, proposes corresponding solutions, and summarizes main contributions of the thesis.

- **Part II: On-Road Cooperative Truck Platooning Trials**

- **Chapter 2** presents the experimental setup and test conditions for on-road cooperative truck platooning trials, and analyzes the collected dataset to assess platooning performance, including safety, energy, and emissions outcomes.

- **Part III: Truck Platoon Modelling and Validation**

- **Chapter 3** develops longitudinal and lateral truck-platoon system models and validates them using real-world data from the on-road cooperative truck platooning trials.

- **Part IV: Truck Platooning Control System Design**

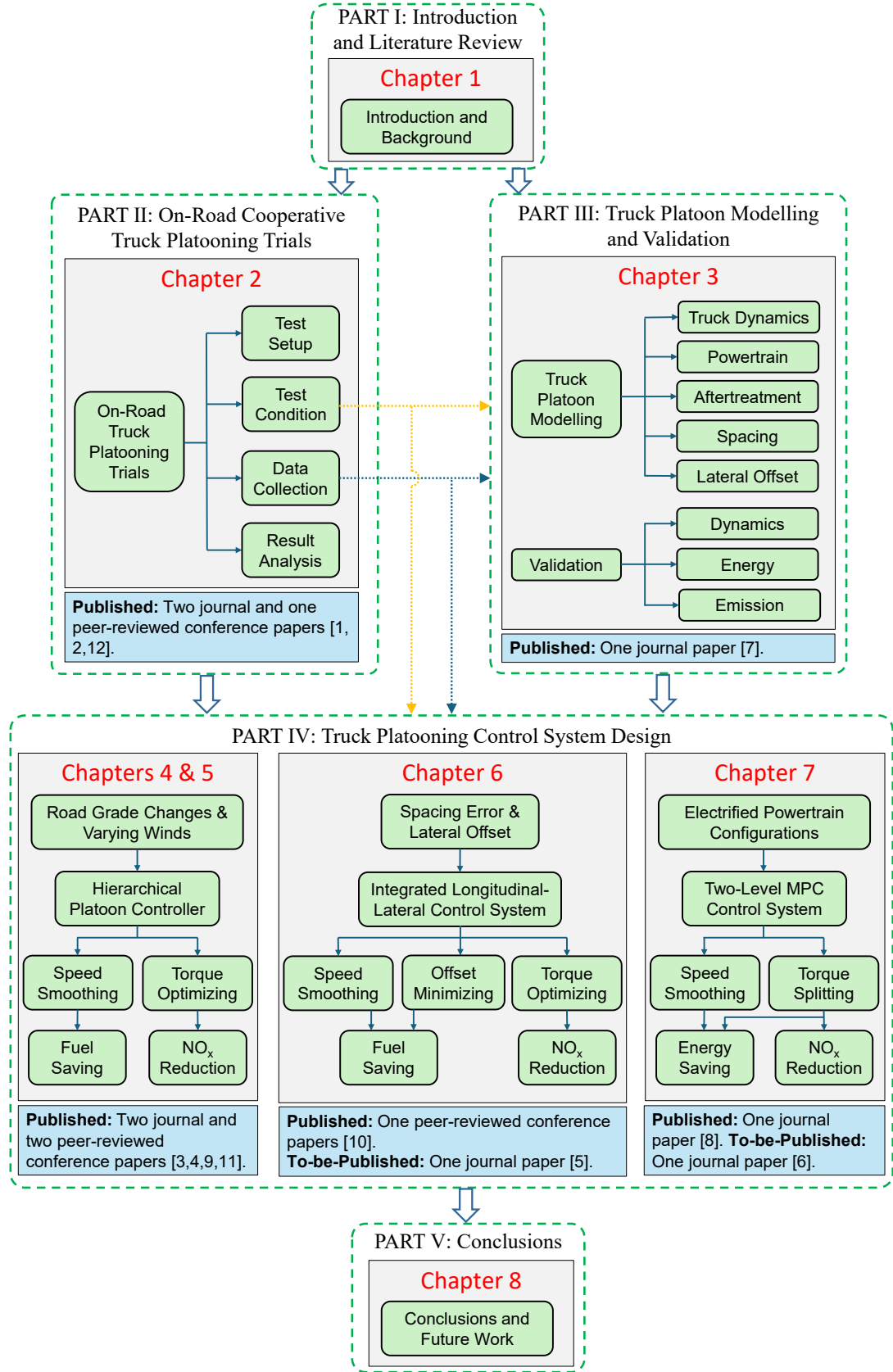


Figure 1.8: Schematic of the thesis organization.

- **Chapter 4** designs a hierarchical platoon controller that accounts for road-grade variations to optimize truck speed and engine torque. The controller targets fuel savings and tailpipe NO_x reduction while maintaining regulation-compliant large spacing to meet safety requirements.
- **Chapter 5** extends coordination to windy environments by explicitly modelling wind-affected aerodynamic interactions and incorporating wind information to improve robustness in large-spacing platoons.
- **Chapter 6** introduces an integrated longitudinal–lateral framework that couples spacing regulation and lateral alignment to mitigate fuel-loss mechanisms caused by offset and crosswind disturbances, while limiting torque transients relevant to NO_x.
- **Chapter 7** develops a hierarchical two-level control framework for electrified platoons and evaluates how electrification level alters achievable energy and emission outcomes.

- **Part V: Conclusions**

- **Chapter 8** summarizes the main conclusions of the thesis and outlines directions for future research.

1.5.2 Contributions

A summary of the main contributions of this thesis are as follows (publications resulting from each chapter are noted; a full list of publications is provided in Appendix A):

- **Chapter 2: On-Road Cooperative Truck Platooning Trials**

- Conducted cooperative truck platooning trials on Canadian public roads under winter operating conditions and demonstrated the feasibility of commercial operations with regulation-compliant time gaps [1].

- Collected an extensive real-world dataset covering a broad range of operating and environmental conditions relevant to commercial freight operations.
- Quantitatively assessed platooning performance under commercial operation, including longitudinal spacing, lateral offset, fuel consumption, and NO_x emissions [2, 12].

- **Chapter 3: Truck Platoon Modelling and Validation**

- Developed longitudinal and lateral truck-platoon system models that capture aerodynamic interactions induced by both longitudinal spacing and lateral offset [3, 4].
- Established the linkage between engine-torque transient features and tailpipe NO_x emissions, and incorporated these features as inputs to a long short-term memory (LSTM) model to improve prediction accuracy [7].
- Validated the developed models using real-world data from the on-road cooperative truck platooning trials, demonstrating high fidelity under commercial operation on hilly terrain with varying wind conditions.

- **Chapter 4: Truck Platooning under Varying Road Grades**

- Designed an innovative truck platooning controller that improves inter-vehicle coordination under road-grade variations by smoothing speed profiles [11].
- Integrated powertrain dynamics into the platoon controller to attenuate engine torque oscillations and thereby lower tailpipe NO_x emissions [3].

- **Chapter 5: Truck Platooning in Windy Environments**

- Proposed a coordination strategy that accounts for aerodynamic interactions under windy conditions to improve large-spacing platoon coordination and achieve speed smoothing [9].
- Incorporated real-time wind information into the control framework to enhance operational safety while maintaining fuel savings and reducing tailpipe NO_x emissions [4].

- **Chapter 6: Integrated Longitudinal and Lateral Controls**

- Designed a cooperative control system that accounts for aerodynamic interactions affected jointly by longitudinal spacings and lateral offsets [10].
- Developed a coordinated longitudinal–lateral control framework in which the longitudinal controller maintains safe spacing while mitigating engine torque oscillations to reduce tailpipe NO_x emissions, and the lateral controller ensures lane keeping and minimizes lateral offsets to preserve fuel savings under crosswind conditions [5].

- **Chapter 7: Electrified Truck Platooning**

- Proposed a two-level hierarchical energy and emissions management framework for truck platoons spanning multiple powertrain electrification levels [6].
- Provided a systematic assessment of electrification levels and quantified their impacts on energy optimization and emissions performance during platooning [6].

PART II: On-Road Cooperative Truck Platooning Trials

Chapter 2

On-Road Cooperative Truck Platooning Trials¹

The first-of-its-kind cooperative truck platooning trials on Canadian public roads were conducted to investigate platooning feasibility and performance under commercial operations with regulation-compliant time gaps. This chapter begins by describing the experimental platform, including the hardware stack, and sensor suite deployed on two Class-8 tractor-trailer trucks to enable cooperative platooning. The chapter then details the test protocols and operating conditions and presents an analysis of the collected data to quantify platooning performance. Beyond reporting measured benefits, the analysis provides deployment-oriented insights by identifying practical limitations and areas where cooperative platooning technology requires further refinement for routine public-road operation. The collected real-world data also form a foundation for subsequent chapters, supporting model development and validation, controller design, and numerical evaluation under regulation-relevant conditions.

2.1 Experimental Setup

To enable the on-road truck platooning, each truck in the platoon was equipped with Cooperative Truck Platooning Systems (CTPS). As shown in Figure 2.1, the CTPS integrated V2V communication, a global positioning system (GPS) with real-time

¹ This chapter is partially based on [1, 2, 12].

kinematic (RTK) corrections, an inertial measurement unit (IMU), a radar, and a forward-facing camera. V2V communication was enabled via long-term evolution (LTE) antennas to support low-latency exchange of platooning data (e.g., position, and speed). The GPS/RTK unit provided high-accuracy vehicle positioning, including the lateral position information used for lateral alignment estimation, while the IMU measured six-degree-of-freedom motion and attitude. A front-bumper radar estimated the relative distance and velocity of surrounding vehicles, and the windshield-mounted camera detected lane markings and traffic ahead to support lane keeping. Camera-radar sensor fusion further improved relative-distance estimation between the two trucks. However, no dedicated sensor fusion was implemented between the GPS and radar data. An onboard electronic control unit (ECU) fused the sensor and V2V information and executed cooperative control algorithms to generate drive-by-wire commands for acceleration, braking, and steering. Together, these components enabled simultaneous lane centering and adaptive cruise control, consistent with SAE Level 2 automation [75].

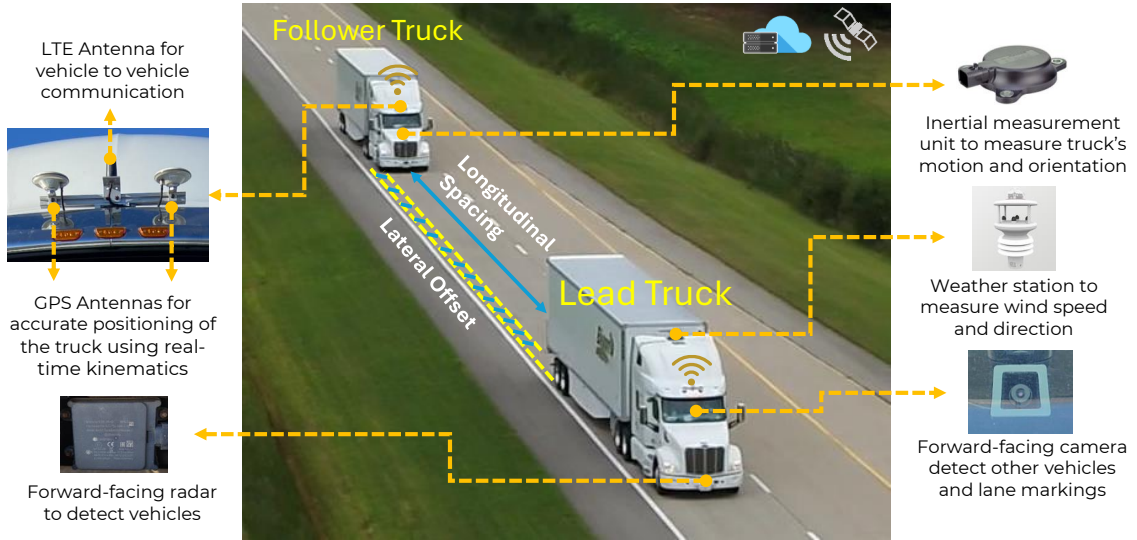


Figure 2.1: Hardware components used to enable the cooperative truck platooning trials on public roads in Canada.

Three engagement conditions must be satisfied before the two trucks can form a platoon:

- (1) Both the lead truck (also referred to as AB1 in the CTPS trial) and follower truck (also referred to as AB2 in the CTPS trial) should be engaged in Copilot mode. The Copilot mode uses the combination of lane centering control (LCC) and adaptive cruise control (ACC), which could keep the truck centered in the lane, and adjust its speed while maintaining a safe following distance from the preceding vehicle.
- (2) Both AB1 and AB2 should be in the same lane.
- (3) No other vehicles should be between AB1 and AB2.

When the above conditions are met, the systems will automatically enter into platoon mode. In practice, however, a range of real-world driving conditions could affect the platooning system’s engagement status. Table 2.1 summarizes the conditions that would affect the Copilot and CTPS platooning engagement status.

Table 2.1: Cooperative truck platooning system engagement rules.

Condition	AB2 Copilot Function	Platoon Engagement
AB1 driver brakes	✓	✓
AB1 driver accelerates	✓	✓
AB1 cancels copilot	✓	×
Apply torque by the AB1 driver to steering wheel for a certain time duration	✓	×
Apply torque by the AB2 driver to steering wheel for a certain time duration	×	×
AB2 driver brakes hard	✓	✓
AB2 driver accelerates	✓	✓
Cut-in	✓	×
Signal lost	✓	×
Bad road conditions (e.g., faded lane markings, shadows, etc.)	×	×
Inclement weather conditions (e.g., blowing snow, slush, etc.)	×	×
Bad sensor conditions (e.g., radar/camera is obstructed, etc.)	×	×
AB2 cannot see AB1 on a curvy road	✓	×

*AB1 stands for the lead truck, and AB2 stands for the follower truck.

2.1.1 Instrumentation and Data Collection

In addition to the CTPS hardware, a mobile weather station was installed on the top center of the lead truck tractor to measure the wind speed and direction (Figure 2.1). It is noted that the wind measurements provided by this station were only used for post-processing and analysis; they were not integrated into the CTPS control logic during the trials. As shown in Figure 2.2, instantaneous fuel consumption was measured using a fuel flow meter installed between the fuel tank and the engine diesel fuel supply line. This setup enabled accurate measurement of fuel consumption with a measurement uncertainty of less than 0.5%, based on a volumetric oscillating piston principle. Furthermore, NO_x emissions were measured using NO_x sensors located before and after the exhaust aftertreatment system. It should be noted that the tailpipe NO_x emissions reported here are based on NO_x sensor readings, which may include the combined effect of both NO_x and NH_3 . Therefore, the reported reductions should be interpreted as reductions in sensor-measured tailpipe emissions rather than pure NO_x alone.

Data were collected from multiple subsystems at sampling rates up to 10 Hz, including controller area network (CAN) buses, the weather station, the fuel-flow meter, and onboard perception sensors (e.g., cameras). Vehicle operating data were primarily acquired via the in-vehicle networks; both the SAE J1939 CAN and the CTPS CAN were logged on each truck to support detailed reconstruction of vehicle states, platoon operation, and emissions-relevant powertrain behavior during the trials. To enable high-frequency, time-synchronized recording across vehicles, a custom integrated central data acquisition (DAQ) system was designed and implemented. The DAQ system provided robust data logging, safeguarding, and remote live monitoring, and it was installed on both the lead and follower trucks. Data streams from the two vehicles were synchronized using a GPS time reference. In addition, the fleet management platform (Omnitracs) was used to track the real-time location of the trucks during

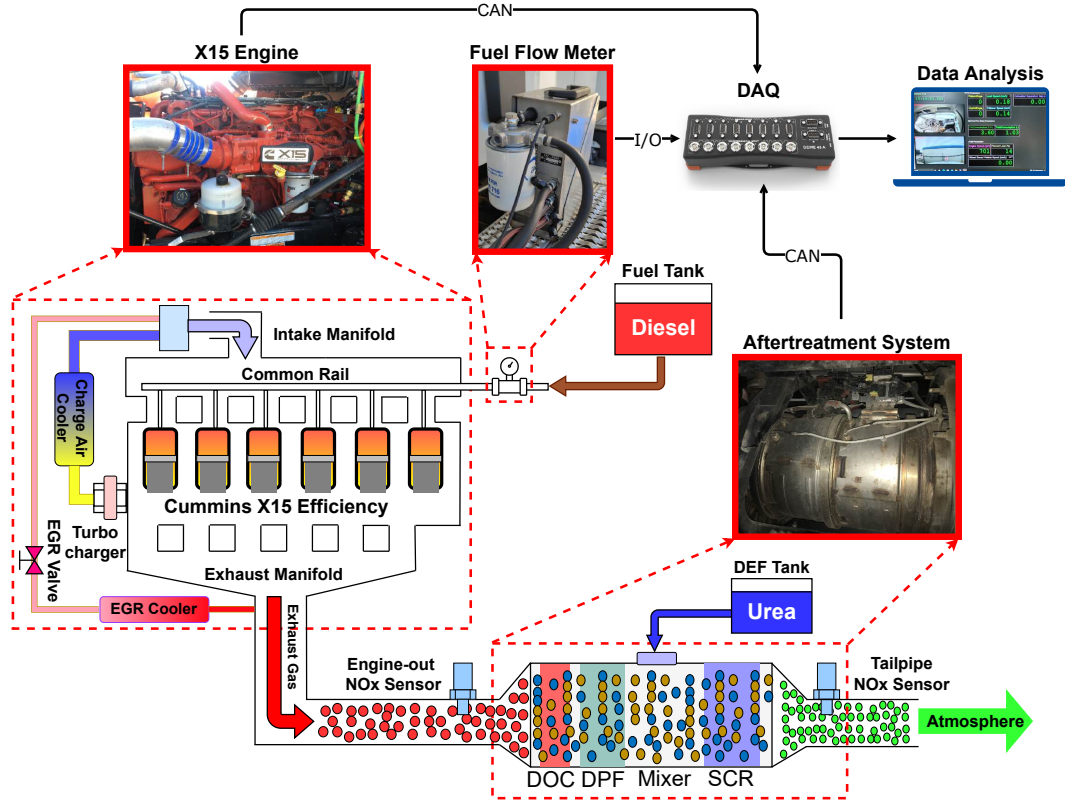


Figure 2.2: Experimental sensor configuration for fuel consumption and NO_x emission measurements. DOC, DPF and SCR stand for diesel oxidation catalyst, diesel particulate filter and selective catalytic reduction, respectively. It is noted that the diagram is schematic and does not represent the actual physical layout of the engine; it is included only for illustration of the system’s working principle.

operations. After completing each run, the logged data were transferred wirelessly to offline servers for storage and post-processing.

Overall, 339 parameters were recorded from the two trucks during each CTPS trial, spanning platoon operation metrics, powertrain and vehicle states, traffic interactions, and environmental conditions.

2.1.2 Test Route and Procedure

The trials were conducted on Alberta Highway 2, one of Canada’s busiest freight corridors connecting Edmonton and Calgary, thereby providing representative operating conditions for long-haul commercial freight transportation. The designated route is

predominantly a four-lane divided highway. As shown in Figure 2.3, the single-trip distance between the origin and destination is 323 km, and the planned platooning trip was aimed for a portion of Highway 2 between Airdrie and Leduc with a distance of 234 km.

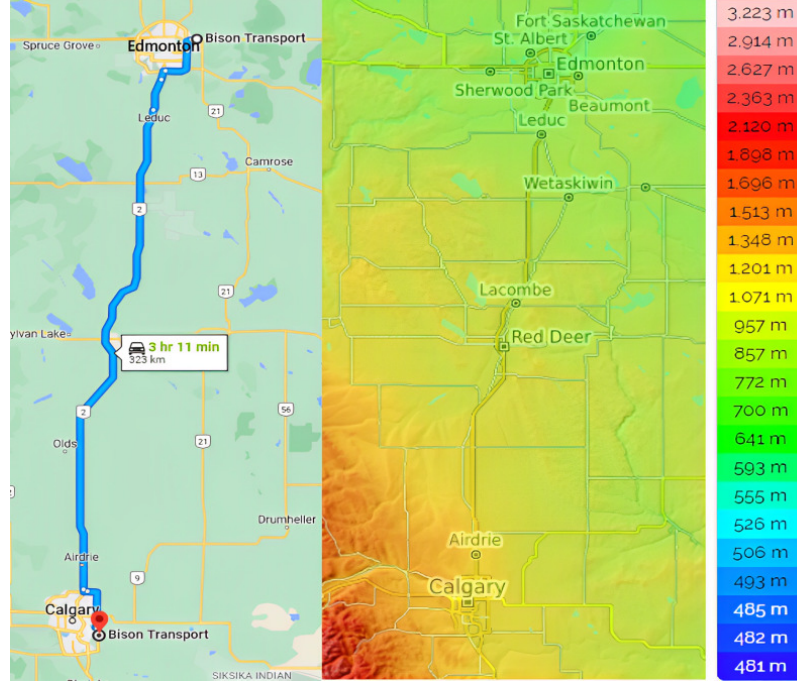


Figure 2.3: The test route in this study and the topographic map showing the elevation from sea level for the test route. The left-side navigation map was generated from Google Maps and the right-side topographic map is generated from topographic-map.

Each daily CTPS trial consisted of two highway trips: Trip 1 from Calgary to Edmonton and Trip 2 from Edmonton to Calgary, as shown in Figure 2.4. Cooperative platooning was engaged only on the Airdrie–Leduc segment. The driver on the lead truck set up the platooning speed but without exceeding the maximum speed limit of the selected route on Alberta Highway 2. Then the driver on the follower truck adjusted the following distance by selecting the permitted time gaps of 3, 4, or 5 sec. To properly assess the effect of platooning on fuel saving by the follower truck, a third truck denoted as the “control truck” is typically needed according to the SAE J1321 standar [139]. However, given the separation distances between the lead and follower trucks during platooning were substantial, the lead truck would not have

measurable fuel saving under platooning [34]. Therefore, the lead truck was used as the “control truck” and treated as the baseline, while the follower truck served as the evaluation vehicle for quantifying platooning-induced changes in fuel consumption and emissions.

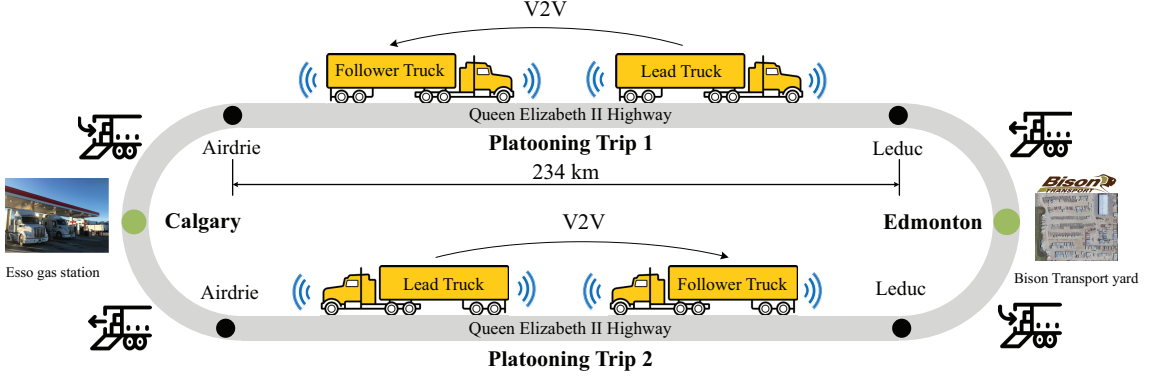


Figure 2.4: Schematic of the CTPS test procedure.

2.2 Test Condition

The platooning trials were conducted in the real-world environment, and therefore involved a number of variables. As shown in Figure 2.5, the primary controlled variables included the operating mode (platooning versus non-platooning), the commanded time gap, and the test route segment. In contrast, several influential factors were not directly controllable during public-road operation, including weather and wind conditions, traffic interactions, payload mass, and trailer configuration. Because these factors can materially affect fuel consumption and NO_x emissions, the key test conditions are summarized below to contextualize the experimental results.

Truck Weights

The maximum allowable gross vehicle weight (GVW) for a five-axle tractor-trailer combination is 39,500 kg in Alberta, Canada [140]. Across all platooning and non-platooning trips, the GVWs of the lead and follower trucks ranged from 15,963 to 39,342 kg. Figure 2.6 summarizes the truck weights during platooning trips, which

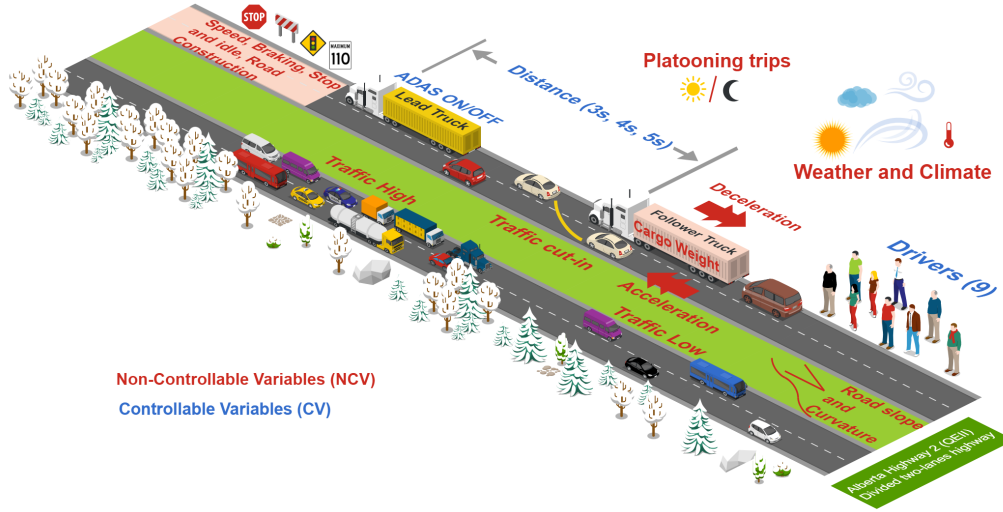


Figure 2.5: Schematic of the cooperative truck platooning trials including controllable and non-controllable variables.

span 15,963 to 38,785 kg. The lead truck was heavier than the follower truck in 86% of the platooning trips. In 31% of the platooning trips, the two trucks operated at comparable weights, defined here as within $\pm 5\%$ of each other.

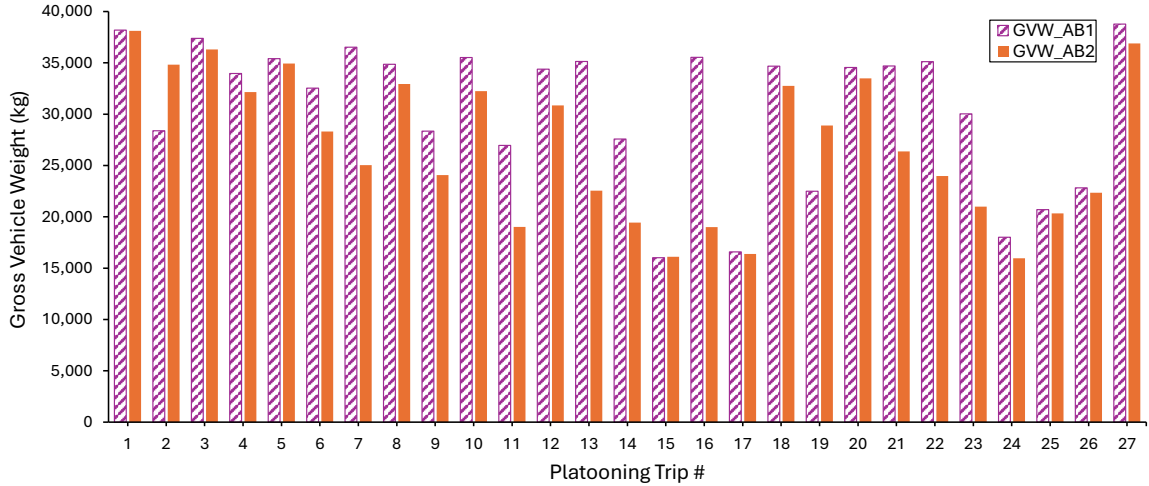


Figure 2.6: Gross vehicle weight (GVW) of trucks during platooning trials.

Road Geometries

As illustrated in Figure 2.7, the elevation difference between the trip endpoints is approximately 360 m. From Edmonton to Calgary, road grades range from -5.1% to

5.6%, and the minimum horizontal curve radius is 680 m. In the opposite direction, grades vary from -5.8% to 5.3% , with a minimum curve radius of 700 m. Round trips were performed to collect data in both directions, capturing directional differences in geometric and topographic characteristics.

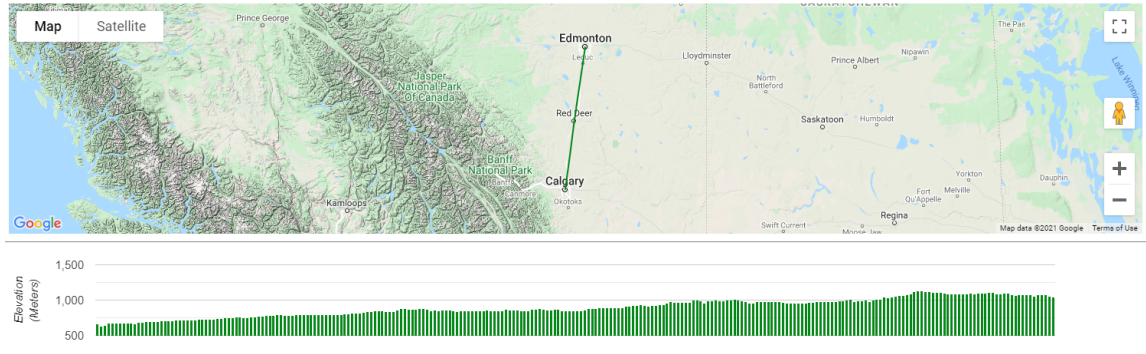


Figure 2.7: Elevation changes between Edmonton (left) and Calgary (right) obtained from Distanceto.

Apparent Wind

Wind speed measurements collected by the mobile weather station mounted on the lead truck are reported here as “apparent wind speed”, defined as the relative wind velocity experienced by a moving vehicle. As shown in Figure 2.8, apparent wind speeds during the platooning trials ranged from 33 to 153 km/h. The maximum within-trip spread in apparent wind speed was 95 km/h (Trip 21), whereas the minimum spread was 24 km/h (Trip 19). Overall, the mean apparent wind speed varied around the mean vehicle speed, consistent with the fact that apparent wind reflects the superposition of ambient wind and vehicle motion.

2.3 Test Results and Analyses

The CTPS trials were conducted on Alberta’s busiest highway corridor and comprised 41 incident-free commercial trips, covering a total distance of 22,855 km. Of these, 27 trips involved cooperative platooning tests operated with regulation-permitted time

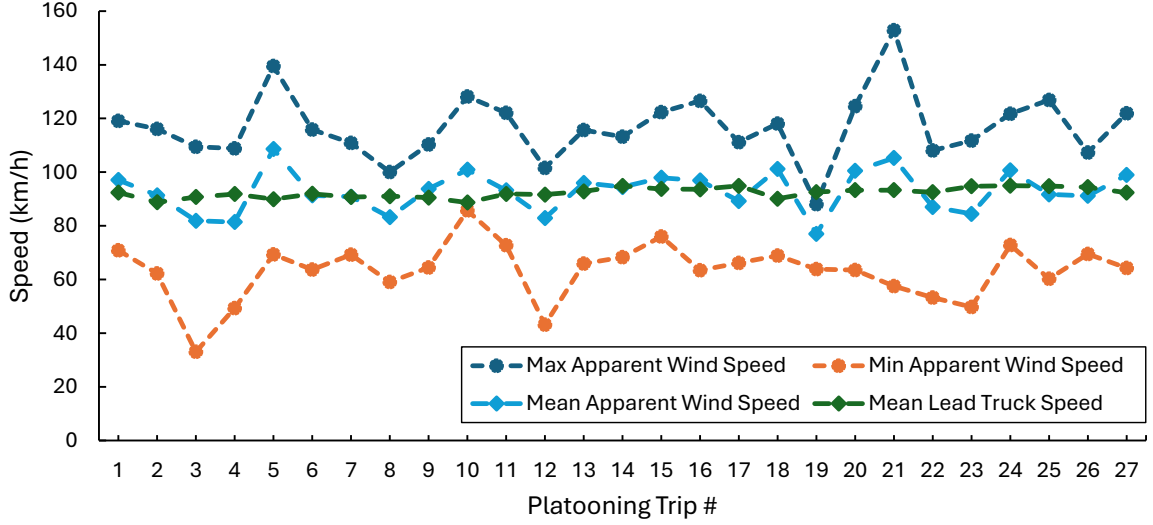


Figure 2.8: Apparent wind speeds during platooning trials.

gaps of 3, 4, and 5 s, as authorized by Alberta Transportation. Across the platooning trials, the platoon engagement ratio reached a maximum of 96% and averaged 62%, despite substantial variability in road surface conditions, including bare dry, bare wet, partially snow covered, and shoulder ice/snow.

The trials were carried out over a wide range of winter operating conditions. Ambient temperatures across all trips ranged from -27°C to 12°C , while the platooning subset was conducted between -24°C and 8°C . Notably, 78% of the platooning trips experienced minimum temperatures below 0°C . Non-platooning trips were primarily performed as baselines for comparison; therefore, this section focuses on results from platooning operation. Due to a software-related logging issue, data from two platooning trips were not recorded properly. Accordingly, the analyses presented in this chapter are based on the remaining 25 platooning trips.

2.3.1 Truck Speed

During the platooning segment of the route, the lead-truck driver set the cruising speed which complied with the posted speed limit, and the follower-truck driver selected one of the permitted time gaps to regulate inter-vehicle spacing. As a result, the

realized speed trajectories of the two vehicles directly influence CTPS performance. Figure 2.9 summarizes the observed platooning speed statistics for both trucks. The lead truck's mean speed ranged from 89 to 95 km/h. Across trips, its maximum speed under platooning varied from 100 to 116 km/h, while the minimum speed ranged from 57 to 84 km/h; the lower speed values were primarily attributable to traffic interactions along the highway. The follower truck exhibited comparable mean speeds, ranging from 89 to 95 km/h. Its maximum speed spanned 100 to 115 km/h, and its minimum speed ranged from 63 to 81 km/h.

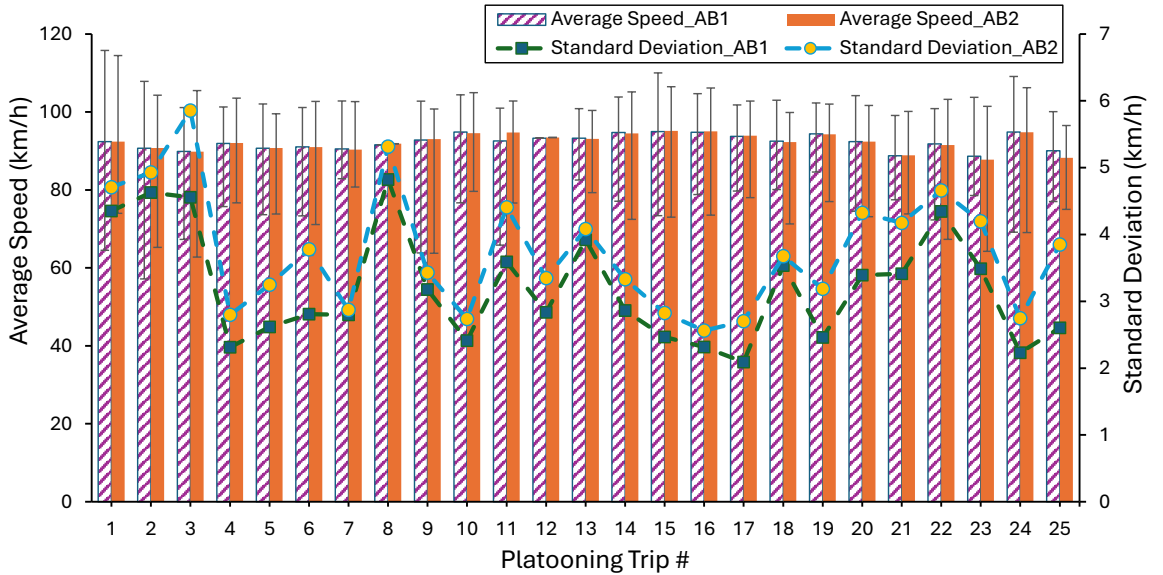


Figure 2.9: Truck speed profiles under platooning.

Figure 2.9 further indicates that the lead truck's speed standard deviation ranged from 2.1 to 4.8 km/h, with an average of 3.2 km/h. By comparison, the follower truck's speed standard deviation ranged from 2.6 to 5.9 km/h, with an average of 3.6 km/h. Overall, the two vehicles operated at similar mean speeds during platooning, while the follower truck experienced slightly larger speed fluctuations than the lead truck.

2.3.2 Fuel Consumption

Effect of Truck Weight on Fuel Consumption

Truck weight has a significant effect on fuel consumption. Two representative real-world tests were conducted to isolate the effect of truck weight under comparable operating conditions: (i) a tractor-only configuration and (ii) a tractor with a loaded trailer. The tractor-only configuration had a GVW of 9,340 kg, whereas the tractor-trailer configuration had a GVW of 35,270 kg. Both runs were performed on the same route at a nominal cruising speed of 100 km/h using cruise control.

Figure 2.10 compares the corresponding engine operating points on the brake-specific fuel consumption (BSFC) map. The heavier configuration operated more frequently in the high-efficiency region, indicating improved engine efficiency at the associated load and speed. In contrast, when vehicle mass deviated substantially from the rated operating range, the engine spent more time in lower-efficiency regions of the BSFC map. Consistent with this observation, the real-world test results show that the freight transportation specific fuel consumption² of the lighter configuration was 2.19 kg/(ton·100 km), which is approximately 3.3 times that of the heavier configuration (0.66 kg/(ton·100 km)). These results underscore the importance of accounting for vehicle load when comparing fuel consumption across trips and when interpreting platooning fuel-economy outcomes.

Fuel Consumption under Platooning

Because vehicle weights differed across platooning trips, direct comparison of absolute fuel consumption can be misleading. Accordingly, the reported fuel consumption values were normalized based on truck weights – freight transportation specific fuel consumption. Figure 2.11 shows the freight transportation specific fuel consumption of 25 platooning trips, varying from about 0.5 to 1.3 kg/(ton·100km). For the heavy

²Freight transportation specific fuel consumption is calculated by dividing fuel consumption [kg/100 km] by gross vehicle weight [ton].

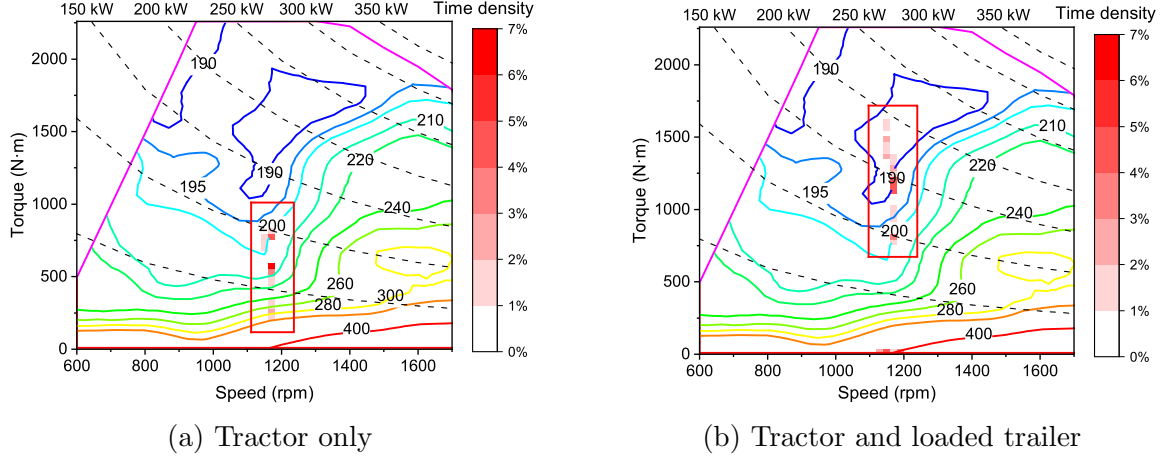


Figure 2.10: Engine BSFC map with engine operating conditions and time percentage spent at each condition.

configurations, the freight transportation specific fuel consumption got close to the value of $0.65 \text{ kg}/(\text{ton} \cdot 100\text{km})$. When there was a substantial weight difference between the lead and follower trucks, the freight transportation specific fuel consumption of the heavier truck was lower than that of the lighter one.

Therefore, trucks with a similar weight ($\pm 5\%$) are important data to help understand the effect of platooning on fuel consumption. A strong trend is not observed in Figure 2.11, since in some of the trips (marked in dotted yellow rectangle) the lead truck's freight transportation specific fuel consumption was more than the follower truck's under similar weights, while in some other trips (marked in solid green rectangle) the trend was opposite. This could be caused by the different road grades. Table 2.2 illustrates the follower truck saved fuel with a ratio of 1.6% on the flat road during platooning, but consumed up to 12.5% more fuel when encountering hilly terrain. Furthermore, the follower truck's fuel saving ratio decreased when the road sections became steeper. These results suggest that the platooning status on different types of road sections led to the different trends of trucks' overall freight transportation specific fuel consumption.

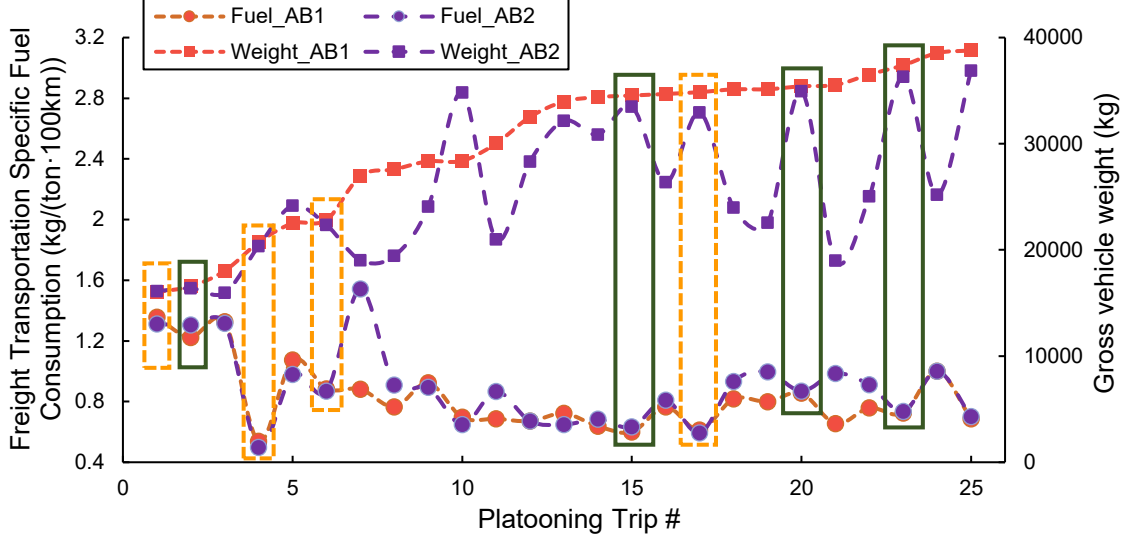


Figure 2.11: Freight transportation specific fuel consumption of lead and follower trucks during platooning.

Table 2.2: Follower truck’s fuel saving ratio under different road grades.

Road Type ¹	Road Grade (%)	Fuel Saving Ratio (%) ²
Steep downhill	-2.5 to -5.8	-7.8
Downhill	-1 to 2.5	-3.9
Flat	-1 to 1	1.6
Uphill	1 to 2.5	-4.8
Steep uphill	2.5 to 5.6	-12.5

¹ The definition of the road type is based on the road geometries of selected test route on Alberta Highway 2 in the trial.

² Fuel saving ratio with a minus sign indicates an increase in fuel consumption of follower truck.

2.3.3 NO_x Emissions

Across the platooning trips, the lead truck’s average engine-out NO_x emissions ranged from 318 to 905 g/100 km, while the follower truck’s ranged from 190 to 737 g/100 km. Figure 2.12 reports the corresponding weight-normalized engine-out NO_x emissions for each platooning trip. Light vehicle configurations (< 20 ton) exhibited elevated specific NO_x emissions, spanning 10–30 g/(ton·100 km) with a mean of

22 g/(ton·100 km). For medium-weight configurations (20–30 ton), specific NO_x emissions ranged from 12 to 34 g/(ton·100 km), with an average of 17.1 g/(ton·100 km). In contrast, heavy configurations (> 30 ton) showed lower specific NO_x emissions of 10–21 g/(ton·100 km), with the mean decreasing to 15.6 g/(ton·100 km). Overall, these results indicate that vehicle weight is a dominant factor influencing specific engine-out NO_x emissions, primarily through its effect on the distribution of engine operating points and associated combustion conditions.

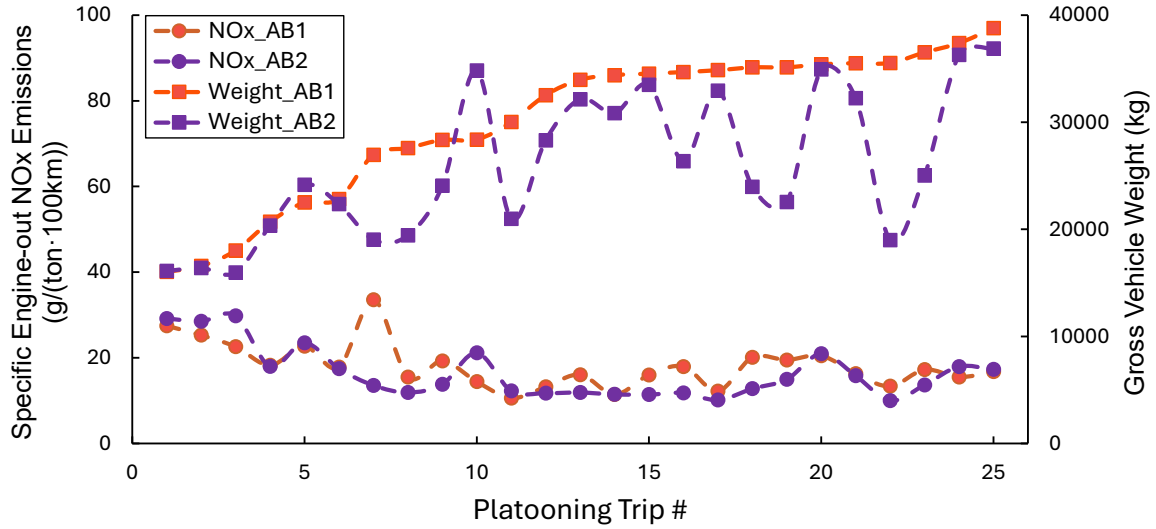


Figure 2.12: Specific engine-out NO_x emissions during platooning.

Figure 2.13 presents the corresponding specific tailpipe NO_x emissions for each platooning trip. Compared with the engine-out trends, tailpipe NO_x exhibits a slightly different pattern because it reflects the combined effects of upstream engine-out formation and downstream aftertreatment conversion efficiency, which varies with exhaust temperature, mass flow rate, catalyst state, and reductant dosing dynamics. The lead truck emitted 0.9–4.4 g/(ton·100 km) at the tailpipe, with a mean of 1.8 g/(ton·100 km). The follower truck emitted 0.8–3.4 g/(ton·100 km), with a mean of 1.6 g/(ton·100 km). Overall, the aftertreatment systems on both vehicles achieved an average NO_x conversion efficiency exceeding 90% across the trials.

Figure 2.14 shows that both the frequency and amplitude of engine-torque tran-

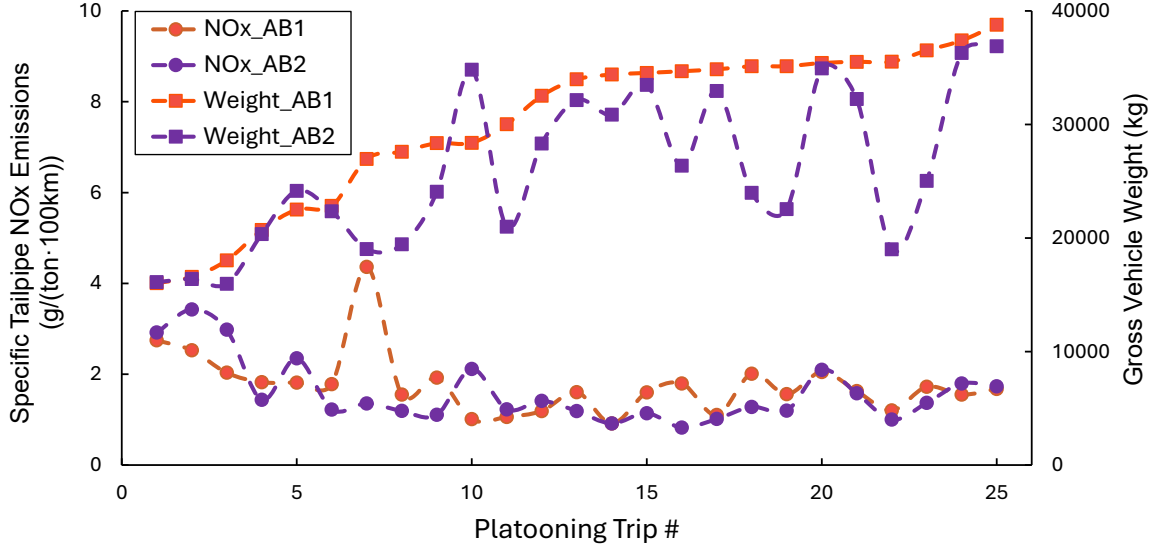
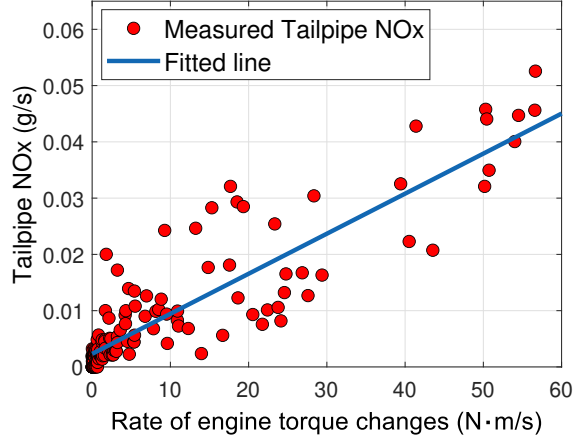


Figure 2.13: Specific tailpipe NO_x emissions during platooning.

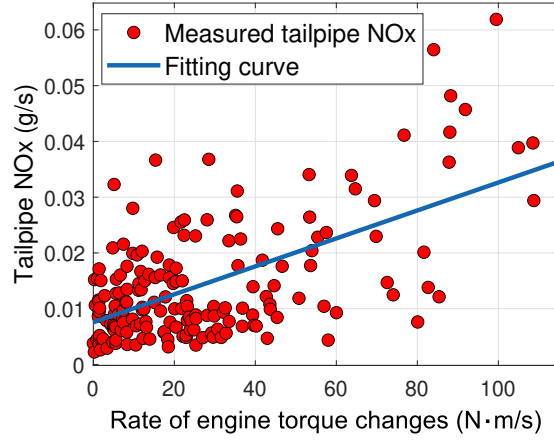
sients are strongly correlated with tailpipe NO_x emissions. The Pearson correlation coefficients between tailpipe NO_x and engine torque change rate are 0.89 and 0.85 for the low (0–790 N·m) and high (1580–2372 N·m) torque ranges, respectively. These correlation coefficients are higher than the coefficient of 0.6 observed in the medium engine torque range (790–1580 N·m). The experimental data in Figure 2.14 suggest that the tailpipe NO_x emissions are generally higher during large engine transients compared to steady-state operation when engine torque variation is not substantial (i.e., $\Delta T_e \leq 20$ N·m). The experimental data further indicate that tailpipe NO_x emissions tend to increase during pronounced torque transients compared with near steady-state operation, where torque variations remain small (e.g., $\Delta T_e \leq 20$ N·m). These observations motivate (i) incorporating torque-transient metrics as informative inputs for tailpipe NO_x modeling and (ii) reducing tailpipe NO_x through torque-command optimization that avoids unnecessary rapid changes in engine torque.

2.4 Summary of Chapter

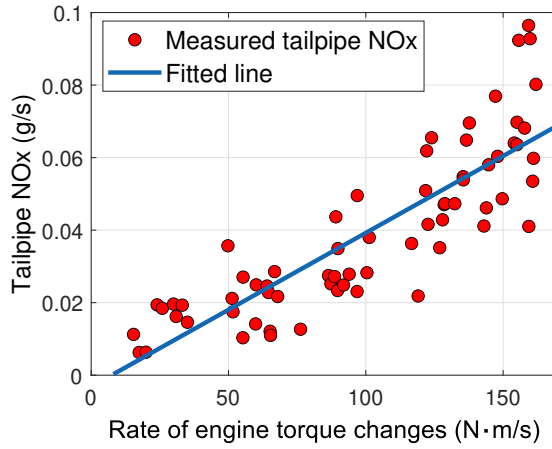
This chapter presented Canada’s first on-road cooperative truck platooning trials and the associated dataset collected from two extensively instrumented Class-8 tractor–



(a) Low engine torque range



(b) Medium engine torque range



(c) High engine torque range

Figure 2.14: Correlation between tailpipe NO_x and engine torque change rate for different engine loads: (a) $0 \leq T_e \leq 790 \text{ N}\cdot\text{m}$, (b) $790 < T_e < 1580 \text{ N}\cdot\text{m}$, and (c) $1580 \leq T_e \leq 2372 \text{ N}\cdot\text{m}$.

trailer trucks operating under commercial conditions. The campaign comprised 41 incident-free trips totaling 22,855 km between Calgary and Edmonton. The main findings are summarized as follows:

- (1) Cooperative truck platooning was demonstrated to be feasible under regulation-compliant time gaps in Canadian winter operations, including ambient temperatures down to -24°C and a range of road-surface conditions.
- (2) Although the two trucks operated at similar mean speeds during platooning, the follower truck generally exhibited larger speed fluctuations than the lead truck.
- (3) No consistent fuel-saving trend observed throughout platooning trips due to variations in trucks' specific fuel consumption on different road geometries. Fuel savings were noted on flat roads but increased fuel consumption on hilly terrain, highlighting the sensitivity of platooning benefits to road grades.
- (4) Specific engine-out NO_x emissions were higher for light vehicle configurations than for heavier configurations, consistent with differences in engine operating regimes. Tailpipe emissions reflected both engine-out formation and aftertreatment conversion behavior.

Overall, these trials provide deployment-relevant evidence for cooperative truck platooning on public roads and highlight the importance of coordination between platooning control and powertrain dynamics to achieve robust safety, fuel, and emissions outcomes at regulation-compliant time gaps. The collected extensive real-world data also form a foundation for subsequent chapters, supporting model development and validation, controller design, and numerical evaluations under regulation-relevant conditions.

PART III: Truck Platoon Modelling and Validation

Chapter 3

Truck Platoon Modelling and Validation¹

Truck platoon modelling involves developing mathematical and computational representations of inter-vehicle aerodynamic interactions, vehicle and powertrain dynamics, and emissions processes to support simulation, analysis, and controller synthesis. Model fidelity directly governs the achievable performance and robustness of platooning control systems; insufficiently accurate dynamics can degrade tracking performance, reduce stability margins, and bias estimates of energy and emissions benefits. At the same time, heavy-duty trucks exhibit pronounced nonlinearities due to powertrain behavior, aerodynamic effects, and actuator constraints. Therefore, practical control design requires an appropriate balance between model accuracy and complexity to preserve real-time implementability while retaining essential physical fidelity.

This chapter develops an integrated modelling framework for a Class-8 truck platoon that captures the dominant mechanisms governing vehicle motion, energy consumption, and tailpipe emissions. The framework includes longitudinal and lateral vehicle dynamics; a conventional diesel powertrain model comprising the engine, transmission, propeller shaft, rear differential, and drive wheels; and map-based fuel-consumption estimation using a brake-specific fuel consumption map. Tailpipe NO_x

¹ This chapter is partially based on [3, 4, 7].

emissions are modelled using deep-learning long short-term memory (LSTM) networks trained on real-world on-road measurements, comprising over 4 million data points collected across more than 10,000 km under diverse operating conditions. The resulting models are validated against independent datasets recorded during the CTPS trials to establish their accuracy and suitability for subsequent controller design and performance evaluation.

3.1 Longitudinal Platoon Modelling

The longitudinal platoon model consists of an individual truck model and an inter-vehicle coupling model that captures longitudinal interactions within the platoon. The longitudinal truck model consists of two major components: powertrain dynamics and vehicle dynamics. The longitudinal vehicle dynamics are governed by the balance between tractive/braking forces and the dominant resistive loads, including aerodynamic drag, tire rolling resistance, and gravitational forces induced by road grade. The powertrain model represents the torque path from the internal combustion engine (ICE) through the transmission, propeller shaft, differential/final drive, and drive wheels, thereby relating commanded torque to vehicle acceleration under drivetrain inertias and losses.

At the platoon level, the longitudinal coupling is defined by the spacing policy that specifies the desired inter-vehicle distance as a function of operating conditions (e.g., constant time-gap policy). This policy determines the reference spacing trajectory and error states used in the longitudinal control design and, consequently, the platoon behavior.

3.1.1 Longitudinal Vehicle Dynamics

A heavy-duty truck in motion is subjected to several external forces along its longitudinal axis, as illustrated in Figure 3.1. These forces include the tractive force at the wheels, F_t ; the braking force, F_b , which acts opposite to the direction of travel when

the service brakes are applied; aerodynamic drag, F_w ; rolling resistance, F_r ; and the gravitational (grade) force, F_g .

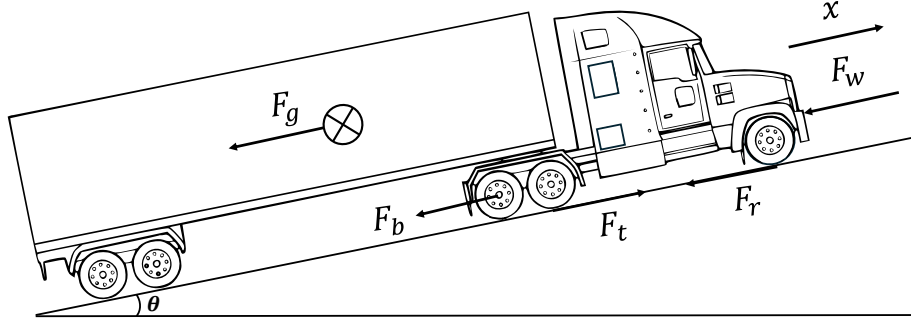


Figure 3.1: The longitudinal forces acting on a heavy-duty truck in motion.

Applying Newton's second law along the longitudinal direction yields the force balance:

$$m\ddot{x} = F_t - F_b - F_w - F_r - F_g \quad (3.1)$$

where m denotes the vehicle mass, x is the longitudinal position of the truck, and $\ddot{x}$ is the longitudinal acceleration of the truck.

The external forces in Eq. 3.1 are described in details as below.

Longitudinal Tractive Force

The longitudinal tractive force at the wheels arises from the tire-road friction force acting along the direction of travel. In the normal operating regime, where the longitudinal slip remains small, the tire force can be approximated by a linear relationship with respect to the slip ratio [141]. Accordingly, the longitudinal tire force is modeled as

$$F_t = C_\sigma \sigma_x \quad (3.2)$$

where C_σ is the longitudinal tire stiffness and σ_x is the longitudinal slip ratio. The slip ratio is defined as

$$\sigma_x = \frac{r_{\text{eff}}\omega_w - v_x}{v_x} \quad (\text{braking}) \quad (3.3)$$

$$\sigma_x = \frac{r_{\text{eff}}\omega_w - v_x}{r_{\text{eff}}\omega_w} \quad (\text{accelerating}) \quad (3.4)$$

where r_{eff} is the effective tire radius, ω_w is the wheel angular speed, v_x is the longitudinal vehicle speed.

Braking Force

In this thesis, the service brake is represented using a pressure–force relationship in which the effective braking force is assumed proportional to the pneumatic brake pressure [142]. Specifically, the braking force is modeled as

$$F_b = K_b P_b \quad (3.5)$$

where P_b denotes the brake chamber pressure, and K_b is an equivalent gain that lumps the pneumatic actuation.

Aerodynamic Drag

A vehicle traveling at speed experiences aerodynamic forces due to its relative motion with respect to the surrounding air. The component of this force opposing the direction of travel is referred to as aerodynamic drag, and it becomes increasingly dominant at highway speeds because the required propulsive power grows approximately with the cube of the relative airspeed. For heavy-duty trucks operating on highways, aerodynamic drag can account for more than 50% of the total resistive force [143, 144]. The longitudinal aerodynamic drag force is modeled as

$$F_w = \frac{1}{2} \rho C_D A (v_x + v_{wx})^2 \quad (3.6)$$

where ρ is the air density, C_D is the air drag coefficient, A is the frontal area of the truck, v_{wx} is the longitudinal wind (headwind/tailwind) speed.

Air density varies with ambient conditions, particularly temperature, and therefore directly modulates aerodynamic drag. In cold Canadian climates, winter air density can increase aerodynamic drag by nearly 20% relative to standard conditions [145], making temperature effects non-negligible in real-world efficiency assessments.

For a single vehicle, the drag coefficient C_D is commonly identified from standardized coastdown tests [146]. In platooning, however, C_D becomes a function of inter-vehicle spacing (and, when applicable, lateral alignment) because the pressure distribution around each vehicle is altered by wake interactions, particularly at the front of the follower truck.

In addition, ambient wind modifies the relative airspeed appearing in (3.6), and because drag scales with $(v + v_{wx})^2$, headwinds and tailwinds can substantially amplify or reduce the aerodynamic resistance. Accordingly, wind conditions need to be considered when modeling aerodynamic drag under real-world operations.

Rolling Resistance

Rolling resistance arises from viscoelastic losses associated with cyclic tire deformation as the wheel rolls through the contact patch. Because the road surface is typically much stiffer than the tire, the dominant deformation occurs in the tire carcass and tread. Owing to internal hysteresis and damping, a portion of the mechanical energy required to deform the tire is dissipated as heat rather than being fully recovered during unloading, producing a resistive force that opposes forward motion. A commonly used quasi-static model represents rolling resistance as proportional to the normal load:

$$F_r = \mu mg \cos \theta \tag{3.7}$$

where μ is the rolling resistance coefficient, g is the gravitational acceleration, and θ is the road angle.

The rolling resistance coefficient depends on tire construction, inflation pressure, temperature, and road surface conditions.

Gravitational Force

When a vehicle travels on a road with nonzero grade, gravity contributes a component along the longitudinal direction of motion. For heavy-duty trucks, this component

can be a dominant load because of the large gross vehicle weight (e.g., approximately 40 ton for a fully loaded Class-8 tractor-trailer). On sufficiently steep upgrades, the required tractive effort may exceed the available wheel power, leading to power-saturated operation and an inability to maintain the desired cruising speed. For example, prior work reported that a heavy-duty truck traveling uphill at 80 km/h can encounter power saturation when the grade exceeds approximately 2.9%. Conversely, on downgrades the vehicle can accelerate even with zero fueling input, and speed increases have been reported for slopes milder than about -0.9% due to gravitational assistance [147]. Therefore, the longitudinal component of gravitational force can act as either a resistive or an assisting force depending on the road grade. It is modeled as

$$F_g = mgsin\theta \quad (3.8)$$

3.1.2 Powertrain Dynamics

The Class-8 truck powertrain considered in this thesis comprises a diesel fueled internal combustion engine, 12-speed automated transmission, drive shaft, differential, final drive, and drive wheels, as depicted in Figure 3.2.

The test Class-8 Peterbilt truck tractor used in the CTPS trials was powered by a Cummins X15 Efficiency engine. This engine has six cylinders and a total displacement of 14.9 L, with a rated power of 336 kW and a peak torque of 2273 N·m. To capture the dominant actuator/air-path lag without introducing unnecessary model complexity, the engine dynamics are represented by a first-order inertial transfer function as follows [148]:

$$T_e = \frac{1}{\tau_e s + 1} T_{e,des} \quad (3.9)$$

where, T_e is the actual engine torque, τ_e is the responding time of the engine, and $T_{e,des}$ is the desired engine torque.

In addition, instantaneous fuel consumption measurements from the truck are used to construct an engine fuel-consumption map, such that the fuel mass flow rate $\dot{m}_f$

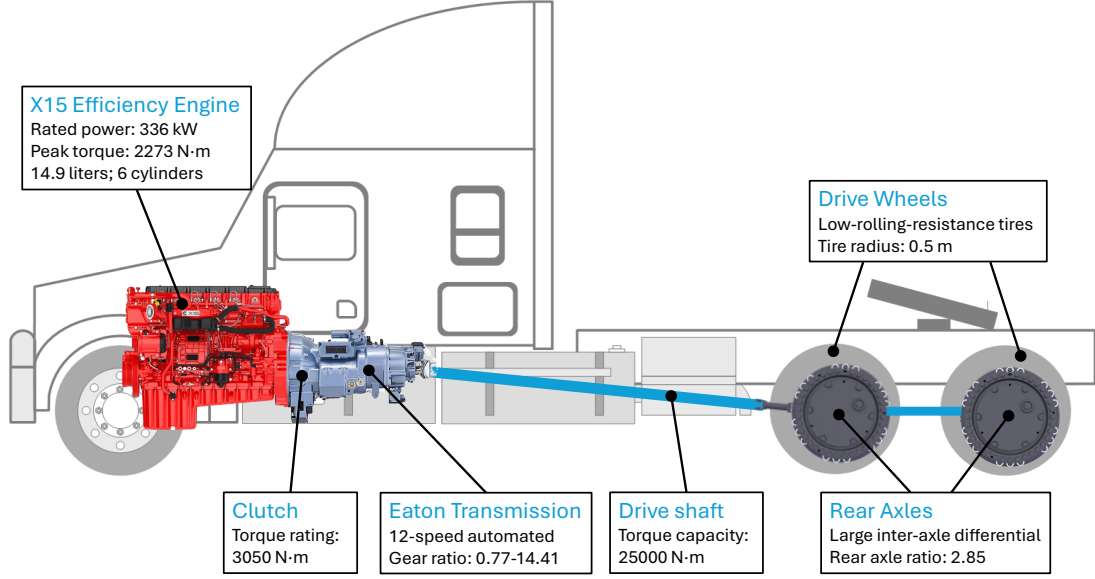


Figure 3.2: Schematic of powertrain system for the Class-8 heavy-duty truck used in the on-road truck platooning trials.

is evaluated as a function of engine torque and speed:

$$\dot{m}_f = f_e(T_e, \omega_e) \quad (3.10)$$

where ω_e is the engine angular speed, $f_e(\cdot)$ represents the engine fuel consumption map. Figure 3.3 shows the fuel consumption map generated from testing the trucks on-road for more than 22,000 km. This map provides accurate estimation of fuel consumption for each truck during platooning operations.

The drivetrain transmits mechanical power from the engine to the drive wheels through the clutch, gearbox, propeller shaft, and final drive. The drivetrain model explicitly accounts for mechanical losses through an overall drivetrain efficiency factor η_t . Specifically, η_t captures the aggregate power losses across the transmission, propeller shaft, differential, final drive, and associated bearings and gear meshes, and it is applied to map the engine torque T_e to the delivered wheel traction torque T_t as

$$T_t = T_e i_g i_0 \eta_t \quad (3.11)$$

where i_g is the gear ratio, and i_0 is final drive ratio.

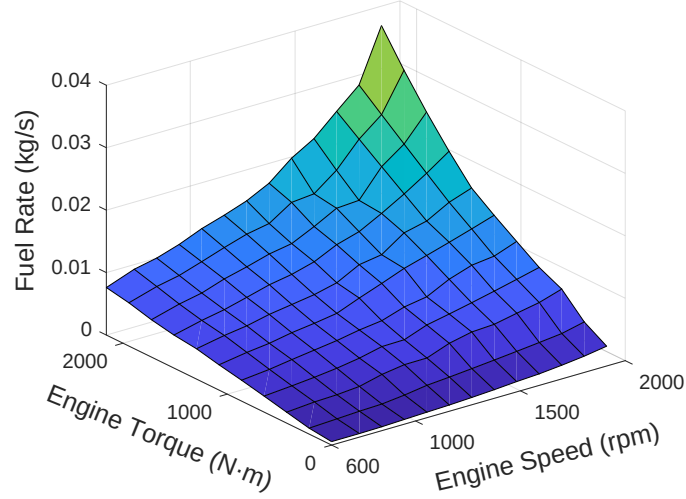


Figure 3.3: Fuel consumption map of the X15 diesel engine for the Peterbilt 579 Class-8 truck.

In addition to the torque transmission, the kinematic relationship between the engine rotational speed and the wheel speed is required to couple the powertrain dynamics to the longitudinal vehicle model. Under the assumption of a rigid driveline and negligible clutch slip during normal operation, the wheel angular speed is related to the engine speed through the total gear reduction $i_g i_0$. By converting wheel angular speed to the circumferential speed using the effective tire radius r_{eff} , the truck wheel speed v_{wh} is expressed as

$$v_{wh} = \frac{\omega_e r_{\text{eff}}}{i_g i_0} \quad (3.12)$$

Combining the driveline relations with the longitudinal force balance yields the complete longitudinal vehicle dynamics:

$$\delta_m m a = \frac{T_t - T_b}{r_{\text{eff}}} - \frac{1}{2} \rho C_D A (v_x + v_{wx})^2 - \mu m g \cos \theta - m g \sin \theta \quad (3.13)$$

where, δ_m is the rotational mass coefficient, a is the acceleration of the truck, and T_b represents the brake torque.

3.1.3 Exhaust Aftertreatment System

In response to increasingly stringent emissions regulations, modern diesel-powered heavy-duty trucks have incorporated aftertreatment systems to reduce exhaust emis-

sions. Selective catalytic reduction (SCR) is the primary component of a vehicle's aftertreatment system to reduce NO_x . The SCR system uses a urea-based diesel exhaust fluid as a reductant to facilitate the removal of NO_x through a catalytic process. The operating principle of SCR was previously illustrated in Figure 1.4 and is briefly recalled as follows. The DEF is injected into the exhaust stream before it reaches the SCR catalyst. The urea in the DEF vaporizes and decomposes to form ammonia (NH_3) in a mixer. The NH_3 is then absorbed onto the surface of the SCR catalyst, where it reacts with the NO_x emissions. Through a series of de NO_x reactions, the majority of NO_x and NH_3 are converted into harmless nitrogen and water.

Pre-calibrated look-up table is widely used to estimate tailpipe NO_x emissions. However, due to the complex transient processes within the SCR system, this empirical mapping approach often fails to accurately predict tailpipe NO_x emissions under varying operating conditions, resulting in approximately a 10% error rate [149]. To improve tailpipe NO_x estimation, this thesis develops a deep learning model based on a long short-term memory (LSTM) network. The LSTM model employs a special form of time series sequential machine learning algorithms, which are a category of recurrent neural network (RNN) algorithms that utilize gates to address the vanishing gradient issue. An LSTM network comprises three gates: an update gate G_u , a forget gate G_f , and an output gate G_o , along with a memory cell $c^{<t>}$ and an activation $a^{<t>}$ [150]. The memory cell is referred to as the long-term state, while the activation is known as the short-term state. The update gate evaluates whether the memory cell's information needs updating using a cell candidate $\tilde{c}^{<t>}$. The output gate determines $a^{<t>}$ based on $c^{<t>}$. Within an LSTM network, $c^{<t>}$ initially passes through the forget gate to discard irrelevant information. Subsequently, the update gate incorporates new information into $c^{<t>}$ using $\tilde{c}^{<t>}$. Finally, the output gate calculates $a^{<t>}$ from $c^{<t>}$ at each time step. A distinguishing feature of LSTM compared to other RNN algorithms like gated recurrent units (GRUs) is its ability to independently remove and add information to $c^{<t>}$, which enhances the algorithm's

predictive capacity for time series. However, this increased capability also leads to a more time-intensive training process due to a greater number of trainable parameters compared to other RNN algorithms. Table 3.1 presents a summary of equations for the designed LSTM network.

Table 3.1: Equations of the LSTM sequence model.

Component	Equation*
Input/update gate	$G_u^{(t)} = \sigma(W_u[a^{(t-1)}, x^{(t)}] + b_u)$
Forget gate	$G_f^{(t)} = \sigma(W_f[a^{(t-1)}, x^{(t)}] + b_f)$
Output gate	$G_o^{(t)} = \sigma(W_o[a^{(t-1)}, x^{(t)}] + b_o)$
Cell candidate	$\tilde{c}^{(t)} = \tanh(W_c[a^{(t-1)}, x^{(t)}] + b_c)$
Memory cell (long-term state)	$c^{(t)} = G_u^{(t)} \odot \tilde{c}^{(t)} + G_f^{(t)} \odot c^{(t-1)}$
Activation (short-term state)	$a^{(t)} = G_o^{(t)} \odot \tanh(c^{(t)})$

* In the equations, $W_{u,f,o,c}$ and $b_{u,f,o,c}$ are weight matrices and bias vectors, $\sigma(\cdot)$ is the sigmoid function, $\tanh(\cdot)$ is the hyperbolic tangent, $\odot$ denotes element-wise multiplication, and $[a^{(t-1)}, x^{(t)}]$ denotes concatenation of the previous hidden state and current input.

The proposed LSTM model is trained using real-world tailpipe NO_x data collected from the lead truck during the on-road cooperative truck platooning trials. Prior to training, the time delay between selected inputs and measured tailpipe NO_x is compensated to ensure that the input sequence is temporally aligned with the output measurements. Figure 3.4 illustrates the designed LSTM network architecture. Six parameters including engine torque, engine torque change rate, fuel rate, engine speed, truck speed and ambient temperature are selected as inputs. Payload effects are implicitly captured through engine torque and fuel consumption; therefore, payload is not included as a separate input to maintain model simplicity while still accurately reflecting its influence. The network architecture begins with fully connected layers, where the number of hidden neurons increases as the network deepens. This is followed by an LSTM layer, which consists of two main components: the memory cell and the gating mechanism. The memory cell summarizes historical inputs and stores

information, while the gating mechanism controls the gates for outputting state and updating memory. Subsequent to the LSTM layer, the network includes a series of fully connected layers with a decreasing number of hidden neurons for each layer. The architecture culminates with an output layer.

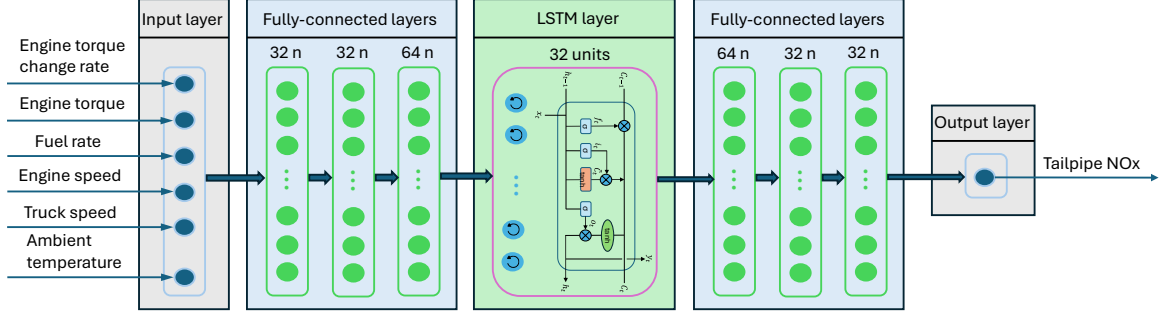


Figure 3.4: Architectures of deep learning LSTM model structure using for the prediction of tailpipe NO_x emissions.

Figure 3.5 shows that the predicted instantaneous tailpipe NO_x closely tracks experimental measurements, achieving a mean absolute percentage error (MAPE) of 2%. These results indicate that the developed LSTM model can accurately capture the transient behavior of tailpipe NO_x emissions under diverse real-world operating conditions.

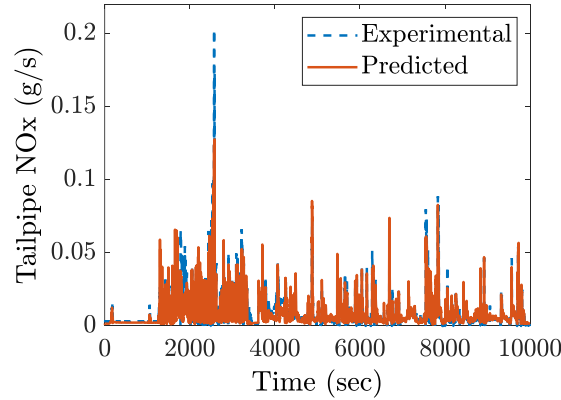


Figure 3.5: Comparison of experimental and predicted instantaneous tailpipe NO_x emissions for the developed model.

3.1.4 Spacing Policy

The spacing policy is an essential component of a vehicle platoon control system. It prescribes the desired inter-vehicle following distance during platoon operation, and the controller then regulates engine/braking torque to track this reference. Ideally, the distance between adjacent vehicles should be small and well regulated; however, excessively small spacing increases safety risk in real traffic, while overly large spacing reduces roadway utilization and can encourage cut-ins and lane changes that degrade platoon cohesion. Accordingly, spacing-policy design directly affects platoon safety, traffic interactions, and achievable operational efficiency.

Longitudinal spacing policies are commonly classified as fixed-spacing and variable-spacing strategies [151]. Under a fixed-spacing policy, the desired distance to the preceding vehicle is maintained at a constant value independent of vehicle speed. Fixed spacing is conceptually simple and can increase vehicle density, but selecting a universally suitable distance is challenging. Moreover, fixed spacing is generally more sensitive to disturbances and communication delays and may compromise platoon stability and safety under non-ideal conditions [152]. Variable-spacing policies adapt the desired distance to operating conditions; the most widely used form is time-gap-based spacing, including the constant time gap (CTG) and variable time gap (VTG) strategies. Here, the time gap is defined as the time interval between the rear end of the preceding vehicle and the front end of the following vehicle passing the same road cross-section. Compared with fixed spacing, time-gap-based strategies are better suited to real-world operation because the spacing reference scales with speed. Although VTG policies can further adapt to changing conditions, they typically require additional state information and more complex computation, which increases implementation burden [153]. Therefore, considering the intended operating environment and the need for real-time implementability, this thesis adopts the CTG strategy, which provides a favorable balance between computational simplicity and practical

robustness.

For a platoon consisting of one lead truck and n follower trucks operating under a constant time gap policy, the desired spacing d_{si} between trucks $i - 1$ and i is defined as:

$$d_{si} = d_0 + \tau_g v_i \quad (3.14)$$

where, d_0 is the standstill safe distance, τ_g is the time gap, and v_i is the longitudinal velocity of truck i .

The corresponding spacing error e_{si} for follower truck i is given by

$$e_{si} = x_{i-1} - x_i - L - d_{si} \quad (3.15)$$

where x_{i-1} and x_i are the longitudinal positions of trucks $i - 1$ and i , respectively, and L denotes the truck length.

The kinematic differential equation of the vehicle spacing error is:

$$\dot{e}_{si} = v_{i-1} - v_i - \tau_g a_i \quad (3.16)$$

where v_{i-1} is the longitudinal velocity of truck $i - 1$, and a_i is the longitudinal acceleration of truck i .

3.1.5 Longitudinal Truck Platoon Model

Figure 3.6 shows the configuration of a longitudinal truck platoon, which consists of a lead vehicle and n follower vehicles. Each truck's position, velocity, and acceleration can be measured by the on-board sensors and this information can be shared via V2V communication. In this thesis, V2V communication delays are neglected.

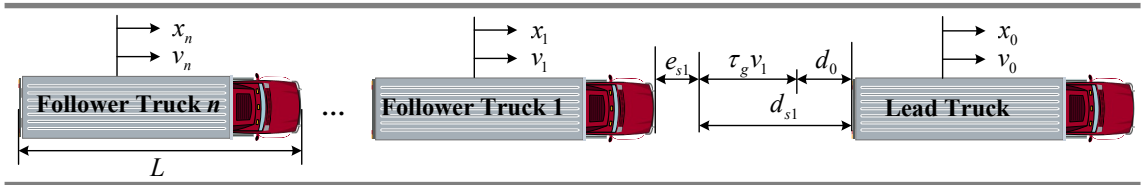


Figure 3.6: Schematic of the longitudinal truck platoon structure.

In a platoon, the drag-reduction phenomenon is achieved through the wake effect generated by the preceding truck, allowing each follower to operate in a modified flow field with reduced effective dynamic pressure. As the follower truck moves closer to the lead vehicle, it becomes increasingly immersed in the low-momentum wake region and experiences a lower effective stagnation pressure at the front of its tractor. This wake sheltering alters the pressure distribution around the follower—particularly at the frontal surfaces—and reduces the pressure differential between the front and rear of the vehicle, leading to a net reduction in aerodynamic drag. Accordingly, the aerodynamic drag coefficient of the follower truck can be modeled as an explicit function of spacing [154], which is represented by an empirical relationship of the form

$$C_D = C_0(1 - \frac{C_1}{C_2 + d}) \quad (3.17)$$

where C_0 , C_1 , and C_2 are calibrated constants and d denotes the longitudinal spacing between adjacent platoon trucks.

Figure 3.7 presents the calibrated results obtained using the experimental data. As the longitudinal spacing decreases, the follower-truck drag coefficient correspondingly decreases, consistent with stronger wake sheltering and enhanced aerodynamic interaction.

To support controller synthesis, the preceding component models are now integrated into a unified platoon-level longitudinal dynamics description. Specifically, for each truck i , the longitudinal motion is governed by the kinematic relation between position and speed, together with a force balance that combines (i) driveline-generated tractive/braking torque, (ii) spacing-dependent aerodynamic drag, and (iii) road-load forces arising from rolling resistance and gravity. To capture the dominant actuator and powertrain response lag while retaining a tractable model for real-time control, the commanded wheel torque is assumed to track a desired torque input through a first-order inertial element. Finally, the inter-vehicle coupling is introduced through the spacing policy, which defines the spacing error dynamics in terms of relative speed

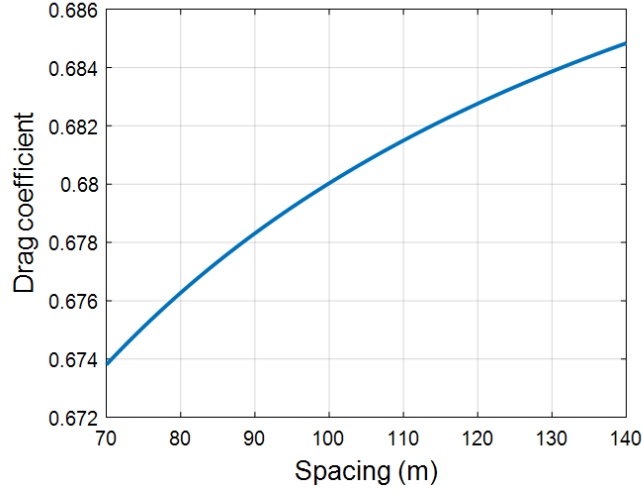


Figure 3.7: Relationship between drag coefficient and inter-vehicle spacing.

and the time-gap term. By collecting these relations, the nonlinear longitudinal dynamics of truck i within the platoon can be written as:

$$\begin{cases} \dot{x}_i = v_i & (3.18a) \\ \dot{v}_i = (\frac{T_i}{r_{eff}} - \frac{1}{2}C_{Di}A\rho(v_x + v_w x)^2 - \mu m_i g \cos \theta_i - m_i g \sin \theta_i) / \delta m_i & (3.18b) \\ \dot{T}_i = (T_{des,i} - T_i) / \tau_a & (3.18c) \\ \dot{e}_{di} = v_{i-1} - v_i - \tau_g \dot{v}_i & (3.18d) \end{cases}$$

where τ_a is the inertial delay of truck's longitudinal dynamics; T_i and $T_{des,i}$ represent the actual and desired driving/braking torque, respectively.

The nonlinear longitudinal platoon model in (3.18) is written in a non-minimal four-state form, since the spacing error e_{di} is algebraically related to x_i and v_i under the constant time-gap policy. Therefore, controllability is examined using the minimal realization $\mathbf{x}_{lon,r,i} = [v_i \ T_i \ e_{di}]^\top$, $\mathbf{u}_{lon,i} = T_{des,i}$. By freezing the exogenous variables at the operating condition, the reduced-order nonlinear system is written in the control-affine form

$$\dot{\mathbf{x}}_{lon,r,i} = f_i(\mathbf{x}_{lon,r,i}) + g_i \mathbf{u}_{lon,i} \quad (3.19)$$

with

$$f_i = \begin{bmatrix} c_i T_i + \phi_i(v_i, e_{di}) \\ -T_i/\tau_a \\ \bar{v}_{i-1} - v_i - \tau_g (c_i T_i + \phi_i(v_i, e_{di})) \end{bmatrix}, \quad g_i = \begin{bmatrix} 0 \\ 1/\tau_a \\ 0 \end{bmatrix} \quad (3.20)$$

where $c_i = 1/\delta m_i r_{\text{eff}}$, and $\phi_i(\cdot)$ collects the nonlinear drag, rolling resistance, road grade, and wind-related terms. The first two Lie brackets are

$$[f_i, g_i] = \begin{bmatrix} -c_i/\tau_a \\ 1/\tau_a^2 \\ \tau_g c_i/\tau_a \end{bmatrix} \quad (3.21)$$

and

$$[f_i, [f_i, g_i]] = \begin{bmatrix} \frac{c_i}{\tau_a^2} \left(\tau_a \left(\frac{\partial \phi_i}{\partial v_i} - \tau_g \frac{\partial \phi_i}{\partial e_{di}} \right) - 1 \right) \\ \frac{1}{\tau_a^3} \\ \frac{c_i}{\tau_a^2} \left(\tau_a \left(\tau_g^2 \frac{\partial \phi_i}{\partial e_{di}} - \tau_g \frac{\partial \phi_i}{\partial v_i} - 1 \right) + \tau_g \right) \end{bmatrix} \quad (3.22)$$

Hence,

$$\begin{bmatrix} g_i & [f_i, g_i] & [f_i, [f_i, g_i]] \end{bmatrix} = \begin{bmatrix} 0 & -c_i/\tau_a & \frac{c_i}{\tau_a^2} \left(\tau_a \left(\frac{\partial \phi_i}{\partial v_i} - \tau_g \frac{\partial \phi_i}{\partial e_{di}} \right) - 1 \right) \\ 1/\tau_a & 1/\tau_a^2 & 1/\tau_a^3 \\ 0 & \tau_g c_i/\tau_a & \frac{c_i}{\tau_a^2} \left(\tau_a \left(\tau_g^2 \frac{\partial \phi_i}{\partial e_{di}} - \tau_g \frac{\partial \phi_i}{\partial v_i} - 1 \right) + \tau_g \right) \end{bmatrix} \quad (3.23)$$

Its determinant is

$$\det \begin{bmatrix} g_i & [f_i, g_i] & [f_i, [f_i, g_i]] \end{bmatrix} = -\frac{c_i^2}{\tau_a^3}. \quad (3.24)$$

Therefore, the Lie algebra rank condition is satisfied when

$$\delta \neq 0, \quad m_i \neq 0, \quad r_{\text{eff}} \neq 0, \quad \tau_a \neq 0. \quad (3.25)$$

Since $\delta > 0$, $m_i > 0$, $r_{\text{eff}} > 0$, and $\tau_a > 0$ for the physical truck system considered in this thesis, the reduced-order nonlinear longitudinal system satisfies the Lie algebra rank condition; therefore, it is locally controllable around the nominal operating condition [155].

3.2 Lateral Platoon Modelling

3.2.1 Lateral Truck Dynamics

The test vehicles used in the on-road CTPS trials are two five-axle tractor-trailer trucks. The tractor includes one steer axle and two rear solid axles, while the trailer is equipped with two solid axles. The lateral dynamics describe the response of the articulated vehicle to lateral tire forces, yaw moments, and aerodynamic forces, as illustrated in Figure 3.8. To balance modeling accuracy with simplicity and computational efficiency for control applications, the following assumptions are made:

- (1) The articulation angle between the tractor and trailer is small, and the tractor and trailer have equal speeds.
- (2) The lateral tire force is linearly proportional to the slip angle.

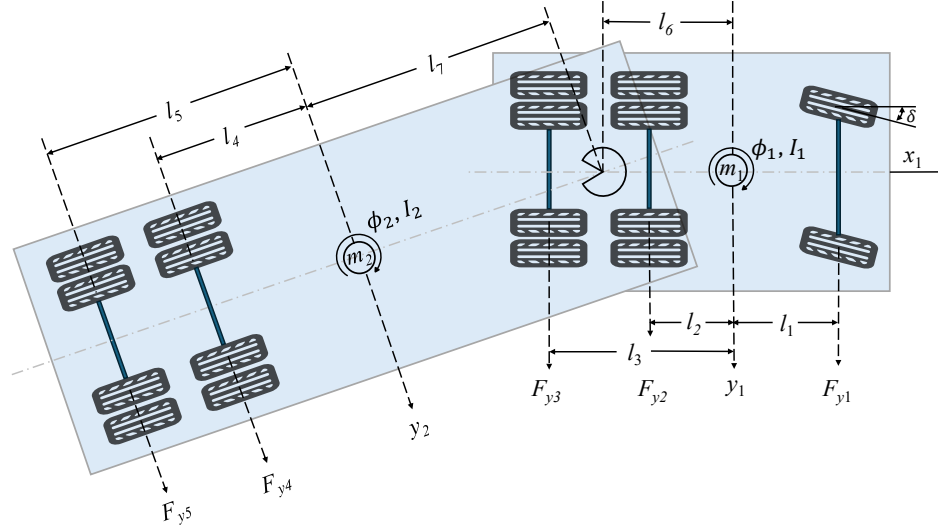


Figure 3.8: The lateral forces acting on a tractor-trailer truck.

Crosswinds can introduce non-negligible lateral aerodynamic loads on both the tractor and trailer, which in turn induce lateral displacement and yaw responses. The lateral aerodynamic force F_{wy} is modeled as:

$$F_{wy} = \frac{1}{2} \rho C_S A_s (\beta) v_{wy}^2 \quad (3.26)$$

where C_S is the aerodynamic side force coefficient, $A_s(\beta)$ is the lateral projection area dependent on apparent wind angle β , and v_{wy} is lateral wind (crosswind) speed.

The lateral motion of the tractor-trailer truck is described by the following equations:

$$m_1\ddot{y}_1 + m_1\dot{x}_1\dot{\phi}_1 = F_{y1} + F_{y2} + F_{y3} - F_h - F_{wy1} \quad (3.27a)$$

$$I_1\ddot{\phi}_1 = l_1F_{y1} - l_2F_{y2} - l_3F_{y3} + l_6F_h \quad (3.27b)$$

$$m_2\ddot{y}_2 + m_2\dot{x}_2\dot{\phi}_2 = F_{y4} + F_{y5} + F_h - F_{wy2} \quad (3.27c)$$

$$I_2\ddot{\phi}_2 = -l_4F_{y4} - l_5F_{y5} + l_7F_h \quad (3.27d)$$

$$\ddot{y}_1 + \dot{x}_1\dot{\phi}_1 - l_6\ddot{\phi}_1 = \ddot{y}_2 + \dot{x}_2\dot{\phi}_2 + l_7\ddot{\phi}_2 \quad (3.27e)$$

where m_1 is the mass of the tractor, $\ddot{y}_1$ is the lateral acceleration of the tractor, $\dot{x}_1$ is the longitudinal speed of the tractor, $\dot{\phi}_1$ is the yaw rate of the tractor, F_{y1} is the lateral wind force acting at the first axle, F_{y2} is the lateral wind force acting at the second axle, F_{y3} is the lateral wind force acting at the third axle, F_h is the lateral force acting at the hitch, F_{wy1} is the lateral wind force acting at the tractor, I_1 is the yaw moment of inertia of the tractor, $\ddot{\phi}_1$ is the yaw acceleration of the tractor, l_1 is the distance from the steering axle to the tractor's center of gravity (CG), l_2 is the distance from the second axle to the tractor's CG, l_3 is the distance from the third axle to the tractor's CG, l_6 is the distance from the hitch to the tractor's CG, m_2 is the mass of the trailer, $\ddot{y}_2$ is the lateral acceleration of the trailer, $\dot{x}_2$ is the longitudinal speed of the trailer, $\dot{\phi}_2$ is the yaw rate of the trailer, F_{y4} is the lateral wind force acting at the fourth axle, F_{y5} is the lateral wind force acting at the fifth axle, F_{wy2} is the lateral wind force acting at the trailer, I_2 is the yaw moment of inertia of the trailer, $\ddot{\phi}_2$ is the yaw acceleration of the trailer, l_4 is the distance from the fourth axle to the trailer's CG, l_5 is the distance from the fifth axle to the trailer's CG, and l_7 is the distance from the hitch to the trailer's CG.

Each force component in Eq. (3.27) is expressed as:

$$F_{y1} = C_1\alpha_1 = C_1(\delta - \frac{\dot{y}_1 + l_1\dot{\phi}_1}{\dot{x}_1}) \quad (3.28a)$$

$$F_{y2} = C_2\alpha_2 = C_2(-\frac{\dot{y}_1 - l_2\dot{\phi}_1}{\dot{x}_1}) \quad (3.28b)$$

$$F_{y3} = C_3\alpha_3 = C_3(-\frac{\dot{y}_1 - l_3\dot{\phi}_1}{\dot{x}_1}) \quad (3.28c)$$

$$F_{y4} = C_4\alpha_4 = C_4(-\frac{\dot{y}_2 - l_4\dot{\phi}_2}{\dot{x}_2}) \quad (3.28d)$$

$$F_{y5} = C_5\alpha_5 = C_5(-\frac{\dot{y}_2 - l_5\dot{\phi}_2}{\dot{x}_2}) \quad (3.28e)$$

$$F_{wy1} = \frac{1}{2}\rho C_S A_{s1}(\beta)v_{wy}^2 \quad (3.28f)$$

$$F_{wy2} = \frac{1}{2}\rho C_S A_{s2}(\beta)v_{wy}^2 \quad (3.28g)$$

where C_1 , C_2 , C_3 , C_4 and C_5 are the cornering stiffness of tires on the first, second, third, fourth and fifth axles, respectively; δ is the steering angle; A_{s1} and A_{s2} are the side projection areas of the tractor and trailer, respectively.

3.2.2 Lateral Offset

The lateral component of atmospheric wind induces lateral offsets between platoon trucks. As shown in Figure 3.9, the lateral offset between two consecutive trucks $i-1$ and i is defined as:

$$e_{l,i} = y_{1,i-1} - y_{1,i} \quad (3.29)$$

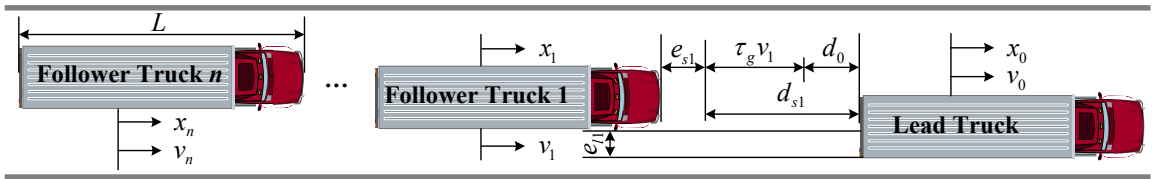


Figure 3.9: Schematic of an n -truck platoon including longitudinal spacing error and lateral offset.

The lateral offset between platooning vehicles modifies the follower truck's exposure to the lead vehicle's wake, thereby altering the pressure distribution around the

follower and changing its aerodynamic drag. This effect is practically relevant because even moderate misalignment can partially move the follower out of the low-drag wake region and reduce the drag-reduction benefit, leading to a measurable increase in fuel consumption [59]. Wind-tunnel measurements of a two-truck platoon further confirm that the follower's drag coefficient varies systematically with lateral offset [74]. Based on the experimental data in Figure 3.10, the relationship between the lateral offset error e_l and the corresponding change in follower drag coefficient ΔC_D is identified using an empirical polynomial fit:

$$\Delta C_D = -1.4558 e_l^6 + 4.6275 e_l^5 - 5.22 e_l^4 + 2.2965 e_l^3 - 0.2264 e_l^2 + 0.0477 e_l + 0.0004 \quad (3.30)$$

where ΔC_D represents the incremental change relative to the laterally aligned condition ($e_l = 0$). In the platoon model, this term can be used to correct the follower's drag coefficient by augmenting the spacing-dependent baseline. For broader applicability, the lateral offset error may be normalized by the vehicle width.

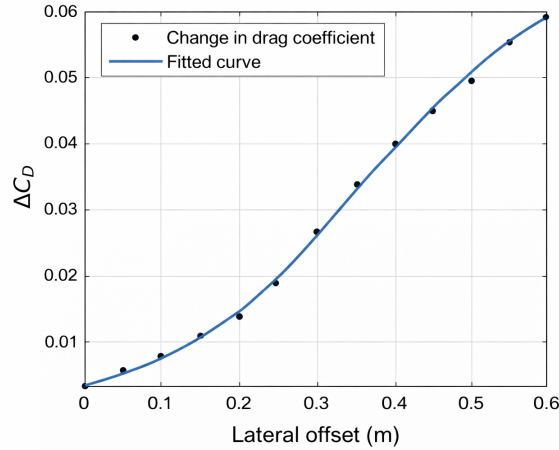


Figure 3.10: Change in the follower truck's drag coefficient due to lateral offset using experimental truck data.

3.2.3 Lateral Truck Platoon Model

For control synthesis, the tractor-trailer lateral dynamics in Eqs. (3.26)–(3.29) can be written in a compact nonlinear state-space form for truck i as:

$$\left\{ \begin{aligned} m_{1,i}\ddot{y}_{1,i} + m_{2,i}\ddot{y}_{2,i} &= C_1\delta_i - \frac{C_1 + C_2 + C_3}{\dot{x}_{1,i}}\dot{y}_{1,i} - \left(\frac{C_1l_1 - C_2l_2 - C_3l_3}{\dot{x}_{1,i}} \right. \end{aligned} \right. \quad (3.31a)$$

$$\begin{aligned} &+ m_{1,i}\dot{x}_{1,i})\dot{\phi}_{1,i} - \frac{C_4 + C_5}{\dot{x}_{1,i}}\dot{y}_{2,i} \\ &+ \left(\frac{C_4l_4 + C_5l_5}{\dot{x}_{1,i}} - m_{2,i}\dot{x}_{1,i}\right)\dot{\phi}_{2,i} \\ &- \frac{1}{2}\rho C_S(A_{s1,i}(\beta) + A_{s2,i}(\beta))v_{wy,i}^2 \end{aligned}$$

$$I_{1,i}\ddot{\phi}_{1,i} - l_6m_{2,i}\ddot{y}_{2,i} = C_1l_1\delta + \frac{-C_1l_1 + C_2l_2 + C_3l_3}{\dot{x}_{1,i}}\dot{y}_{1,i} \quad (3.31b)$$

$$\begin{aligned} &- \left(\frac{C_1l_1^2 + C_2l_2^2 + C_3l_3^2}{\dot{x}_{1,i}} - m_{1,i}\dot{x}_{1,i}\right)\dot{\phi}_{1,i} \\ &+ \frac{C_4l_6 + C_5l_6}{\dot{x}_{1,i}}\dot{y}_{2,i} - l_6\left(\frac{C_4l_4 + C_5l_5}{\dot{x}_{1,i}} \right. \\ &- m_{2,i}\dot{x}_{1,i})\dot{\phi}_{2,i} + \frac{1}{2}l_6\rho C_S A_{s2,i}(\beta)v_{wy,i}^2 \end{aligned}$$

$$I_{2,i}\ddot{\phi}_{2,i} - l_7m_{2,i}\ddot{y}_{2,i} = \frac{C_4l_4 + C_4l_7 + C_5l_5 + C_5l_7}{\dot{x}_{1,i}}\dot{y}_{2,i} \quad (3.31c)$$

$$\begin{aligned} &- \left(\frac{C_4l_4^2 + C_4l_4l_7 + C_5l_5^2 + C_5l_5l_7}{\dot{x}_{1,i}} \right. \\ &- l_7m_{2,i}\dot{x}_{1,i})\dot{\phi}_{2,i} + \frac{1}{2}l_7\rho C_S A_{s2,i}(\beta)v_{wy,i}^2 \end{aligned}$$

$$\ddot{y}_{1,i} - l_6\ddot{\phi}_{1,i} - \ddot{y}_{2,i} - l_7\ddot{\phi}_{2,i} = -\dot{x}_{1,i}\dot{\phi}_{1,i} + \dot{x}_{2,i}\dot{\phi}_{2,i} \quad (3.31d)$$

$$\dot{e}_{l,i} = \dot{y}_{1,i-1} - \dot{y}_{1,i} \quad (3.31e)$$

The lateral dynamics are coupled with the longitudinal dynamics via the longitudinal velocity, which is modeled as a time-varying parameter in the lateral dynamics.

Since the lateral model in (3.31) is non-minimal when both $y_{1,i}$ and $e_{l,i}$ are retained, the controllability analysis is performed using the reduced state vector $\mathbf{x}_{lat,r,i} = [e_{l,i} \ \phi_{1,i} \ \dot{y}_{1,i} \ \dot{\phi}_{1,i} \ \dot{y}_{2,i} \ \dot{\phi}_{2,i}]^\top$, with steering angle δ_i as the control input.

For a frozen operating condition satisfying $\dot{x}_{1,i} = \dot{x}_{2,i} > 0$, define

$$\mathbf{q}_i = [\dot{y}_{1,i} \ \dot{\phi}_{1,i} \ \dot{y}_{2,i} \ \dot{\phi}_{2,i}]^\top. \quad (3.32)$$

Then, the four lateral dynamic equations in (3.31) can be written as

$$M_i \dot{\mathbf{q}}_i = N_i \mathbf{q}_i + b_i \delta_i + d_i, \quad (3.33)$$

where

$$M_i = \begin{bmatrix} m_{1,i} & 0 & m_{2,i} & 0 \\ 0 & I_{1,i} & -l_6 m_{2,i} & 0 \\ 0 & 0 & -l_7 m_{2,i} & I_{2,i} \\ 1 & -l_6 & -1 & -l_7 \end{bmatrix}, \quad b_i = \begin{bmatrix} C_1 \\ C_1 l_1 \\ 0 \\ 0 \end{bmatrix}. \quad (3.34)$$

If M_i is nonsingular, the reduced model can be expressed as

$$\dot{\mathbf{x}}_{lat,r,i} = A_{lat,i} \mathbf{x}_{lat,r,i} + B_{lat,i} \delta_i + \mathbf{d}_{lat,i}. \quad (3.35)$$

Because the frozen-speed model is affine in the state, the Lie algebra rank condition reduces to

$$\text{rank} [B_{lat,i} \ A_{lat,i} B_{lat,i} \ A_{lat,i}^2 B_{lat,i} \ A_{lat,i}^3 B_{lat,i} \ A_{lat,i}^4 B_{lat,i} \ A_{lat,i}^5 B_{lat,i}] = 6. \quad (3.36)$$

Therefore, the reduced lateral model is locally controllable around the nominal operating condition, provided that $\dot{x}_i > 0$, M_i is nonsingular, $C_1 > 0$, and no singular combination of geometric, inertia, and tire parameters causes the above rank condition to drop.

3.3 Model Validation

To assess the accuracy of the developed platoon truck dynamics models, experimental data from the on-road truck platooning trials are used. Figure 3.11 summarizes the validation procedure. The recorded steering angle, engine torque, and brake torque of the follower truck are applied as inputs to the developed platoon truck models. In addition, measured road profiles as well as the recorded wind speed and direction are incorporated to represent realistic driving environments. The model outputs are then compared with the experimental measurements, which serve as the ground truth for validation.

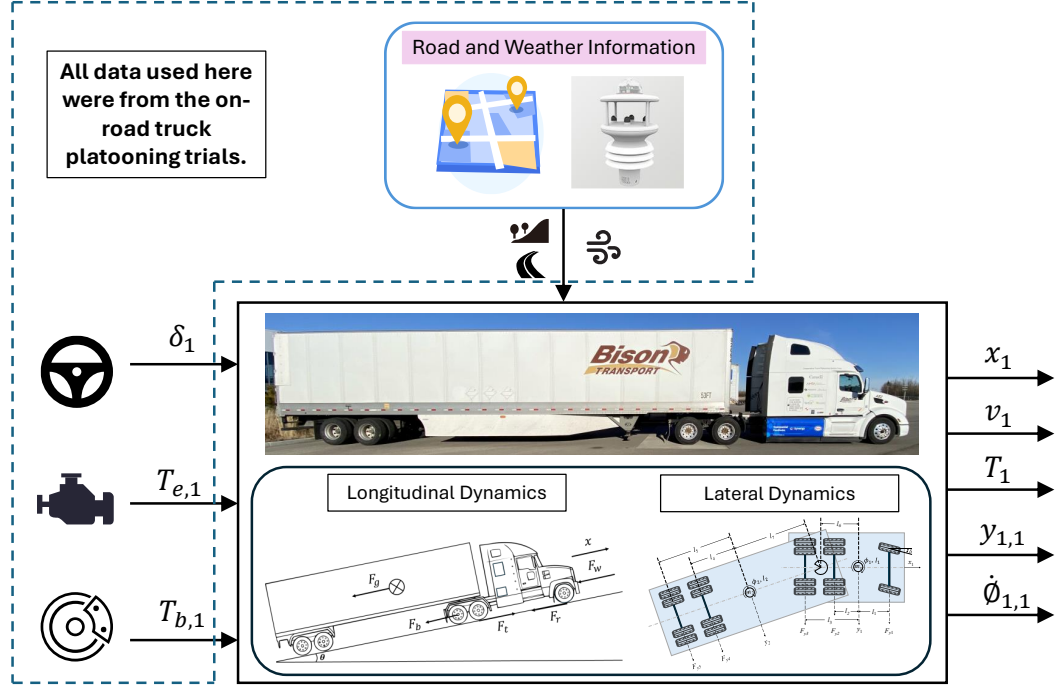


Figure 3.11: Validation procedures for the developed truck dynamics models.

A representative route segment on Alberta Highway 2 is selected to validate the developed models. Over this segment, the gross vehicle mass is 25,030 kg, the road grade ranges from -4.9% to 5.3% , the wind speed ranges from 5 to 19 km/h, and the wind direction varies from 13° to 45° .

Figure 3.12 presents the validation results of the developed truck models against experimental platooning data. As shown in Figure 3.12(a), the simulated truck speed trajectory closely matches the experimental measurements, with a mean absolute percentage error (MAPE) of 0.9%. Furthermore, the simulated longitudinal spacing agrees well with the measurements, yielding an MAPE of 1.8% (Figure 3.12(b)). These errors are small relative to the typical speed and spacing variations observed in heavy-duty truck platooning, supporting the longitudinal fidelity of the model. Figure 3.12(c) demonstrates the model's lateral performance, where the lateral offset shows close agreement with an MAPE of 1.7%, which is within the tolerance required for safe platoon operation.

The model further demonstrates high fidelity in predicting energy and emissions

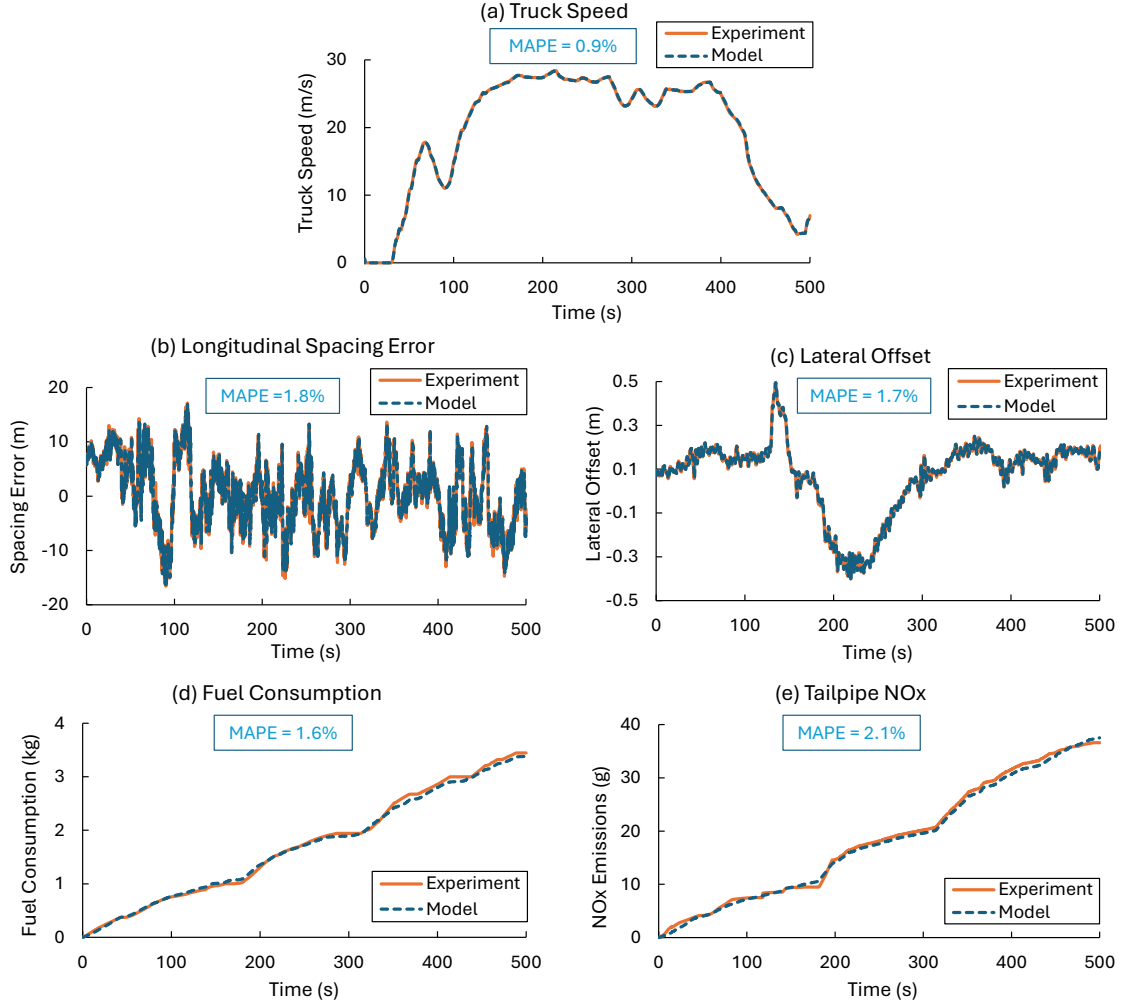


Figure 3.12: Validation results of the developed truck models under the selected Alberta Highway 2 driving cycles with the test conditions: The gross vehicle mass is 25,030 kg, the road grade ranges from -4.9% to 5.3% , the wind speed ranges from 5 to 19 km/h, the wind direction varies from 13° to 45° , and the ambient temperature is -7°C . MAPE stands for mean absolute percentage error.

metrics. The cumulative fuel consumption deviates by only 1.6% from the experimental measurement (Figure 3.12(d)), while the tailpipe NO_x prediction differs by 2.1% (Figure 3.12(e)). These deviations are smaller than the uncertainties typically associated with on-road emissions measurements [149]. Collectively, the results indicate that the developed models can reliably reproduce vehicle dynamics, energy consumption, and tailpipe NO_x emissions under realistic operating conditions, providing a credible foundation for the platoon control design in subsequent chapters.

3.4 Summary of Chapter

This chapter developed and validated an integrated modelling framework for Class-8 truck platoons to support simulation, analysis, and subsequent controller synthesis under regulation-relevant operating conditions. The main findings are summarized as follows:

- (1) A unified modelling structure was established to capture the key mechanisms governing platoon motion, energy consumption, and tailpipe emissions, including longitudinal and lateral vehicle dynamics, conventional diesel powertrain dynamics, map-based fuel-consumption modelling, and data-driven tailpipe NO_x prediction.
- (2) A nonlinear longitudinal platoon model was formulated by combining longitudinal force balance with a powertrain/driveline model. Aerodynamic interactions were incorporated by modelling the follower-truck drag coefficient as an explicit function of inter-vehicle spacing, enabling the model to represent drag reduction due to wake sheltering effects.
- (3) A control-oriented tractor-trailer lateral dynamics model was derived under reasonable assumptions (small articulation angle and linear tire cornering stiffness). Crosswind-induced lateral aerodynamic forces were included, and the effect of lateral offset on the follower's drag coefficient was characterized using an empirical fit, thereby linking lateral alignment to aerodynamic and energy outcomes.
- (4) Motivated by the observed correlation between tailpipe NO_x emissions and engine torque transients, a deep-learning LSTM model was developed to predict tailpipe NO_x based on powertrain and operating variables. The model was trained on a large real-world dataset collected during the on-road trials to capture transient aftertreatment behavior beyond map-based approaches.
- (5) The developed models were validated against independent datasets from the

CTPS trials by applying measured control inputs (steering angle, engine torque, and brake torque) and incorporating recorded road grade and wind conditions. The validation demonstrated close agreement between simulated and measured trajectories and outcomes, achieving low errors in speed, longitudinal spacing, yaw rate, lateral offset, cumulative fuel consumption, and tailpipe NO_x emissions.

Overall, the results confirm that the proposed modelling framework can accurately reproduce the coupled longitudinal–lateral dynamics of truck platoons and provide reliable estimates of fuel consumption and tailpipe NO_x emissions under realistic driving conditions such as road-grade changes and wind variations. These validated models provide the high-fidelity foundation for the hierarchical platoon coordination and integrated longitudinal–lateral control designs developed in the subsequent chapters.

PART IV: Truck Platooning Control System Design

Chapter 4

Truck Platooning under Varying Road Grades¹

Road grade is a key factor that can substantially alter the energy and emissions performance of truck platoons, particularly under regulation-compliant time gaps where aerodynamic benefits are more limited and powertrain effects become more pronounced. Results from the on-road CTPS trials indicate that, on flat road segments, the follower truck achieved an average fuel saving of 1.6% at permitted time gaps. However, this benefit deteriorated markedly when the route included sustained grade changes. As summarized in Table 4.1, the follower truck experienced increased fuel consumption on both uphill and downhill sections, with the largest penalty observed on steep uphill, where fuel consumption increased by up to 12.5%. Similar grade-induced degradations have also been reported in close-spacing platooning experiments [66, 67].

This trend is primarily attributable to insufficient coordination between the lead and follower vehicles when traversing varying slopes. Under conventional same-speed tracking objectives, the follower can be forced into non-optimal longitudinal actions because it reacts to the lead truck’s speed changes without accounting for the spatial mismatch in road grade along the route. For instance, when the lead truck enters an uphill segment and slows due to power saturation, the follower may unnecessarily

¹ This chapter is partially based on [3, 11].

reduce speed while still on a flatter segment, thereby inducing avoidable deceleration and subsequent re-acceleration. More generally, grade variations induce large swings in required tractive effort: uphill segments increase power demand and engine load, whereas downhill segments may trigger service braking to respect speed limits and spacing constraints. These effects can amplify speed fluctuations and lead to inefficient torque trajectories, which increases fuel consumption relative to well-coordinated operation.

Table 4.1: Follower truck’s fuel saving and tailpipe NO_x reduction under different road grades in the on-road truck platooning trials.

Road Type	Road Grade (%)	Fuel Saving (%)	NO _x Reduction (%)
Steep downhill	-2.5 to -5.8	-7.8	-32.1
Downhill	-1 to 2.5	-3.9	-17.0
Flat	-1 to 1	1.6	2.7
Uphill	1 to 2.5	-4.8	-22.9
Steep uphill	2.5 to 5.6	-12.5	-38.3

* Fuel saving and NO_x reduction with a minus sign indicates an increase in fuel consumption and NO_x emissions of the follower truck.

Road grade variations also have a pronounced impact on tailpipe NO_x emissions. Table 4.1 shows that grade changes can substantially increase tailpipe NO_x, which is consistent with the fact that grade-induced acceleration/deceleration cycles produce rapid engine torque transients. In particular, the elevated torque fluctuations under hilly operation exacerbate transient NO_x formation and reduce the effectiveness of downstream conversion when exhaust conditions deviate from their optimal ranges.

To improve fuel savings and reduce tailpipe NO_x under real-world operation with varying road gradients, this chapter develops an innovative hierarchical two-level control system, including a platoon-level coordination controller and a lower-level controller. The proposed architecture explicitly incorporates road-grade effects into speed and torque planning while maintaining regulation-compliant spacing for safety. Moreover, motivated by the observed sensitivity of tailpipe NO_x to engine torque

oscillations, the controller is designed to suppress unnecessary torque transients by coordinating the follower’s longitudinal response with the lead vehicle’s motion and the upcoming grade profile. This two-level design provides a practical pathway to recover platooning benefits on hilly terrain while simultaneously improving energy efficiency and emission outcomes.

4.1 Hierarchical Control Architecture

This chapter adopts a hierarchical architecture for designing the longitudinal truck-platooning control system to enable regulation-compliant operation while improving energy efficiency and mitigating tailpipe NO_x emissions. As shown in Figure 4.1, the control system comprises an upper-level supervisory controller and a lower-level actuator-tracking controller. The upper-level platoon controller determines a desired torque trajectory for the follower truck by leveraging road-grade information and V2V-shared lead-vehicle states. This supervisory layer enforces platoon-level coordination objectives, including maintaining safe spacing, smoothing the follower’s response, and suppressing unnecessary powertrain transients. The lower-level controller then tracks the desired torque by generating realizable actuator commands (fuel injection quantity/brake pressure) subject to drivetrain dynamics. For the lead truck, a corresponding lower-level controller tracks a prescribed speed profile (driving cycle) and produces the commanded torque that defines the lead-vehicle motion shared with the follower through V2V communication. This hierarchical separation allows coordination decisions to be made at the platoon level while preserving real-time execution at the actuator level.

Because platoon dynamics are nonlinear, the upper-level controller is implemented using nonlinear model predictive control (NMPC), while the lower-level controller employs proportional–integral–derivative (PID) control for torque tracking. Figure 4.2 presents the NMPC–PID block diagram. The NMPC computes the follower’s torque request from V2V-shared vehicle states and onboard measurements; communication

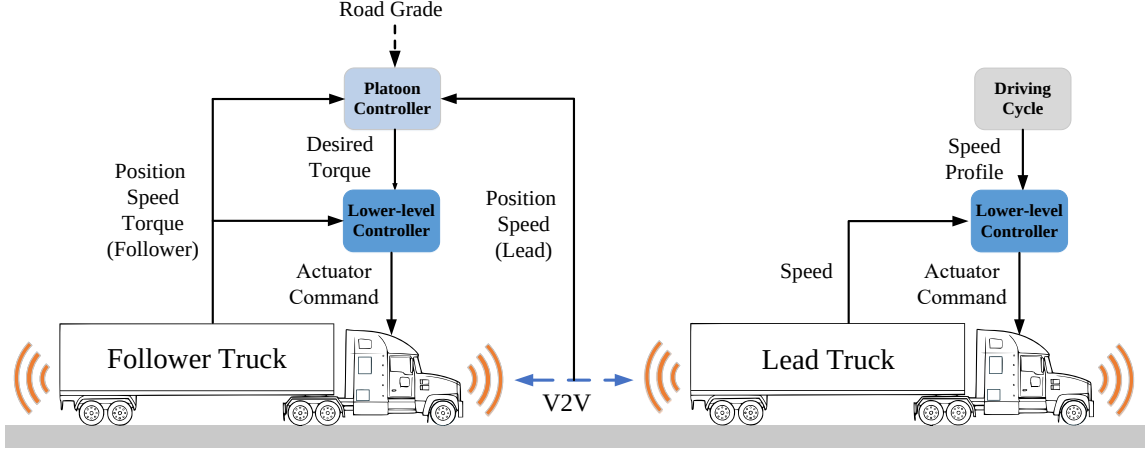


Figure 4.1: Designed control structure of the truck platooning system.

delays are not considered in this thesis (see [156, 157] for delay-aware designs). The PID layer maps the requested net torque to actuator commands, applying fueling for traction (positive torque) and braking for deceleration (negative torque).

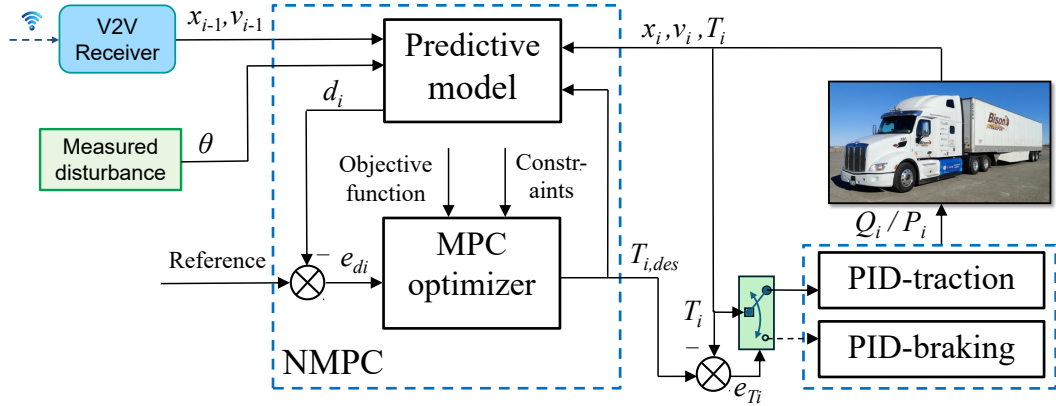


Figure 4.2: Block diagram of NMPC-PID control structure. NMPC stands for non-linear model predictive control, and PID stands for proportional–integral–derivative.

4.2 Design of Platoon Control System

4.2.1 Upper-Level Controller

The proposed NMPC is implemented in a discrete domain, with the inter-vehicle dynamics model discretized using the first-order Euler approximation [158], yielding

$$\begin{cases} x_i(k+1) = x_i(k) + v_i(k)\Delta t & (4.1a) \\ v_i(k+1) = v_i(k) + \frac{\Delta t}{\delta m_i} \left(\frac{T_i(k)}{r_{eff}} - F_i(v_i(k)) \right) & (4.1b) \\ T_i(k+1) = T_i(k) + \frac{1}{\tau_a} T_{des,i}(k)\Delta t - \frac{1}{\tau_a} T_i(k)\Delta t & (4.1c) \\ e_{di}(k+1) = e_{di}(k) + v_{i-1}(k)\Delta t - v_i(k)\Delta t \\ \quad - \tau_g \frac{\Delta t}{\delta m_i} \left(\frac{T_i(k)}{r_{eff}} - F_i(v_i(k)) \right) & (4.1d) \\ F_i(v_i(k)) = \frac{1}{2} C_{Di} A \rho (v_i(k) + v_{wxi}(k))^2 \\ \quad - \mu m_i g \cos \theta(x_i(k)) - m_i g \sin \theta(x_i(k)) & (4.1e) \end{cases}$$

where Δt is the sampling time; k represents the current time step; T_i and $T_{des,i}$ represent the actual and desired driving/braking torque for truck i , respectively.

A receding control framework is employed to implement the NMPC, where at each discrete time step, an optimal set of control inputs is computed over a prediction horizon using an optimizer. An objective function is defined including four sections as shown in Eq. (4.2). The first term penalizes errors in tracking the desired spacing between trucks. This ensures the safety of the truck platoon by maintaining appropriate inter-vehicle distances. The second term penalizes engine torque to optimize fuel efficiency, encouraging the use of minimal and efficient torque settings. The third term penalizes fluctuations in the follower truck's speed. This smooths the speed profile, which also helps in reducing fuel consumption. The fourth term penalizes aggressive changes in engine torque to reduce tailpipe NO_x emissions. Since transient engine torque variations are closely correlated with NO_x spikes, this term serves as a computationally efficient proxy for modeling NO_x emission dynamics within a real-time optimization framework. An alternative approach for reducing NO_x is using NO_x

term directly in the cost function, but this approach will increase the computation cost substantially due to complex and nonlinear terms in the NO_x model.

Accordingly, the NMPC optimization problem is formulated as:

$$\begin{aligned}
\min_{T_{des,i}(k)} J(k) = & \overbrace{\sum_{j=1}^{N_p} \sum_{i=1}^n \frac{w_e}{s_e} \|e_{di}(k+j|k)\|^2}^{\text{Platoon Safety}} + \overbrace{\sum_{j=1}^{N_c} \sum_{i=1}^n \frac{w_T}{s_T} \|T_{ei,des}(k+j|k)\|^2}^{\text{Fuel Consumption Reduction}} \\
& + \overbrace{\sum_{j=1}^{N_p} \sum_{i=1}^n \frac{w_v}{s_v} \|v_i(k+j|k) - v_i(k+j-1|k)\|^2}^{\text{Smooth Speed Profile \& Fuel Consumption Reduction}} \\
& + \overbrace{\sum_{j=1}^{N_c-1} \sum_{i=1}^n \frac{w_{\Delta T}}{s_{\Delta T}} \|T_{ei,des}(k+j|k) - T_{ei,des}(k+j-1|k)\|^2}^{\text{NO}_x \text{ Emissions Reduction}}
\end{aligned} \tag{4.2}$$

subject to

$$x_i(k+j+1|k) = x_i(k+j|k) + v_i(k+j|k)\Delta t \tag{4.3a}$$

$$v_i(k+j+1|k) = v_i(k+j|k) \tag{4.3b}$$

$$\begin{aligned}
& + \frac{\Delta t}{\delta_m m_i} \left(\frac{T_i(k+j|k)}{r_{eff}} - F_i(v_i(k+j|k)) \right) \\
T_i(k+j+1|k) = & T_i(k+j|k) + \frac{1}{\tau_a} T_{des,i}(k+j|k)\Delta t
\end{aligned} \tag{4.3c}$$

$$\begin{aligned}
& - \frac{1}{\tau_a} T_i(k+j|k)\Delta t \\
e_{di}(k+j+1|k) = & e_{di}(k+j|k) + v_{i-1}(k+j|k)\Delta t - v_i(k+j|k)\Delta t \\
& - \tau_g \frac{\Delta t}{\delta_m m_i} \left(\frac{T_i(k+j|k)}{r_{eff}} - F_i(v_i(k+j|k)) \right)
\end{aligned} \tag{4.3d}$$

$$\begin{aligned}
F_i(v_i(k)) = & \frac{1}{2} C_{Di} A \rho (v_i(k+j|k) + v_{wxi}(k+j|k))^2 \\
& - \mu m_i g \cos \theta (x_i(k+j|k)) - m_i g \sin \theta (x_i(k+j|k))
\end{aligned} \tag{4.3e}$$

$$e_{di}^{\min} \leq e_{di}(k+j|k) \leq e_{di}^{\max} \tag{4.3f}$$

$$v_i^{\min} \leq v_i(k+j|k) \leq v_i^{\max} \tag{4.3g}$$

$$0 \leq T_{ei}(k+j|k) \leq T_{ei}^{\max} \tag{4.3h}$$

$$0 \leq T_{bi}(k+j|k) \leq T_{bi}^{\max} \tag{4.3i}$$

where N_p is the prediction horizon, N_c is the control horizon, n is the number of platoon vehicles, w is the tuning weight, s is the scale factor, $e_{di}^{\min}$ and $e_{di}^{\max}$ are

the minimum and maximum inter-vehicle distance error, respectively; $v_i^{\min}$ and $v_i^{\max}$ denote the minimum and maximum bounds of the trucks' speed, respectively; $T_{ei}^{\max}$ and $T_{bi}^{\max}$ represent the maximum engine torque and the maximum braking torque, respectively.

For practical implementation, multiple constraints are introduced for each vehicle in the platoon. To ensure that torque commands are dynamically feasible given the current speed, road grade, and resistance forces, the system dynamics constraints are applied, as shown in Eqs. (4.3a)–(4.3e). To ensure safe and stable platoon operation under real-world operations, a safety buffer is reserved to avoid collisions and absorb any deviation of the follower truck from the desired separation distances, as defined in Eq. (4.3f). In addition, to observe the speed limit and maintain traffic efficiency on highways, a constraint is placed on the truck's speed in Eq. (4.3g). Finally, considering the peak torque of the Cummins X15 engine and the capacity of the braking system, constraints are imposed on the engine torque in Eq. (4.3h) and braking torque in Eq. (4.3i) to prevent exceeding allowable engine/braking torque.

In this chapter, the sampling time, Δt , is set to 0.1 sec by considering time scale of vehicle dynamics in truck platooning. The prediction horizon, N_p , is chosen as 50, which provides the prediction time of 5 sec to cover the largest time gap in this thesis. The control horizon, N_c , is often selected to be less than the prediction horizon (10% to 20%) to improve computational efficiency [159]; here, N_c is chosen as 10. The tuning weights are selected to balance the trade-off among tracking state and input reference trajectories and managing actuator excitation. To solve the optimization problem in NMPC, the sequential quadratic programming (SQP) method is adopted. The SQP algorithm as the built-in solver in the Optimization Toolbox in Matlab is used to find the feasible solution. At each control time step, the control input ($T_{des,i}(k)$) is calculated by minimizing the cost function subject to constraints. Only the first part of the computed optimal control input sequence is applied to the system during the next sampling interval, after which the procedure repeats in a receding-horizon

manner.

4.2.2 Lower-Level Controller

The lower-level controller tracks the net torque request from the upper-level NMPC using actuator-level commands while respecting engine/brake dynamics and limits. The desired net torque $T_{\text{des}}(k)$ is allocated using a simple supervisory logic:

$$\text{traction mode: } T_{\text{des}}(k) \geq 0, \quad \text{braking mode: } T_{\text{des}}(k) < 0 \quad (4.4)$$

To avoid frequent mode switching around zero torque (which can lead to actuator chatter and degraded ride quality), a small hysteresis band can be applied in implementation, and the torque request can be smoothly ramped using a rate limiter:

$$|T_{\text{des}}(k) - T_{\text{des}}(k-1)| \leq \Delta T_{\text{max}} \quad (4.5)$$

where ΔT_{max} is the maximum allowable torque step per sampling interval. This treatment is consistent with the upper-level objective that penalizes aggressive torque changes and further improves closed-loop smoothness.

When $T_{\text{des}}(k) \geq 0$, the engine operates in traction mode and the lower-level controller regulates the injected fuel quantity Q so that the delivered engine torque tracks the desired value. A proportional–integral–derivative (PID) controller is employed:

$$Q(k) = K_{P,e}E(k) + K_{I,e}\Delta t \sum_{j=1}^k E(j) + \frac{K_{D,e}}{\Delta t}\Delta E(k), \quad (4.6)$$

where the torque tracking error is defined as $E(k) = T_{\text{des}}(k) - T_{\text{act}}(k)$, $T_{\text{act}}(k)$ is the actual delivered torque, and $\Delta E(k) = E(k) - E(k-1)$. The integral term compensates steady-state bias caused by modeling uncertainty, while the derivative term provides damping to reduce overshoot under fast transients. The commanded fuel quantity is further constrained to remain within physically feasible bounds:

$$Q_{\min} \leq Q(k) \leq Q_{\max} \quad (4.7)$$

For the Cummins X15 engine used in this thesis, the maximum fuel consumption rate is 1.3 L/min.

When $T_{\text{des}}(k) < 0$, a braking torque is required and the service brake is activated. In this mode, the lower-level controller generates the pneumatic brake pressure command $P(k)$ to track the desired braking torque. A PID controller is used:

$$P(k) = K_{P,b}E(k) + K_{I,b}\Delta t \sum_{j=1}^k E(j) + \frac{K_{D,b}}{\Delta t}\Delta E(k) \quad (4.8)$$

The brake pressure command is constrained by the pneumatic system limits:

$$P_{\min} \leq P(k) \leq P_{\max} \quad (4.9)$$

These bounds help prevent excessive brake actuation, reduce the risk of wheel slip on low-friction surfaces. In practice, $P(k)$ can be mapped to an effective braking force through the calibrated relationship $F_b = K_b P_b$ used in the vehicle model.

The lower-level controller operates at the same sampling rate as the upper-level controller and uses onboard measurements to close the torque tracking loop. The PID gains $\{K_{P,e}, K_{I,e}, K_{D,e}\}$ and $\{K_{P,b}, K_{I,b}, K_{D,b}\}$ are tuned to ensure stable and well-damped torque responses under typical operating conditions. With this structure, the upper-level NMPC focuses on planning torque trajectories that satisfy spacing and efficiency objectives, whereas the lower-level controller ensures that these torque requests are accurately and smoothly executed by the physical actuators in real time.

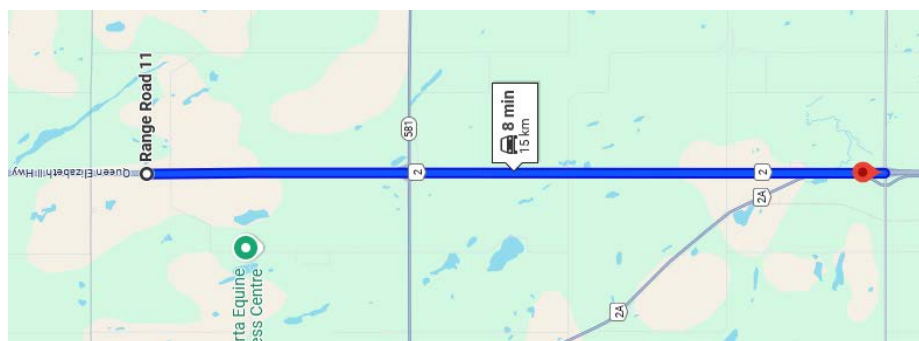
4.3 Numerical Evaluation and Discussion

4.3.1 Simulation Setup

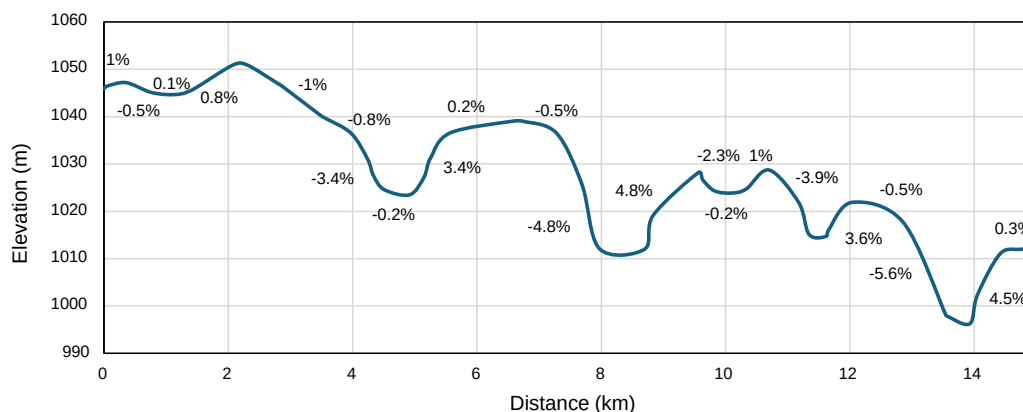
Due to the high logistical and financial demands associated with conducting additional full-scale platooning experiments, this thesis adopts a simulation-based evaluation to assess the performance of the proposed hierarchical platoon control system. All simulations are carried out using the experimentally validated models developed in Chapter 3 and are informed by measurements collected during the on-road cooperative

truck platooning trials. The simulated platoon consists of two identical Class-8 heavy-duty trucks, and the Alberta Highway 2 driving cycle is adopted to represent real-world long-haul operation.

To verify the effectiveness of the designed controller under challenging driving conditions, the simulations incorporate a representative hilly segment of Alberta Highway 2 and the corresponding measured wind profiles, as shown in Figures 4.3 and 4.4. Figure 4.3 summarizes the selected road geometry, which is straight but features significant grade variations. The maximum and minimum grades in the selected section are 4.8% and -5.6%, respectively, capturing the representative hilly terrain characteristics of this corridor.



(a) Selected route on Alberta Highway 2 for simulations. This navigation map was generated from Google Maps.



(b) Road elevations and grades (shown in %) of the selected section of Canada-Alberta Highway 2 used in this study.

Figure 4.3: Road specifications of the selected section of Canada-Alberta Highway 2.

Figure 4.4 shows the measured wind conditions used in the simulations, where wind

speed varies from 5.2 to 7.3 km/h and wind angle ranges between 146° and 181° .

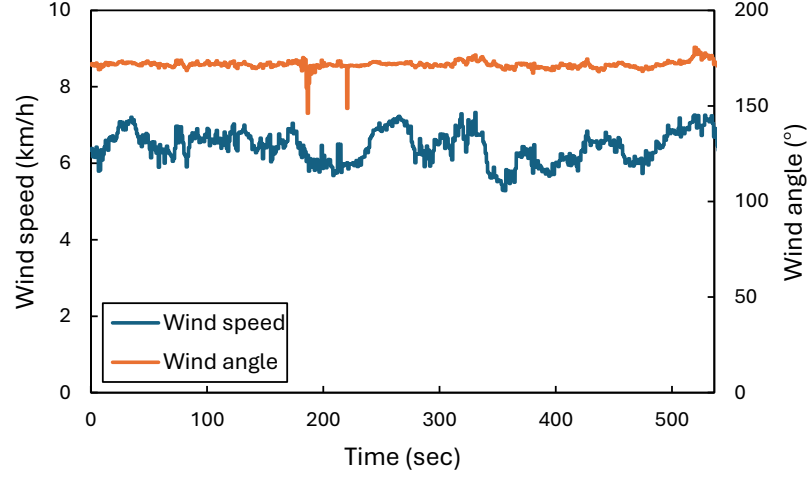


Figure 4.4: Speed and angle of measured winds.

The simulations are implemented in Matlab/Simulink on a PC platform, and the key parameters of the truck and the platoon controller are summarized in Table 4.2. Since the platoon distance between the lead and follower truck in this study is substantial, platooning would not have a measurable influence on the lead truck [34]. Therefore, the lead truck is used as the control truck and serves as the baseline for comparison.

4.3.2 Simulation Results and Analyses

Figure 4.5 presents the results of truck platooning using the designed controller on the hilly sections of Alberta Highway 2 with a time gap of 5 sec. The controller effectively maintains the desired spacing between the follower and lead trucks. Figure 4.5(a) shows that the maximum spacing error is less than 0.3 m, well within the safety buffer of 5 m. This indicates the controller successfully ensures the safety of trucks under platooning. Figure 4.5(b) shows that the follower truck exhibits fewer speed fluctuations compared to the lead truck. To quantify speed smoothness, we use an index defined as the integral of the squared second-order derivative of the speed curve function $C(t)$ [160]. Furthermore, the dimensionless smoothness

Table 4.2: Key parameters of the truck and platoon controller.

Parameter	Symbol	Value	Unit
Truck Mass	m	16022	kg
Rotational Mass Coefficient	δ_m	1.05	-
Effective Rolling Radius	r_{eff}	0.5	m
Air Drag Coefficient	C_D	0.75	-
Frontal Area	A	10	m ²
Air Density	ρ	1.2922	kg/m ³
Rolling Resistance Coefficient	μ	0.006	-
Gravitational Acceleration	g	9.81	m/s ²
Gear Ratio	i_g	0.77-14.41	-
Final Drive Ratio	i_0	2.85	-
Mechanical Efficiency	η_t	0.9	-
Sampling Time	Δt	0.1	sec
Tuning Weight	w_e	10	-
Scale Factor	s_e	1	-
Tuning Weight	w_v	5	-
Scale Factor	s_v	1	-
Tuning Weight	w_T	2	-
Scale Factor	s_T	2372	-
Tuning Weight	$w_{\Delta T}$	5	-
Scale Factor	$s_{\Delta T}$	160	-
Minimum Spacing error	$e_d^{\min}$	-5	m
Maximum Spacing error	$e_d^{\max}$	5	m
Minimum Truck Speed	$v^{\min}$	20	m/s
Maximum Truck Speed	$v^{\max}$	30	m/s
Peak Engine Torque	$T_e^{\max}$	2272	N·m

is calculated by $\int_{t_1}^{t_2} C''(t)^2 dt / C_{max}^2$, where C_{max} represents the maximum value of this curve. Lower smoothness values indicate smoother speed profiles. The lead truck's speed smoothness is 120.8, while the follower truck's speed smoothness is 32.5, confirming the effectiveness of the controller in moderating speed oscillations. This smoother speed trajectory for the follower is also evident in the acceleration profiles in Figure 4.5(c). At this 5-sec time gap, where aerodynamic drag benefits are marginal, the coordinated speed profiling becomes the primary contributor to fuel savings for the follower truck, as reflected in the fuel rate profile shown in Figure 4.5(e). The most significant reductions in fuel rate occur during accelerations and decelerations,

where the proposed control system effectively smoothes the truck speed trajectories. During cruising, however, the fuel rates of the two trucks are more comparable. These results highlight the critical role of speed smoothing in enhancing fuel efficiency under large spacing platooning conditions.

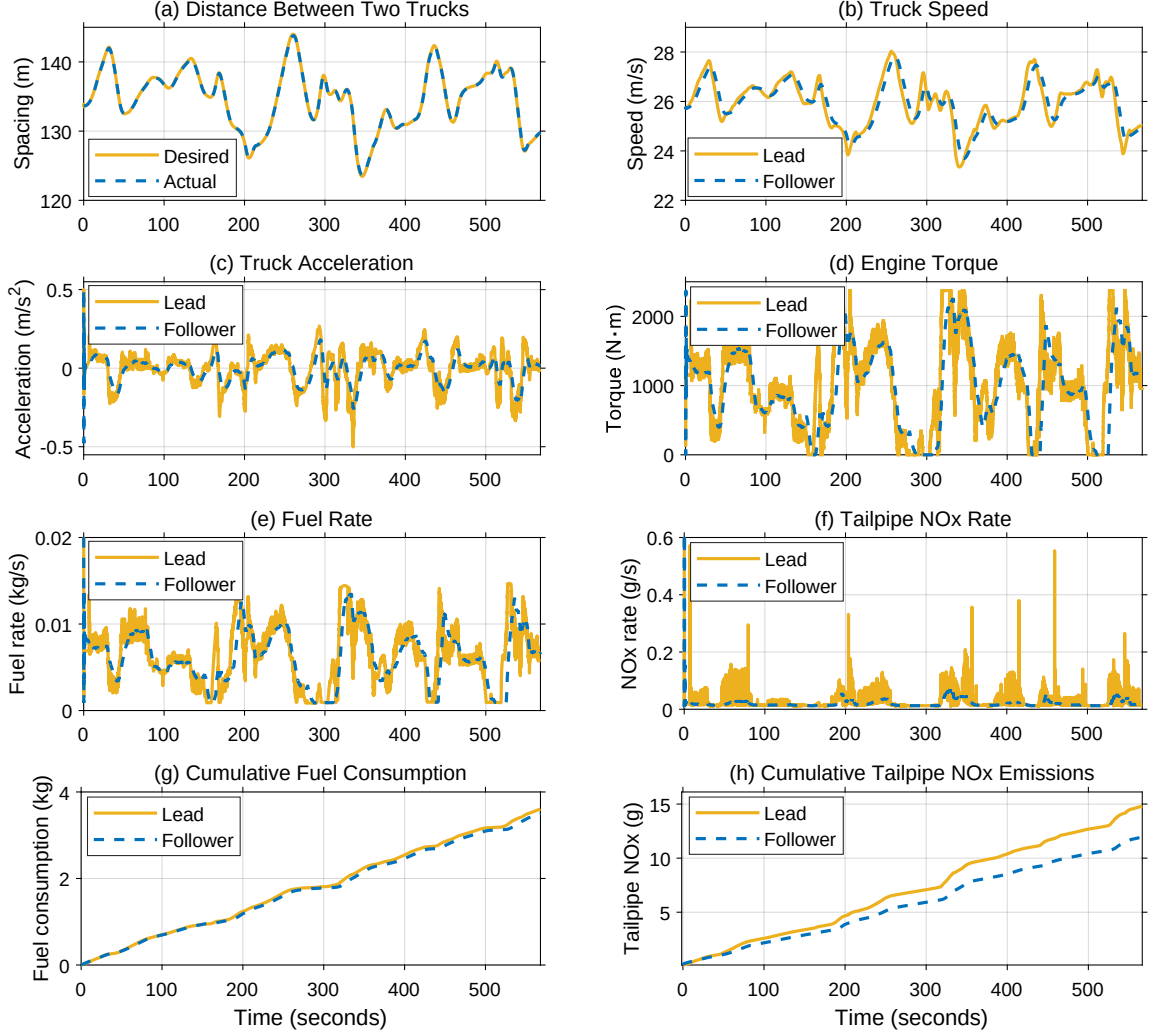


Figure 4.5: Results of 5-sec time gap truck platooning with the designed controller under Alberta Highway 2 driving cycle.

Figure 4.5(d) shows that engine torque fluctuations for the follower truck are significantly suppressed, indicating that the proposed controller effectively optimizes torque behavior, particularly under the demanding conditions of hilly terrain. This reduction in torque variability contributes directly to the lower tailpipe NO_x emis-

sions observed for the follower truck in Figure 4.5(f). To quantitatively assess this effect, we apply the previously defined smoothness index for engine torque, which captures the amplitude and frequency of torque fluctuations. This metric is especially relevant because NO_x emissions are highly sensitive to rapid engine torque changes, which lead to undesired combustion conditions and transient air-fuel imbalances in diesel engines [77]. During sudden torque increases, in-cylinder combustion temperatures spike due to increased fuel injection, which enhances thermal NO_x formation. Additionally, during these transients, the air supply from the turbocharger may lag, temporarily enriching the mixture and promoting further NO_x production. The EGR system also exhibits slower dynamic responses, reducing the effectiveness during rapid engine torque changes. As shown in Figure 4.6, rapid engine torque oscillations coincide with delayed EGR responses and corresponding spikes in tailpipe NO_x emissions. This mechanistic understanding is further supported by Figure 2.14, which illustrates a consistent positive correlation between engine torque variation rates and tailpipe NO_x emissions across varying engine load conditions. In this scenario, the lead truck's torque smoothness index is 47,136, while the follower truck's is dramatically lower at 1,462. This reduction confirms the controller's ability to suppress engine torque transients and reduce NO_x emissions accordingly.

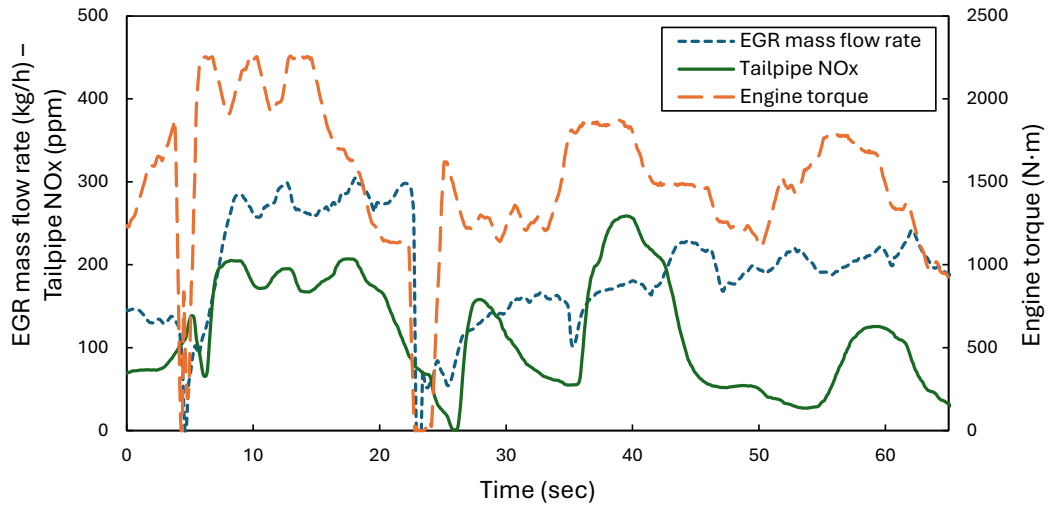


Figure 4.6: EGR system's dynamic response under transient engine torque.

As shown in Figure 4.5(g), under the Alberta Highway 2 driving cycle with hilly terrains, the follower truck achieves 3% fuel savings from platooning when the time gap is set to 5 sec. This is because the designed controller enables the follower truck to drive with the optimized engine torques. Moreover, the designed platoon controller facilitates the coordination between two trucks by eliminating unnecessary accelerations and decelerations, contributing to the reduction in fuel consumption. Figure 4.5(h) demonstrates that the follower truck experiences a 19.6% reduction in cumulative tailpipe NO_x emissions compared to that of the lead truck. This significant reduction is a result of the follower truck’s reduced engine torque fluctuations during the platooning operation. These results outperform those observed in field tests, where the follower truck operated under a commercial platooning system lacking optimum coordination between the powertrain and the high-level platoon controller. These findings highlight the effectiveness of the proposed control strategy in reducing both fuel consumption and emissions in real-world driving conditions. Notably, the reduction in NO_x emissions is more pronounced, reflecting the sensitivity of diesel engine NO_x formation to transient engine operations.

In addition to the 5-sec time gap scenario, time gaps of 2, 3, and 4 sec are also evaluated under the same Alberta Highway 2 driving cycle to assess controller performance across a broader operating envelope representative of legal following distances in many jurisdictions. The corresponding results are provided in Figures 4.7, 4.8, and 4.9.

Table 4.3 presents a comparison of key performance metrics between the proposed NMPC-based cooperative platoon controller and a PID-based non-cooperative controller across various time gaps (i.e., 2, 3, 4, and 5 sec). For both controllers, decreasing the time gap leads to an increase in maximum spacing error, as shorter gaps reduce the follower truck’s ability to closely track the lead truck’s trajectory. However, the spacing errors consistently remain within the 5-meter safety buffer, confirming that both controllers maintain safe inter-vehicle distances even under highway driving con-

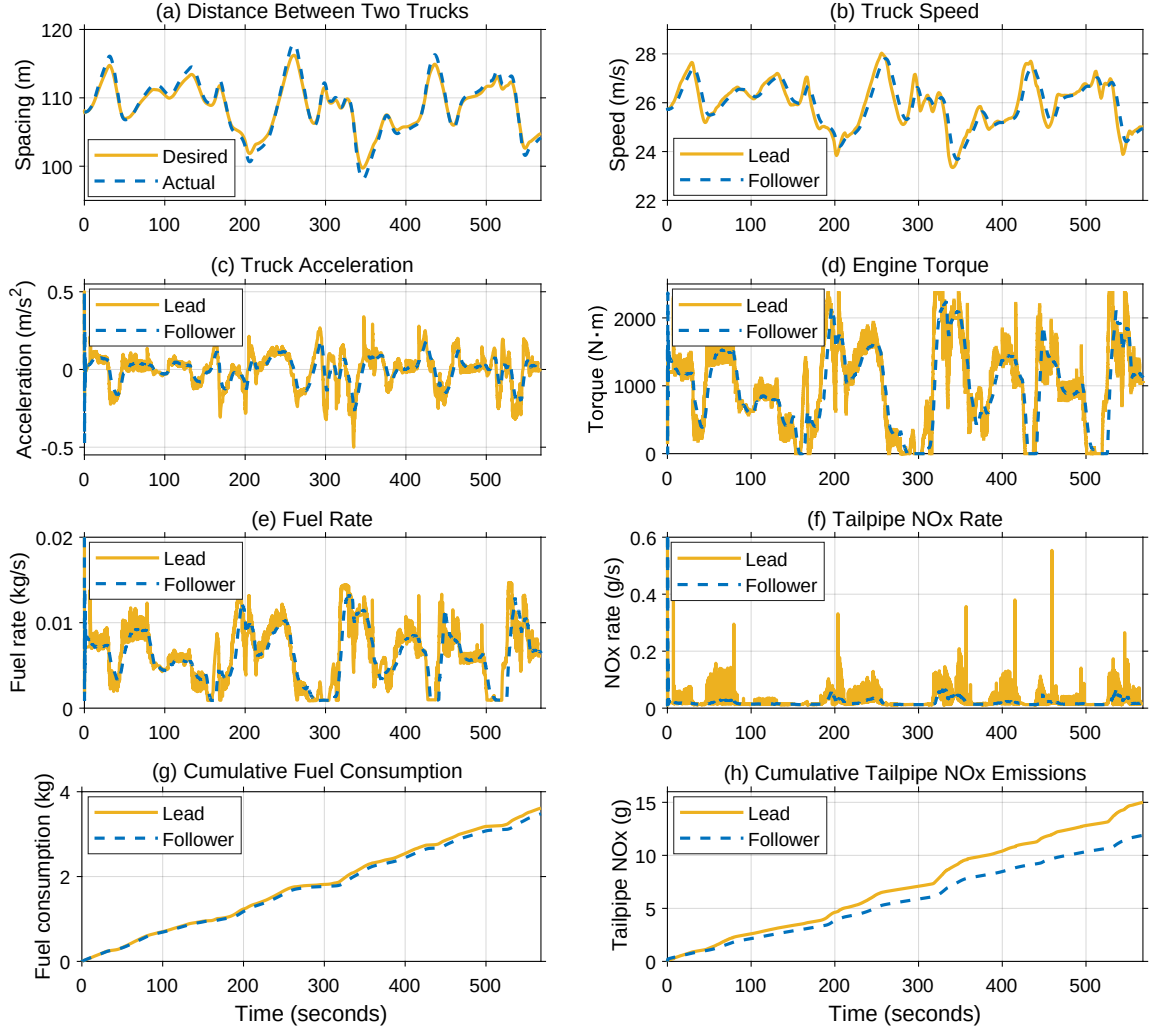


Figure 4.7: Results of 4-sec time gap truck platooning with the designed controller under Alberta Highway 2 driving cycle.

ditions with grade changes. Across all time gaps, the NMPC controller consistently achieves smoother speed profiles for the follower truck compared to the PID controller. This smoother behavior, combined with the benefit of reduced aerodynamic drag, contributes to improved fuel savings. Specifically, the NMPC controller yields fuel savings for the follower truck ranging from 3% to 6.6% as the time gap decreases from 5 to 2 sec, significantly outperforming the PID controller, which achieves only 0.3% to 2.2% fuel savings. A similar trend is observed in engine torque smoothness, where the NMPC controller results in substantially lower torque variation. This smoother torque behavior contributes to lower tailpipe NO_x emissions. Correspondingly, NO_x

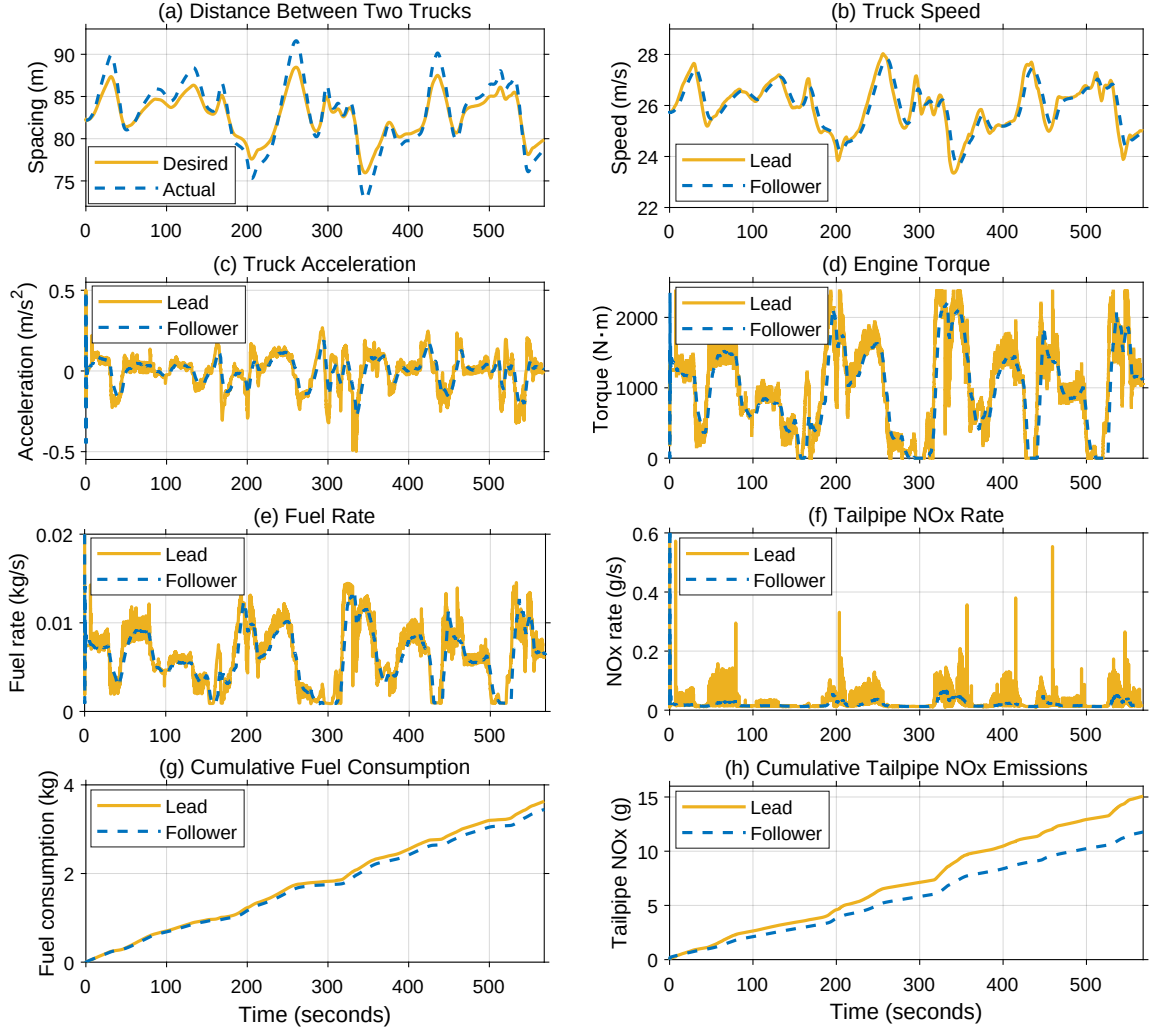


Figure 4.8: Results of 3-sec time gap truck platooning with the designed controller under Alberta Highway 2 driving cycle.

emission reductions using the NMPC controller range from 19.6% to 23.2%, whereas the PID controller achieves only 2.9% to 4.9% reductions over the same range of time gaps. These results indicate that while fuel savings decrease significantly beyond the 3-sec time gap, tailpipe NO_x emission reductions remain substantial. The underlying reason is because the mechanism leading to NO_x formation is different from that governing engine fuel consumption.

As platooning gaps decrease, the follower truck needs to respond more rapidly to changes in the lead truck, especially under sudden deceleration and acceleration. This leads to varying torque commands, increasing the engine torque change rates

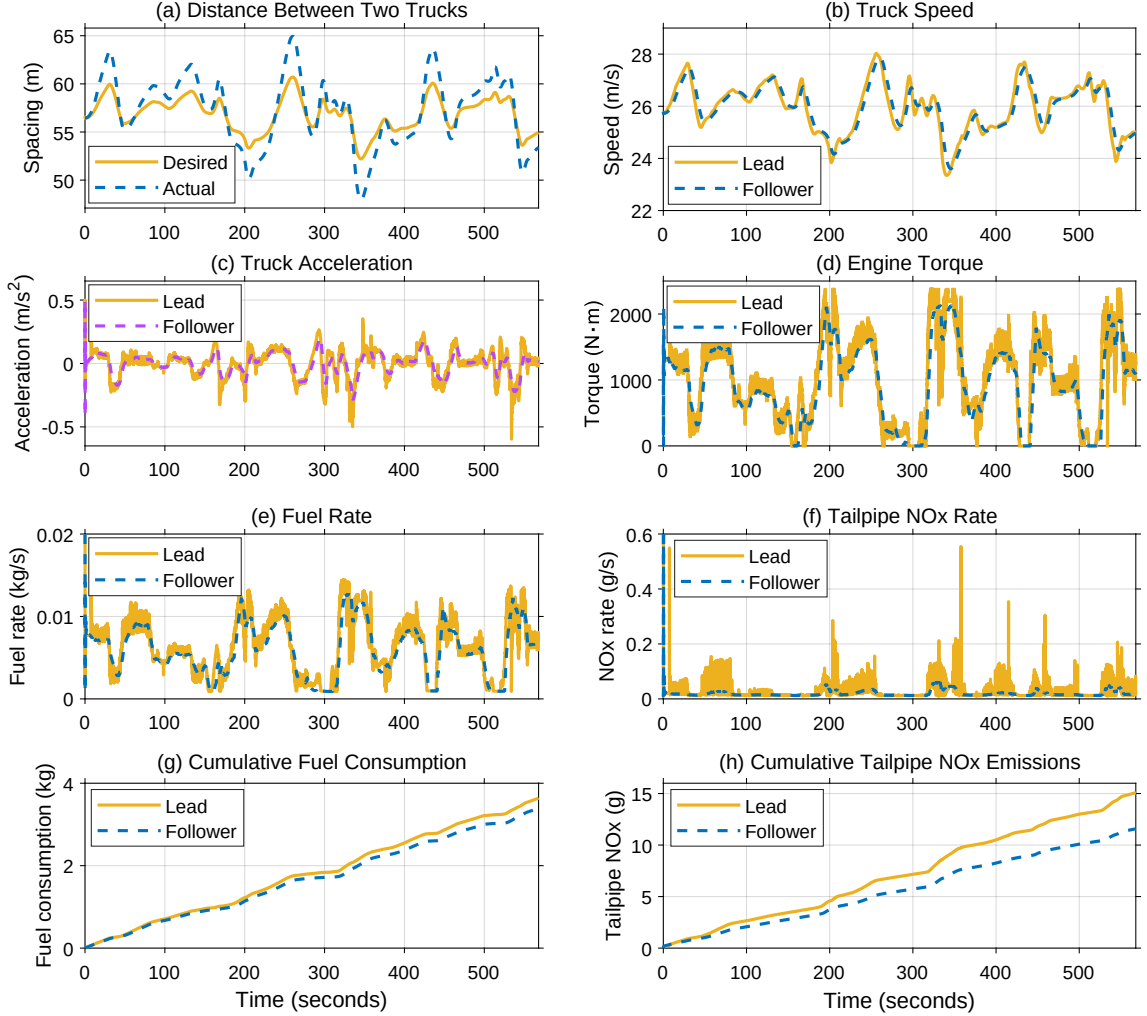


Figure 4.9: Results of 2-sec time gap truck platooning with the designed controller under Alberta Highway 2 driving cycle.

[53]. In contrast, larger time gaps allow for smoother transitions, leading to more stable engine operation with fewer abrupt torque changes. During engine transients, the EGR system has a known lag in response to torque demands, which disrupts the proper amount of EGR delivery into the combustion chamber [161]. At smaller time gaps, the rapid engine torque fluctuations overwhelm the EGR control loop, resulting in temporary lean conditions, reduced inert gas dilution, and higher in-cylinder temperatures—conditions that favor NO_x formation. However, larger time gaps reduce the rate of torque change, giving the EGR valve and turbocharger time to stabilize intake air composition, thereby maintaining low peak combustion temperatures and

Table 4.3: Comparison of key performance metrics between NMPC and PID truck platooning controllers among different time gaps.

Control Methodology	NMPC				PID			
Time Gap (sec)	2	3	4	5	2	3	4	5
Maximum Spacing Error (m)	4.3	3.1	1.7	0.3	4.1	3.2	1.8	0.5
Follower Speed Smoothness (-)	25.6	27.4	31.6	32.8	113.1	110.2	107.4	102.6
Follower Torque Smoothness (-)	1423	1434	1446	1462	16829	16610	16394	16182
Fuel Saving (%) ¹	6.6	5	3.8	3	2.2	1.6	1	0.3
NO _x Reduction (%) ²	23.2	21.9	20.7	19.6	2.9	3.6	4.1	4.9

¹ The fuel saving of the follower truck is calculated by (fuel consumption of lead truck - fuel consumption of follower truck)/fuel consumption of lead truck*100%.

² The NO_x reduction of the follower truck is calculated by (NO_x emissions of lead truck - NO_x emissions of follower truck)/NO_x emissions of lead truck*100%.

suppressing NO_x emissions. Torque transients at small gaps also disrupt boost pressure and intake air delivery, leading to mismatches between injected fuel and fresh air. This impairs in-cylinder mixing and results in thermal stratification, which enhances localized NO_x formation [162]. At larger time gaps, smoother engine load transitions improve air-path stability and promote homogeneous combustion, which suppresses NO_x production.

On the other hand, the fuel consumption is primarily affected by engine load that is determined by the vehicle control unit (VCU) and engine control unit (ECU) in response to the required traction torque to overcome vehicle resistant forces to meet the driver's power demands. At smaller platooning time gaps, the follower truck benefits more from reduced aerodynamic drag, which directly lowers engine load and consequently reduces fuel consumption [57]. However, as platooning gaps increase, these aerodynamic benefits diminish rapidly. Even with enhanced speed coordination strategies, the follower truck's fuel savings begin to plateau, since aerodynamic drag reduction becomes negligible at large spacing distances [1].

Accordingly, the effects of coordination and aerodynamic drag reduction are independently evaluated. Figure 4.10 compares fuel-saving contributions under the

Alberta Highway 2 driving cycle. The fuel savings attributable to aerodynamic drag reduction decrease from 5.5% to 0.6% as time gaps increase from 2 to 5 sec. In contrast, the fuel savings attributable to coordination increase from 1.1% to 2.4% over the same range. These results indicate that, under highway driving with grade changes, aerodynamic drag reduction remains a primary contributor to fuel savings at shorter gaps, whereas coordination between trucks becomes increasingly important as gaps grow.

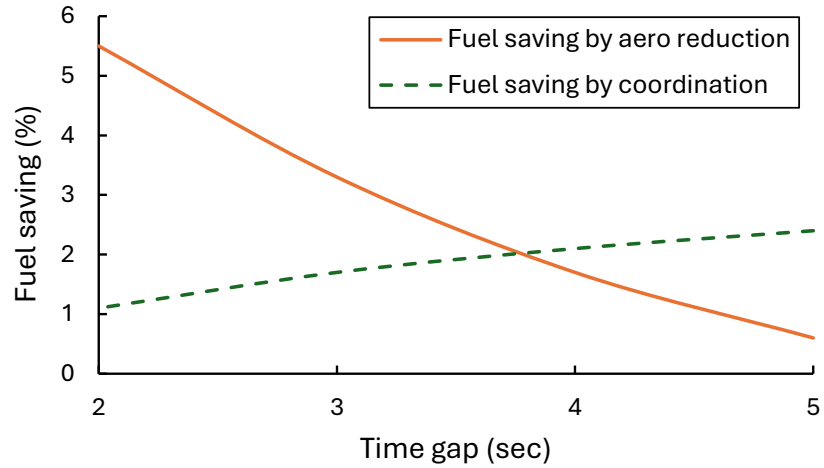


Figure 4.10: Fuel saving from coordination and aerodynamic drag reduction under Alberta Highway 2 driving cycles.

Finally, Figure 4.11 summarizes the key mechanisms underlying the observed trends and highlights the effectiveness of the designed cooperative platoon controller in simultaneously reducing fuel consumption and tailpipe NO_x emissions at regulation-compliant gaps while maintaining safe inter-vehicle spacing under varying road grades.

4.4 Summary of Chapter

This chapter designed and evaluated a truck platooning control system based on an innovative hierarchical framework, with the objectives of maintaining a safe following distance while improving fuel efficiency and reducing tailpipe NO_x emissions. The

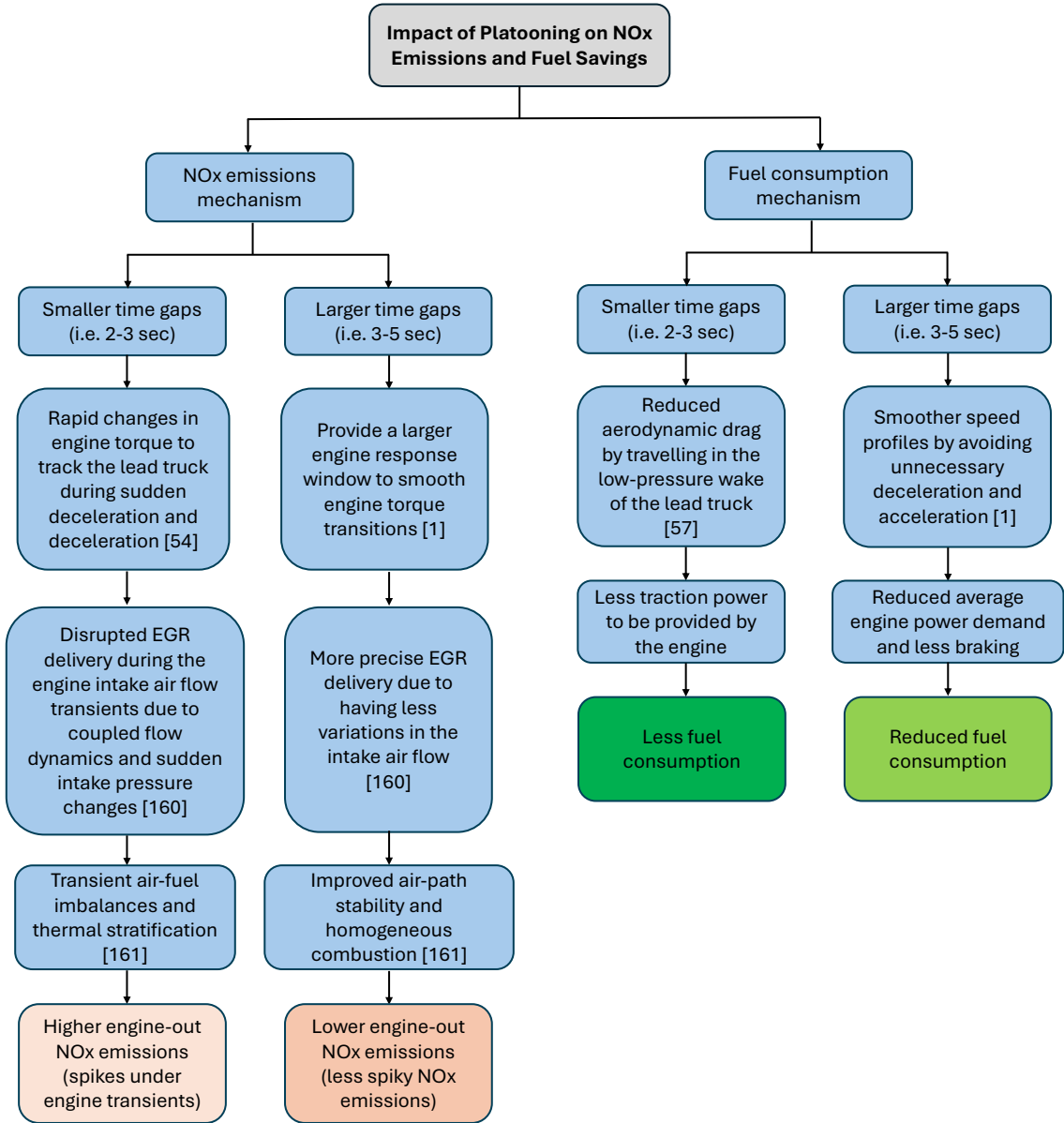


Figure 4.11: Impact of platooning on NO_x emissions and fuel savings.

proposed control system is particularly relevant for highways where small platooning time gaps (i.e., < 2 sec) are not permitted due to safety constraints (e.g., regulations, icy roads, and etc.). This chapter also uses real-world on-road platooning data for simulations. Real-world on-road platooning data were also used to construct the simulation environment and support controller assessment.

Using experimentally validated platoon models and measured road geometry from

the representative hilly terrains of Alberta Highway 2, numerical studies demonstrate that the proposed control system can recover meaningful platooning benefits even at large time gaps. The follower truck achieves fuel savings of up to 6.6% at a 2-sec time gap and maintains a 3% fuel saving at a 5-sec time gap. These results indicate that, beyond aerodynamic drag reduction, coordinated speed profiling plays a pivotal role in improving fuel economy by reducing unnecessary accelerations and decelerations under grade changes. In addition, the NMPC formulation explicitly suppresses abrupt engine torque changes, which are strongly associated with transient NO_x spikes, thereby providing a practical mechanism for emissions mitigation without embedding a highly nonlinear aftertreatment model directly into the real-time optimization. As a result, the follower truck achieves substantial tailpipe NO_x reductions across all tested time gaps, with a maximum reduction of 23.2% at a 2-sec time gap relative to the lead truck.

Overall, the proposed NMPC-based platooning control system demonstrates strong capability to enhance both fuel savings and tailpipe NO_x reduction for large-spacing truck platooning under real-world highway operation with varying road gradients, while maintaining regulation-compliant safety constraints.

Chapter 5

Truck Platooning in Windy Environments¹

One of the main factors in the real-world highway driving operation that can substantially degrade truck platooning performance is wind, particularly under regulation-compliant time gaps where aerodynamic coupling is weaker and powertrain-induced effects become more pronounced. Longitudinal winds—including headwinds and tailwinds—directly modulate the relative airspeed and therefore the aerodynamic drag, leading to measurable changes in tractive power demand and fuel consumption [71]. In addition to altering the mean resistive load, wind disturbances can induce speed fluctuations within the platoon, forcing trucks to apply additional traction/braking effort to maintain the desired speed and spacing, which further increases fuel use [2].

Figure 5.1 compares fuel consumption from on-road truck platooning trials with simulation results that exclude wind effects (no wind). Under windy conditions, the lead truck’s fuel consumption deviated from the baseline by -15% to 25%, while the follower truck exhibited comparable variations of -15% to 23%. These findings highlight the substantial impact of winds on platooning energy outcomes, emphasizing the necessity of incorporating wind effects into platooning control strategies for real-world deployment.

¹ This chapter is partially based on [4, 9].

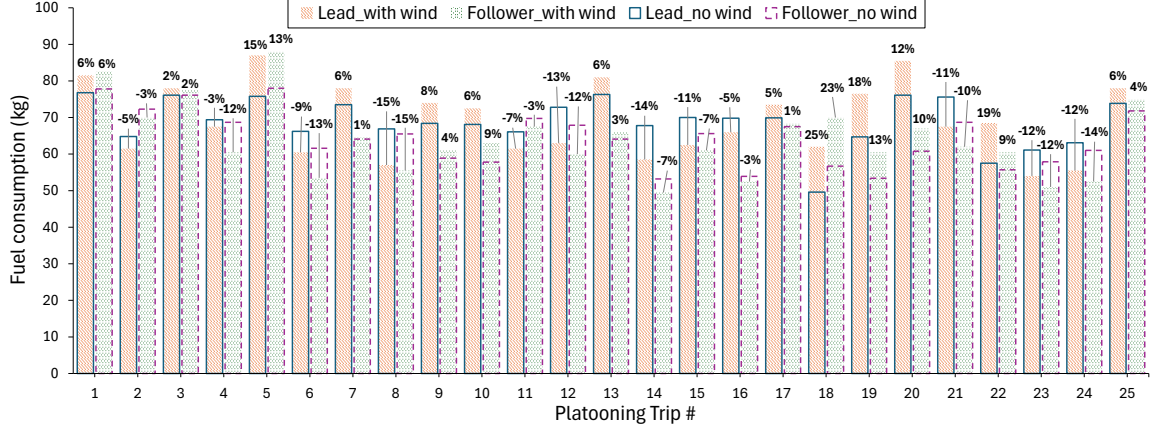


Figure 5.1: Fuel consumption comparison between calm and windy conditions along a route of identical length on Alberta Highway 2. The baseline fuel consumption under calm conditions was estimated through simulation, excluding wind effects. The reported ratios indicate the percentage increase or decrease in fuel consumption under windy conditions relative to the baseline.

Beyond energy impacts, wind disturbances can also exacerbate emissions. Data from the on-road CTPS trials indicate that unpredictable wind variations prompted frequent engine torque adjustments by the engine control unit, producing pronounced torque transients and corresponding spikes in tailpipe NO_x emissions. Motivated by these observations, this chapter investigates control strategies that improve fuel savings while simultaneously mitigating tailpipe NO_x under windy conditions by attenuating unnecessary speed and torque oscillations.

To this end, this chapter develops a novel hierarchical control system tailored for large-spacing truck platooning in windy environments. The proposed strategy enhances coordination between platoon trucks at separation distances corresponding to 3–5 sec time gaps, with the objective of smoothing speed and engine torque trajectories while maintaining regulation-compliant safety. Numerical evaluations are performed using a two-truck platoon simulation built upon experimentally validated models and on-road data, ensuring realism in vehicle dynamics and energy/emissions estimation. The resulting simulation outcomes are then analyzed and discussed to quantify the performance of the proposed approach under representative windy con-

ditions.

5.1 Wind Measurements Calibration

A mobile weather station was mounted on the top center of the lead truck to record real-time wind speed and wind direction. Figure 5.2 shows the schematic of wind vectors. The velocity of the apparent wind, v_a , is defined as the relative wind velocity experienced by a moving vehicle and is computed as the vector difference between the true wind velocity, v_w , and the truck velocity, v_t . Accordingly, the apparent wind speed and apparent wind angle β can be expressed as

$$\begin{cases} v_a = \sqrt{v_w^2 + v_t^2 + 2v_w v_t \cos \alpha} & (5.1a) \\ \beta = \arctan \left(\frac{v_w \sin \alpha}{v_t + v_w \cos \alpha} \right) & (5.1b) \end{cases}$$

where, α is the true wind angle.

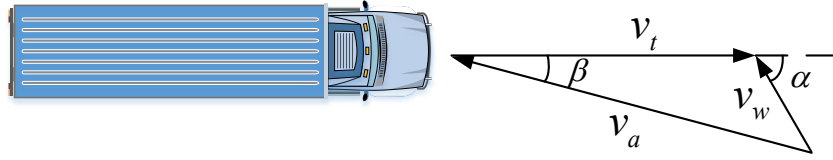


Figure 5.2: Schematic of the wind vectors.

Accurate wind measurements for on-road platooning studies require that the wind sensor be sufficiently elevated above the vehicle body to reduce distortion from the cab-induced boundary layer and separated flow regions. A prior study recommends installing the sensor at least 1 m above the roofline to approximate free-stream conditions [72]. However, due to the height restriction constraints on Alberta highways along the test routes, the weather station in the CTPS trials was installed only 0.3 m above the lead truck roof. Under this configuration, the measured wind signals are affected by vehicle-induced flow interference (e.g., acceleration of the local flow, turbulence, and yaw-dependent separation), and therefore do not directly represent free-stream apparent wind conditions. Consequently, a calibration procedure is required before using the measurements for analysis and simulation inputs.

To correct these installation-induced biases, the wind measurements were calibrated using Alberta’s Road Weather Information System (RWIS), which provides stationary wind speed and direction measurements at roadside weather stations. As shown in Figure 5.3, six RWIS stations (circled in blue) located along the CTPS test routes were selected for the calibration. These stations were chosen because they are distributed along the Calgary–Edmonton corridor, cover the major segments of the test route, and provide repeated encounters of the platoon with station locations across many trips, thereby improving the statistical representativeness of the calibration dataset.

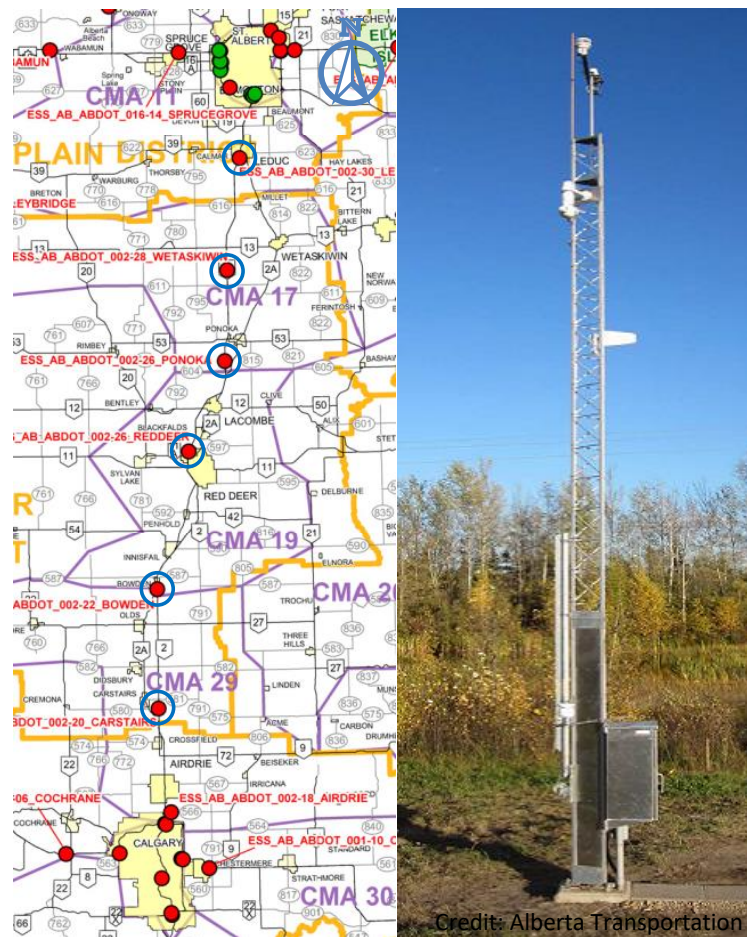


Figure 5.3: Site map of Alberta’s Road Weather Information System (RWIS) along the test routes. Red dots on the map represent the stationary weather stations, and the ones circled in blue were selected for the wind measurements calibration. Map and photo courtesy of Alberta Transportation.

For each RWIS encounter, the stationary wind speed and wind direction were treated as the reference true-wind inputs, and the truck speed recorded from onboard sensors was used to compute the corresponding true apparent wind speed and apparent wind angle through Eqs. (5.1a) and (5.1b). These computed values serve as the baseline apparent wind conditions that would be observed at the truck location in the absence of sensor-placement interference. The simultaneously recorded apparent wind speed and angle from the mobile weather station were then paired with the baseline values to establish a calibration mapping.

Given the need for a computationally efficient calibration that can be applied across the entire dataset, a linear regression model was adopted to map measured values $(v_{a,meas}, \beta_{meas})$ to baseline values $(v_{a,true}, \beta_{true})$. Figure 5.4 shows the calibration results. The strong agreement observed for apparent wind speed indicates that the mobile station captures the dominant variations in wind magnitude, whereas larger deviations are observed in the apparent wind angle due to the higher sensitivity of wind direction measurements to local turbulence and rapid yaw fluctuations. Overall, the coefficient of determination is $R^2 = 0.96$ for the apparent wind speed and $R^2 = 0.84$ for the apparent wind angle, indicating that the linear calibration explains the majority of the observed variance, particularly for wind speed.

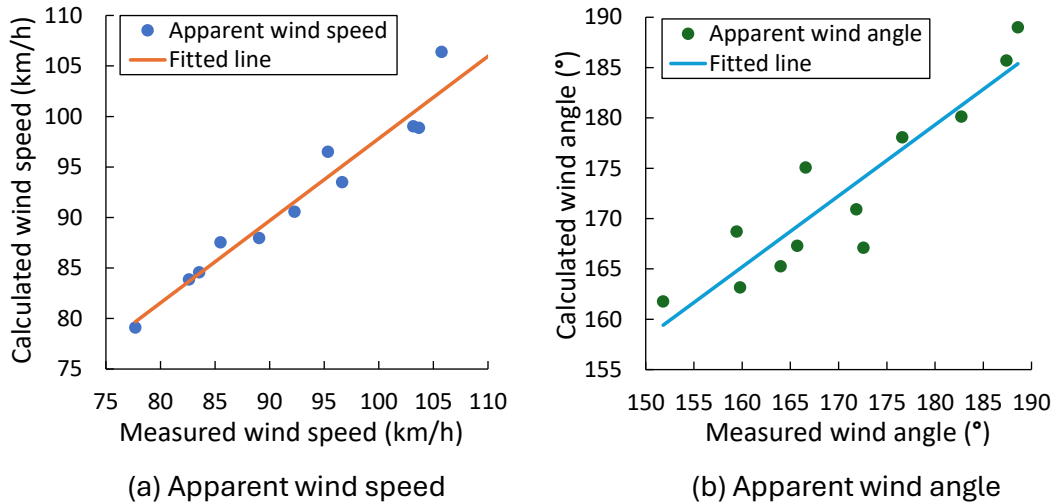


Figure 5.4: Calibration results of mobile weather station measurements.

The resulting calibrated relationships for the apparent wind speed and apparent wind angle under the measurement setup used in this thesis are expressed as

$$\begin{cases} v_{a,true} = 0.81v_{a,meas} + 16.46 & (5.2a) \\ \beta_{true} = 0.71\beta_{meas} + 52.23 & (5.2b) \end{cases}$$

where, $v_{a,true}$ and β_{true} denote true apparent wind speed and angle calculated based on road-side stationary weather-station measurements, respectively; $v_{a,meas}$ and β_{meas} represent measured apparent wind speed and angle from the mobile weather station, respectively.

After calibration, the corrected wind profiles ($v_{a,true}, \beta_{true}$) are used in this chapter to quantify wind impacts and to construct realistic wind inputs for simulation-based evaluations of the proposed control strategy. It is also noted that the wind measurements from the mobile weather station were not available to the CTPS controller during the field trials and therefore were not used for real-time control. As a result, the CTPS trials provide a suitable baseline representing typical commercial platoon-ing operation without explicit wind compensation, against which the performance benefits of the proposed wind-aware hierarchical control strategy can be evaluated.

5.2 Design of Platoon Controller

This chapter proposes a hierarchical control framework for a two-truck platoon, illustrated in Figure 5.5. The platoon controller determines the desired net longitudinal torque command for the follower truck based on real-time measurements from both the lead and follower vehicles and the prevailing wind conditions. The shared information includes vehicle position and speed, as well as calibrated apparent wind speed and direction obtained from the onboard weather station. These signals are exchanged through V2V communication. Communication delays are neglected in this chapter to focus on the core control design; nonetheless, delay-tolerant implementations can be incorporated using approaches such as the event-triggered framework in [163]. The lower-level controller adjusts engine and brake variables to track the

desired torque specified by the platoon controller. Under the setting of large spacings, the aerodynamic coupling between trucks becomes less significant, and the benefits of fuel saving and NO_x emissions reductions primarily depend on leader–follower coordination, which can be effectively represented using a two-truck configuration without compromising generality.

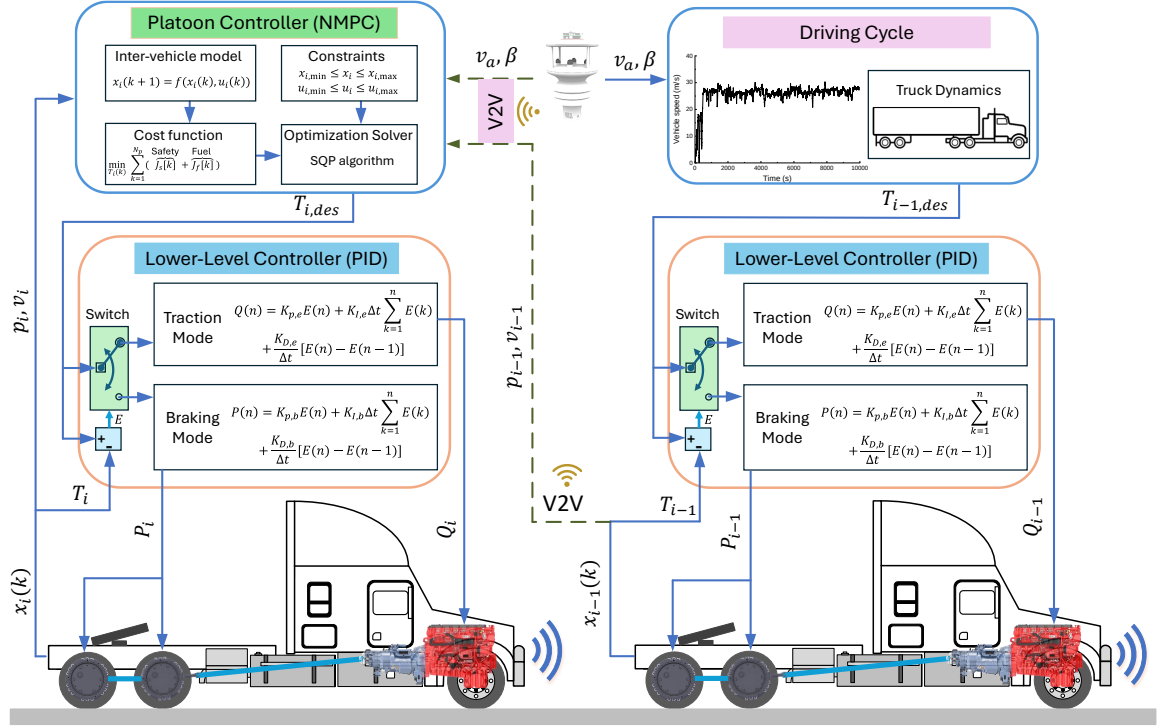


Figure 5.5: Control architecture of the designed truck platooning system.

5.2.1 Upper-Level Controller

The upper-level controller is designed to ensure platoon safety, improve fuel savings, and reduce tailpipe NO_x emissions in windy environments. These objectives are formulated as an NMPC optimization problem, which determines the optimal torque allocation strategy for the follower truck under defined constraints. The cost function

is defined as

$$\begin{aligned}
\min_{T_{des,i}(k)} J(k) = & \overbrace{\sum_{j=1}^{N_p} \sum_{i=1}^n \frac{w_1}{s_1} \|e_{di}(k+j|k)\|^2}^{\text{Platoon Safety}} \\
& + \overbrace{\sum_{j=1}^{N_p} \sum_{i=1}^n \frac{w_2}{s_2} \|v_i(k+j|k) - v_i(k+j-1|k)\|^2}^{\text{Fuel Consumption Reduction}} \\
& + \overbrace{\sum_{j=1}^{N_c-1} \sum_{i=1}^n \frac{w_3}{s_3} \|T_i(k+j|k) - T_i(k+j-1|k)\|^2}^{\text{Tailpipe NO}_x \text{ Emissions Reduction}}
\end{aligned} \tag{5.3}$$

where N_p represents the prediction horizon, N_c represents the control horizon, n denotes the number of follower trucks in a platoon, w is the tuning weight, and s is the scale factor.

The cost function in Eq. (5.3) consists of three weighted components. The first component maintains the desired separation distance from the lead truck in accordance with the adopted spacing policy. The second component promotes fuel savings by smoothing speed profiles, minimizing unnecessary decelerations and accelerations. The third component reduces tailpipe NO_x emissions by limiting rapid engine torque fluctuations, as these fluctuations are strongly correlated with increased tailpipe NO_x emissions.

The optimization problem is subject to system dynamics, as expressed in Eq. (5.4a). Furthermore, both state and input constraints are considered. To ensure safe and stable platoon operation under real-world disturbances, a safety buffer is reserved to avoid collisions and absorb any deviation of the follower truck from the desired separation distances, as defined in Eq. (5.4b). To ensure compliance with real-world traffic regulations, a speed constraint is enforced in Eq. (5.4c). Additionally, engine and brake torques are bounded by their respective capacity constraints,

as outlined in Eqs. (5.4d) and (5.4e).

$$\mathbf{x}_i(k+j+1 | k) = \mathbf{x}_i(k+j | k) + f_p(\mathbf{x}_i(k+j | k), \mathbf{u}_i(k+j | k))\Delta t \quad (5.4a)$$

$$e_{di}^{\min} \leq e_{di}(k+j | k) \leq e_{di}^{\max} \quad (5.4b)$$

$$v_i^{\min} \leq v_i(k+j | k) \leq v_i^{\max} \quad (5.4c)$$

$$0 \leq T_{ei}(k+j | k) \leq T_{ei}^{\max} \quad (5.4d)$$

$$0 \leq T_{bi}(k+j | k) \leq T_{bi}^{\max} \quad (5.4e)$$

Where $f_p(\cdot)$ denotes the nonlinear function; $e_{di}^{\min}$ and $e_{di}^{\max}$ are the minimum and maximum permissible inter-vehicle distance error, respectively; $v_i^{\min}$ and $v_i^{\max}$ denote the minimum and maximum speed bounds of the truck, respectively; $T_{ei}^{\max}$ and $T_{bi}^{\max}$ represent the maximum allowable engine torque and brake torque, respectively.

The control system is discretized with the sampling time, Δt , of 0.1 sec to accommodate fast vehicle dynamics. The prediction horizon, N_p , is selected as 50, providing a prediction window of 5 sec to cover the maximum time gap considered in this study. The control horizon, N_c , is set to 10 (20% of the prediction horizon) to enhance computational efficiency [159]. Tuning weights are chosen as $\omega_1 = 10$, $\omega_2 = 5$ and $\omega_3 = 3$ to balance the trade-offs between tracking state and input reference trajectories and managing actuator excitation. Furthermore, the NMPC optimization problem is solved using the sequential quadratic programming (SQP) method. At the current time step k , the optimal control input sequence ($T_{des,i}(k) = [T_{des,i}(k|k), \dots, T_{des,i}(k+N_p-1|k)]$) is computed by minimizing the cost function subject to constraints over the prediction horizon. Only the first element of the computed optimal control input sequence, $T_{des,i}(k|k)$, is applied to the system. At the next time step $k+1$, the optimization is resolved over a shifted horizon to incorporate updated information on the truck platoon state $\mathbf{x}_i(k+1)$. Again, only the first element of the computed sequence, $T_{des,i}(k+1|k+1)$, is applied to the system. This iterative process allows the controller to continuously adapt to varying wind conditions in real time.

5.2.2 Lower-Level Controller

The lower-level controller determines the appropriate fuel quantity or brake pressure commands to track the desired torque. When the desired torque is positive, the engine operates in traction mode. A PID controller is implemented to regulate the fuel quantity, Q , to achieve the desired torque.

$$Q(k) = K_{P,e}E(k) + K_{I,e}\Delta t \sum_{j=1}^k E(j) + \frac{K_{D,e}}{\Delta t}\Delta E(k) \quad (5.5)$$

where error E is the difference between the desired and actual torques, and $K_{P,e}$, $K_{I,e}$, and $K_{D,e}$ are the proportional, integral, and derivative control gains, respectively.

Conversely, when the desired torque is negative, the brake system engages to regulate the braking torque. In this case, a PID controller generates the brake pressure command, P , for torque tracking.

$$P(k) = K_{p,b}E(k) + K_{I,b}\Delta t \sum_{j=1}^k E(j) + \frac{K_{D,b}}{\Delta t}\Delta E(k) \quad (5.6)$$

where $K_{P,b}$, $K_{I,b}$, and $K_{D,b}$ are the proportional, integral, and derivative control gains for the braking system.

The lower-level controller runs at the same sampling rate as the NMPC and closes the loop using onboard measurements. The PID gains are tuned to ensure stable torque responses over the operating range encountered under windy conditions.

5.3 Numerical Evaluation and Discussion

5.3.1 Simulation Setup

The performance of the designed truck platooning control system is assessed through simulation studies. These simulations leveraged experimentally validated vehicle models and data obtained from the on-road cooperative truck platooning trials, ensuring alignment with real-world dynamics. The simulated platoon consists of two identical Class-8 heavy-duty trucks operating under the Alberta Highway 2 driving

cycle, integrating time-varying wind conditions. To evaluate the controller’s efficacy for truck platooning at large separation distances, time gaps between the trucks are set to 3-5 sec, complying with Alberta’s current traffic regulations. Additionally, two representative wind scenarios extracted from the field trials—tailwinds (trip 2) and headwinds (trip 3), as described in Table 5.1—are incorporated to examine the system’s response under different windy conditions.

Table 5.1: Follower truck’s fuel saving and tailpipe NO_x emissions reduction on flat road segments under wind conditions.

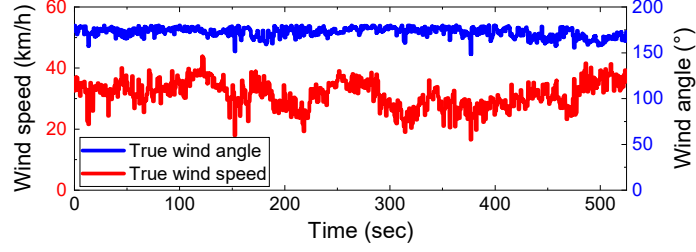
Trip #*	Average Wind Speed	Wind Direction	Fuel Saving	NO_x Reduction
1	3 km/h	Baseline	1.5%	-2.2%
2	31 km/h	Headwind	2.1%	5.5%
3	25 km/h	Tailwind	2.4%	-39.4%

* To eliminate the impacts of road slopes and traffic interactions, only platooning trips from flat road segments without cut-ins were selected.

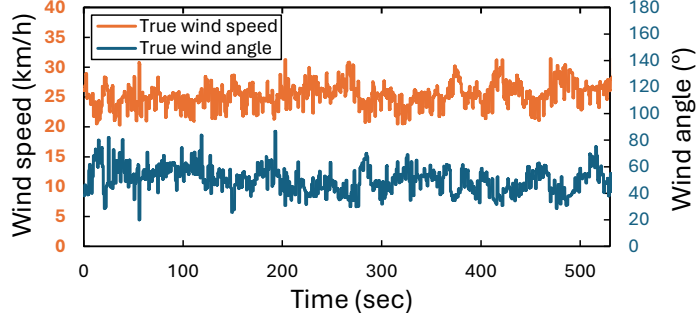
When platoon trucks encounter headwinds, the wind opposes their direction of travel, increasing aerodynamic resistance. Figure 5.6(a) shows the speed and direction of these headwinds during trip 2, with wind speeds ranging from 17 to 44 km/h and directions varying between 148° and 180°. Conversely, when trucks operate in a platoon under tailwinds, the wind aligns with their direction of travel, providing supportive aerodynamic effects. Figure 5.6(b) illustrates the speed and direction of tailwinds encountered during trip 3, where the wind speed ranges from 20 to 32 km/h, and the wind direction varies from 20° to 87°.

5.3.2 Simulation Results and Analyses

Because the platooning time gaps considered in this chapter are relatively large (≥ 3 sec), the aerodynamic influence of the follower on the lead truck is negligible [34]. Therefore, the lead truck is treated as the baseline for comparison, and the follower



(a) Headwind measurements in trip 2.



(b) Tailwind measurements in trip 3.

Figure 5.6: The measured speed and direction of headwinds and tailwinds correspond to trips 2 and 3, as listed in Table 5.1.

truck is controlled using the proposed hierarchical strategy. This subsection first reports representative results at a 4-sec time gap (recommended on Alberta Highway 2 [137]) under headwind and tailwind conditions. Subsequently, performance comparisons across time gaps of 3, 4, and 5 sec are provided.

4-sec Time Gap Platooning under Headwinds

Figure 5.7 presents the performance results for truck platooning at a 4-sec time gap under Alberta Highway 2 driving cycle with headwinds. Figure 5.7(a) shows that the designed control system enables the follower truck to maintain a safe following distance from the lead truck. The spacing error is quantified in Figure 5.7(b), where the maximum spacing deviation is 1.1 m. This value remains bounded within the safety buffer of 5 m, confirming the control system's capability to ensure safe and stable platooning under headwinds. Figure 5.7(c) illustrates the impact of headwinds on vehicle speed. While headwinds lead to fluctuations in the speed profile of the lead truck, the follower truck shows a considerably smoother speed trajectory. This

smoother profile is attributed to the coordination between trucks via V2V communication. Furthermore, the variations in engine torque for both trucks are displayed in Figure 5.7(d). The lead truck shows substantial fluctuations in engine torque due to the headwind disturbances, while the follower truck achieves a more stable torque output, thanks to the control system's performance.

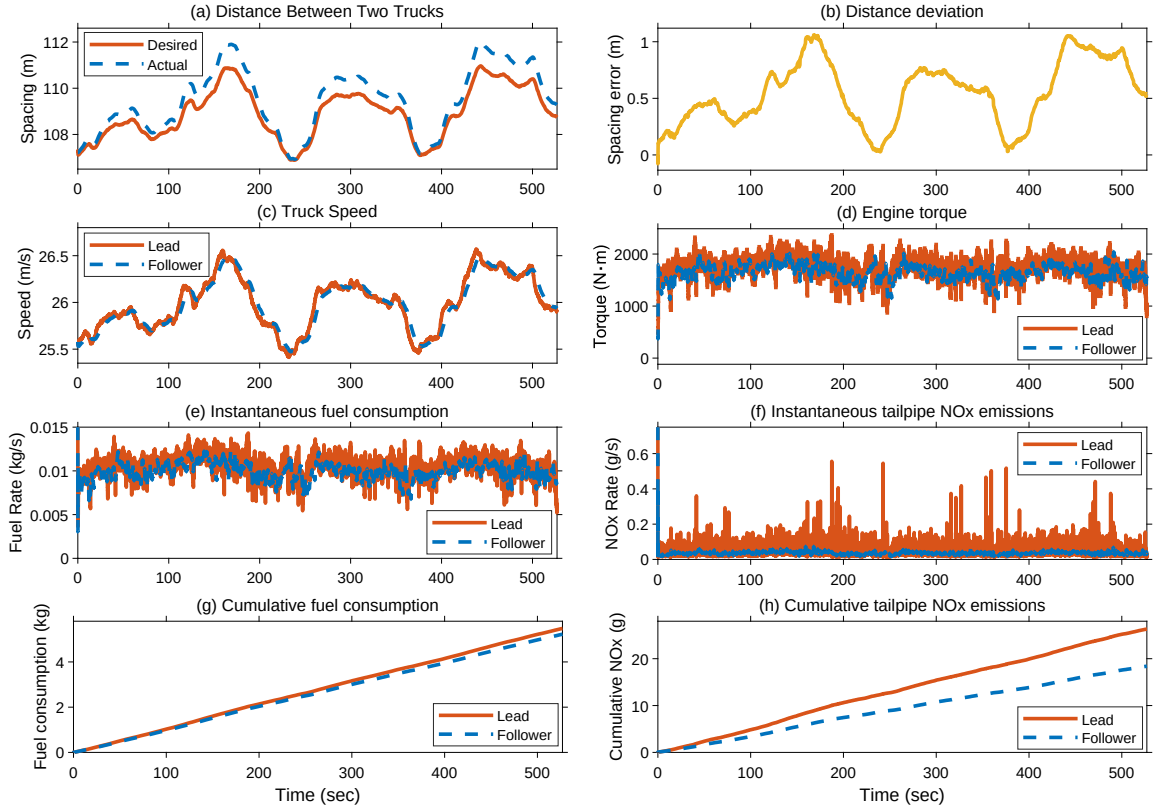


Figure 5.7: Results of 4-sec time gap truck platooning with the designed controller under headwind conditions, as depicted in Figure 5.6(a).

Enhanced coordination between trucks helps to avoid unnecessary decelerations and accelerations, thus having the potential for lowering fuel consumption [67]. Figure 5.7(e) compares the instantaneous fuel consumption rates of both trucks, showing an average rate of 10.4 g/s for the lead truck and 9.9 g/s for the follower. To further compare the fuel conversion efficiencies for trucks under large-spacing truck platooning with the designed controller on the public highway under windy conditions, we examined the distribution of engine operating points on the BSFC map. As shown in

Figure 5.8, the green-shaded region on the BSFC map represents the most efficient zone for the Cummins X-15 diesel engine used in this study, which is less than 190 g/kWh. The results indicate a notable shift of the follower truck's engine operating points toward this optimal zone, relative to those of the lead truck. Specifically, only 12.4% of the lead truck's engine operating points fall within the most efficient zone, while 17.7% of the follower truck's are concentrated in this area. This improved efficiency translates to a 4.6% fuel savings for the follower truck when platooning under headwind conditions, as illustrated in Figure 5.7(g). Compared to trip 2 in the cooperative truck platooning trial, the follower truck achieves an additional 2.5% increase in fuel savings. This underscores the designed control system's effectiveness in smoothing the follower truck's speed profile and mitigating the impact of wind disturbances.

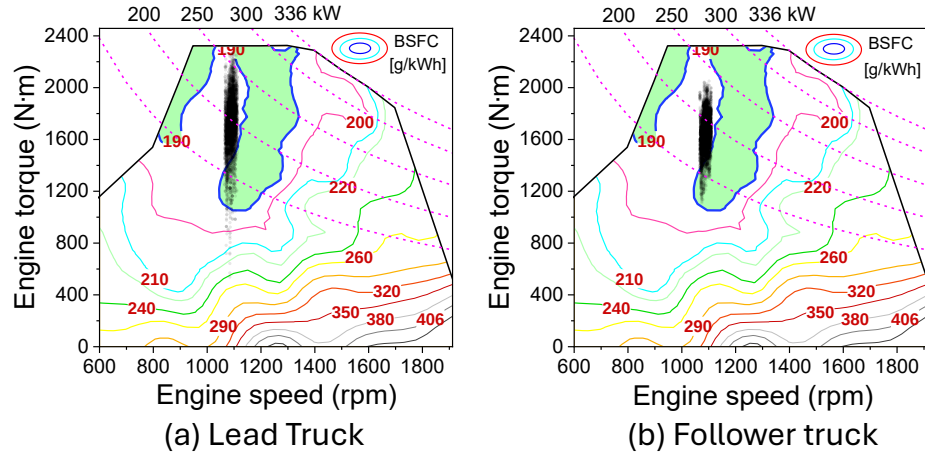


Figure 5.8: Comparison of operating points in BSFC map between lead and follower trucks under windy conditions. Trucks' engine operating points fall within the most efficient zone (green-shaded area): 12.4% of the lead truck vs. 17.7% of the follower truck.

Figure 5.7(f) shows that the instantaneous tailpipe NO_x emissions rate of the follower truck is consistently lower than that of the lead truck. Correspondingly, the follower truck achieves a 30% reduction in NO_x emissions at the 4-sec time gap under headwind conditions, as shown in Figure 5.7(h). This reduction is attributed to the designed controller's ability to optimize engine torque response, thus reducing the

rapid rate of torque changes that typically contribute to higher tailpipe NO_x emissions. When compared to the results of trip 2, the reduction in NO_x emissions for the follower truck improves by 24.5%, underscoring the controller’s effectiveness in reducing tailpipe NO_x emissions during truck platooning in the presence of headwinds.

All simulations were carried out on a PC equipped with an Intel® Core™ i7-9700 CPU @ 3.00 GHz and 32 GB RAM. The average solver computation time for the proposed NMPC was 13.1 ms per 100 ms sampling interval, with a maximum of 31.6 ms. The total computation time for the 530-sec test trip was 71.2 sec, demonstrating the real-time feasibility of the proposed control system.

4-sec Time Gap Platooning under Tailwinds

Figure 5.9 presents the results for truck platooning with a 4-sec time gap under the Alberta Highway 2 driving cycle in tailwind conditions. The control system effectively maintains the desired spacing between the follower and lead trucks, as illustrated in Figure 5.9(a). Figure 5.9(b) shows the maximum spacing error is limited to 0.7 m, which remains well within the 5-meter safety buffer. This consistency demonstrates the controller’s reliability in ensuring safe platooning with the influence of tailwinds.

Figure 5.9(c) illustrates that the follower truck experiences fewer speed fluctuations than the lead truck. Correspondingly, Figure 5.9(e) compares the instantaneous fuel consumption rates of both trucks, indicating that the follower truck exhibits a lower average fuel rate. This improved fuel conversion efficiency results from the follower truck’s smoother speed profile, ultimately yielding a 3.1% fuel savings, as shown in Figure 5.9(g). This improvement stems from the controller’s ability to regulate speed, minimizing fuel-intensive deceleration and acceleration cycles.

Engine torque variations are displayed in Figure 5.9(d), where the follower truck exhibits significantly reduced torque fluctuations compared to the lead truck. This torque smoothing leads to lower instantaneous tailpipe NO_x emissions rate (Figure 5.9(f)) and contribute to a 5% reduction in cumulative tailpipe NO_x emissions (Figure

5.9(h)). Compared to trip 3 in the cooperative truck platooning trial in Table 5.1, the follower truck achieves a substantial 44.4% improvement (from -39.4% to 5%) in tailpipe NO_x emissions reduction. This marked enhancement can be attributed to the controller's effective optimization of engine torque, which mitigates the impact of tailwinds on engine load and reduces the need for frequent torque adjustments.

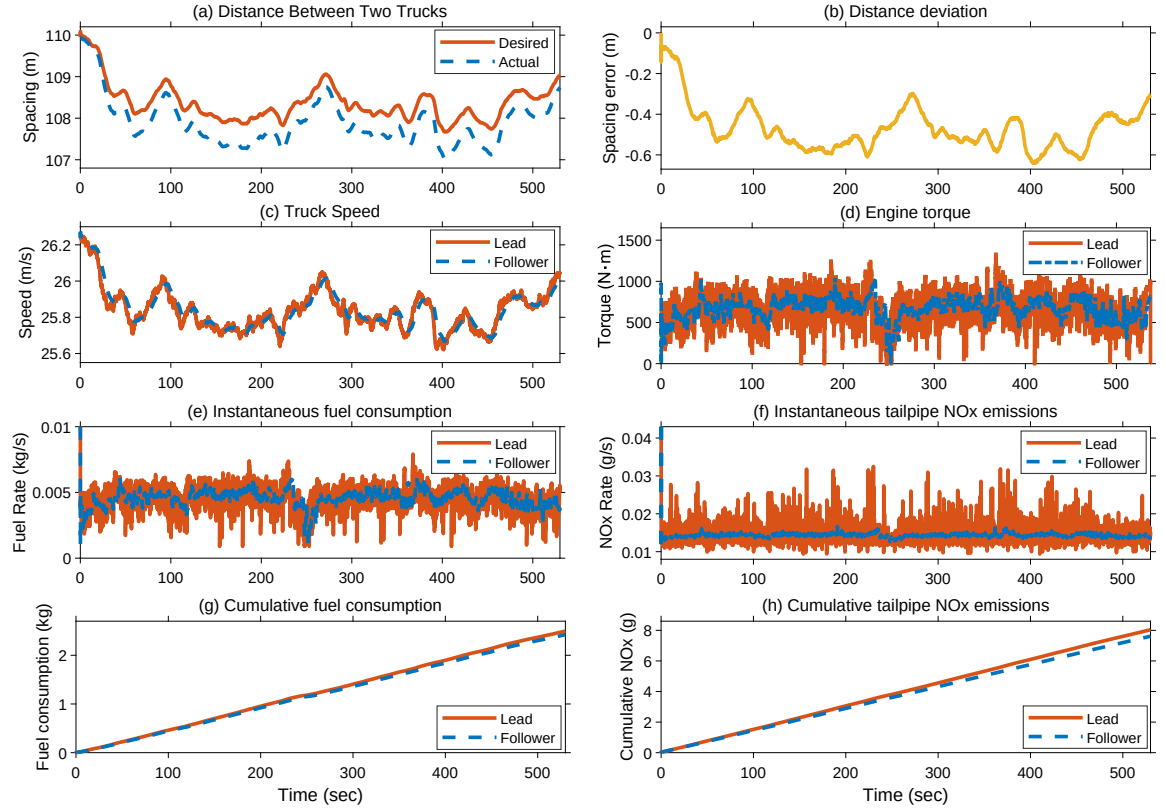


Figure 5.9: Results of 4-sec time gap truck platooning using the designed controller under tailwind conditions, as depicted in Figure 5.6(b).

Performance Comparison

Table 5.2 provides a comparative analysis of key performance for the designed platoon controller at different time gaps (3, 4, and 5 sec) under windy conditions.

In headwind scenarios, the maximum spacing error between the lead and follower trucks reaches 2 m at a 3-sec time gap. As the time gap increases, the maximum spacing error consistently decreases, reaching only 0.2 m at a 5-sec time gap. A

Table 5.2: Key performance comparison between headwinds and tailwinds truck platooning at different time gaps.

Performance	Headwinds			Tailwinds		
	3-sec	4-sec	5-sec	3-sec	4-sec	5-sec
Max Spacing Error (m)	2	1.1	0.2	1.2	0.6	0.2
Fuel Saving (%)	5.5	4.6	3.6	3.9	3.1	2.8
NO _x Reduction (%)	32	30	28.8	5.1	5	4.9

similar trend is observed for large-spacing truck platooning under tailwinds, though with generally smaller maximum spacing errors across all time gaps. Importantly, all observed spacing errors remain well within the 5-meter safety buffer, reflecting the control system’s reliability in maintaining safe following distances regardless of time gaps and wind conditions. The reduction in spacing error with larger time gaps is due to the increased separation distance, allowing the follower truck to more accurately track the lead truck.

Fuel savings show a decreasing trend as time gaps increase, with the follower truck achieving the highest fuel savings of 5.5% at the 3-sec time gap under headwind conditions. This improvement is due to both the coordination between trucks in headwinds and the partial shielding effect of the lead truck, which reduces aerodynamic drag and consequently enhance fuel saving for the follower truck. Conversely, under tailwind conditions, the tailwinds exert a direct influence on the follower truck, leading to relatively more speed fluctuations and, as a result, lowering fuel savings compared to headwind conditions.

The designed control system supports a consistent reduction in tailpipe NO_x emissions for the follower truck by smoothing engine torque variations under headwind conditions. Furthermore, under tailwinds, the follower truck achieves an average reduction in tailpipe NO_x emissions of 5% across time gaps of 3 to 5 sec. This reduction significantly outperforms the results in the cooperative truck platooning trials, highlighting the control system’s effectiveness in minimizing tailpipe NO_x emissions by

stabilizing engine torque responses under windy conditions.

These findings demonstrate the designed controller’s ability to maintain safe platooning distances and improve fuel savings under windy conditions while significantly reducing tailpipe NO_x emissions. The control system’s capacity to stabilize both vehicle speed and engine torque under varying windy conditions highlights its potential for promoting fuel-efficient and environmentally responsible truck platooning in real-world scenarios.

5.4 Summary of Chapter

This chapter designed a novel control system for large-spacing truck platooning under windy conditions, with the objectives of ensuring platoon safety, increasing fuel savings, and reducing tailpipe NO_x emissions. The proposed control system adopted a hierarchical framework comprising an upper-level controller and a lower-level controller. The upper-level controller was developed using the NMPC method to determine the optimal engine torque allocation for the follower truck under wind disturbances, while the lower-level controller was implemented using the PID method to track the desired torque and achieve effective real-time control.

By coordinating the platoon trucks, the designed control system prevented unnecessary decelerations and accelerations and, through optimized torque response, minimized aggressive engine torque changes that are strongly associated with transient tailpipe NO_x emissions. Numerical evaluations were conducted using a two-truck platoon simulated at 3–5 sec time gaps under varying windy conditions. The results demonstrated notable improvements, with fuel savings of up to 5.5% and tailpipe NO_x reductions of up to 32% for the follower truck. Compared to on-road truck platooning trials using Cooperative Truck Platooning Systems (without explicitly considering winds), the maximum fuel savings increased by 2.5 %, while the tailpipe NO_x emissions reduction improved by up to 44.4%. Moreover, the 3-sec time gap scenario exhibited the greatest potential for simultaneously reducing fuel consump-

tion and tailpipe NO_x emissions. Overall, these findings underscored the effectiveness of the designed control system in enhancing fuel savings and reducing tailpipe NO_x emissions under windy highway operation, contributing to safer and more sustainable freight transportation.

Chapter 6

Integrated Longitudinal and Lateral Controls¹

Truck platooning on public highways is exposed to coupled disturbances that act simultaneously in the longitudinal and lateral directions. Road-grade changes primarily excite longitudinal dynamics, whereas crosswinds and imperfect lane-centering performance can introduce persistent lateral deviations. As a consequence, a control strategy that treats longitudinal following and lateral tracking as largely independent problems may remain safe, yet still deliver suboptimal fuel and emission outcomes because lateral offsets can degrade wake utilization and because longitudinal speed corrections can amplify torque oscillations.

This chapter addresses these challenges by developing an integrated longitudinal–lateral platooning framework that explicitly links lateral alignment to aerodynamic benefit and connects longitudinal coordination to fuel and tailpipe NO_x behavior. First, the chapter analyzes on-road CTPS trials data to reveal how spacing errors and lateral offsets vary with fuel savings and NO_x outcomes under windy conditions, thereby motivating the need for coordinated control. Next, an innovative coordinated longitudinal–lateral control system is developed. The longitudinal controller regulates inter-vehicle spacing while optimizing the powertrain response to attenuate unnecessary engine torque fluctuations and generate smooth speed trajectories,

¹ This chapter is partially based on [5, 10].

thereby improving fuel conversion efficiency and reducing tailpipe NO_x emissions. In parallel, the lateral controller enforces platoon lateral safety and actively suppresses lateral offsets to preserve wake alignment and maintain the aerodynamic benefits of platooning.

Finally, the integrated controller is evaluated in high-fidelity simulations using experimentally validated Class-8 truck platoon models and representative Alberta Highway 2 driving conditions that include measured wind profiles and varying road grades. The results demonstrate how coordinated longitudinal–lateral control can simultaneously improve safety margins, recover fuel-saving potential, and reduce tailpipe NO_x emissions under real-world operating constraints.

6.1 Impact of Longitudinal Spacing and Lateral Offset on Platooning Performance

This section investigates how longitudinal spacing regulation and lateral alignment influence the energy, emissions, and safety performance of truck platooning under real-world windy conditions, using test data from the on-road CTPS trials. Table 6.1 summarizes the measured longitudinal spacing errors and lateral offsets between the two trucks, together with the follower truck’s fuel savings and tailpipe NO_x reduction at regulation-compliant time gaps of 3–5 sec. To exclude the effects of traffic interventions, the results are presented for platooning segments without any traffic cut-ins.

While no collisions occurred throughout the truck platooning trials, the longitudinal spacing errors under relatively high wind conditions consistently exceeded the pre-set buffer of 10 m. Such large deviations imply degraded spacing regulation and potentially unstable platoon behavior, which can raise safety concerns under sustained disturbances. The lateral offsets ranged from 0.41 m to 0.67 m, with an average of 0.52 m across all tested time gaps. Notably, larger lateral offsets were observed under high wind conditions compared to light winds. These longitudinal spacing errors and

Table 6.1: Summary of truck platooning performance under longitudinal spacing errors and lateral offsets.

Trip #	Time Gap (sec)	Spacing Error (m)	Lateral Offset (m)	Fuel Saving (%)	NO_x Reduction (%)*
1	3	9.8	0.41	2.6	-6.3
2	3	19.1	0.54	2.1	-19.2
3	4	9.1	0.44	2.2	-7.1
4	4	17.8	0.56	1.6	-25.2
5	5	8.2	0.47	1.1	-6.7
6	5	15.7	0.67	0.3	-32.3

* NO_x reduction with a minus sign indicates an increase in NO_x emissions.

lateral misalignments were primarily attributed to the CTPS’s lack of wind compensation in its control logic. Furthermore, the CTPS supported only lane keeping and did not explicitly manage lateral alignment between trucks, further contributing to the observed offsets.

The follower truck achieved fuel savings across all tested time gaps and wind conditions; however, a decline in fuel savings was observed as wind speed increased. Stronger headwinds directly increased the truck’s aerodynamic drag, which altered the powertrain and braking responses required to track lead truck’s speed trajectories, thereby affecting overall fuel conversion efficiency. In addition, crosswind-induced lateral offset further diminished fuel savings by disrupting the aerodynamic alignment between trucks. The follower truck also emitted more tailpipe NO_x emissions than the lead truck at all time gaps, with a rising trend in NO_x emissions observed under stronger windy conditions. This is because the inadequate coordination between CTPS and vehicle powertrain control system under wind disturbances amplified engine load fluctuations, leading to increased tailpipe NO_x emissions [9].

These findings confirm the feasibility of implementing commercial truck platooning on public roads and demonstrate its potential for fuel savings. However, they also highlight the need for coordinated control strategies that consider both longitudinal

and lateral motions to maximize fuel conversion efficiency while reducing tailpipe NO_x emissions under real-world highway driving environments.

6.2 Design of Integrated Longitudinal–Lateral Platooning Control System

The coupling between lateral and longitudinal vehicle dynamics can materially affect truck-platooning performance on curved roads at highway speeds [164]. In particular, the lateral dynamics are parameterized by the longitudinal velocity, which varies with spacing-control actions and external disturbances (e.g., wind and road). Conversely, lateral misalignment alters the aerodynamic interaction between the platoon trucks, thereby changing the follower’s drag force and the longitudinal traction/braking demand. The lack of integration between the longitudinal and lateral controllers induces concurrent fluctuations in inter-vehicle spacings and lateral offsets. Such variations not only degrade achievable fuel savings but also amplify powertrain transients, leading to oscillatory engine-torque commands that can increase NO_x emissions [3]. Therefore, this chapter proposes a three-layer hierarchical control framework that integrates longitudinal and lateral motions to enable safe truck platooning in real-world environments while optimizing fuel savings and reducing tailpipe NO_x emissions, as illustrated in Figure 6.1.

The upper layer of follower truck i detects environmental factors such as wind and road profiles, and provides necessary information to generate reference trajectory for the platoon truck. The medium layer receives this information together with the lead truck’s broadcast states (e.g., position and speed) via V2V communication, and performs coordinated longitudinal and lateral control to ensure safe platooning while enhancing fuel saving and reducing tailpipe NO_x emissions. The longitudinal controller is designed to maintain the desired inter-vehicle spacing by compensating for environmental disturbances, thereby ensuring safety and preserving aerodynamic benefits. In parallel, the lateral controller is designed by considering the changes of

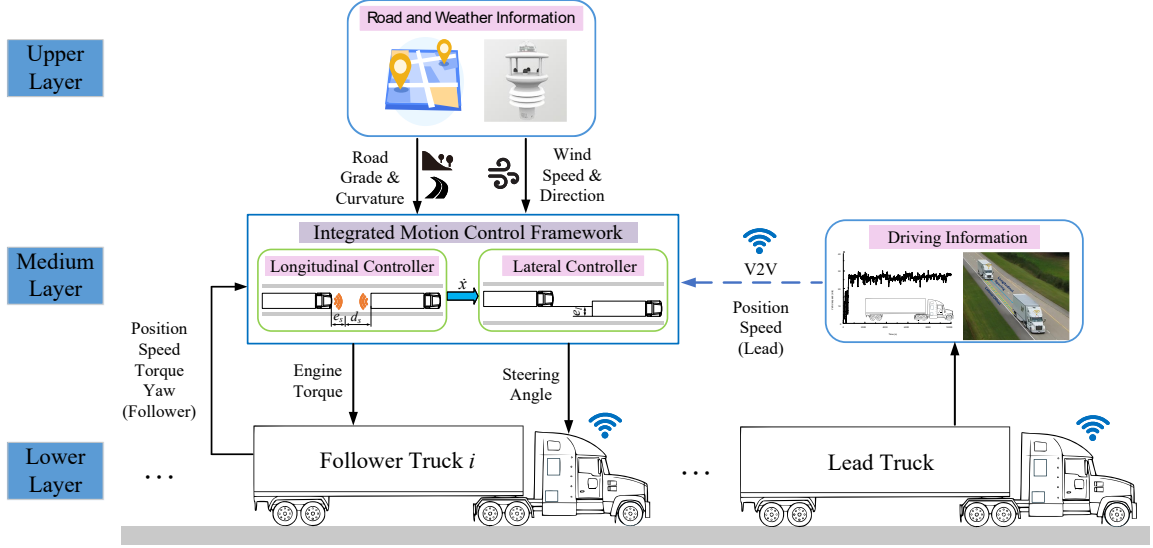


Figure 6.1: Hierarchical 3-layer control framework designed for the integrated longitudinal and lateral truck platooning system.

longitudinal velocity to regulates truck alignment, ensuring lane keeping and minimizing lateral offsets that would otherwise erode the wake benefit in real-world driving conditions. The medium layer translates the reference trajectory into actionable control inputs, which are then executed by the lower layer through the engine, braking, and steering systems. In this chapter, the focus is placed on the middle-layer design of the coordinated longitudinal and lateral controllers.

6.2.1 Longitudinal Platoon Controller

The longitudinal controller acquires state information from the upper-layer perception module and generates control commands that adjust engine or braking torque, enabling the follower truck to accelerate or decelerate as required to achieve the desired objectives. The primary objective of the follower truck is to keep a safe distance from its preceding truck in a platoon. In a connected vehicle environment, an additional objective is to track the lead truck's speed profile smoothly, which provides further fuel savings beyond the benefits of aerodynamic drag reduction [11]. Moreover, the V2V communication enhances powertrain coordination, mitigating aggressive engine transients and thereby reducing tailpipe NO_x emissions [3]. These objectives are for-

mulated as a multi-objective optimal control problem within an NMPC framework.

A receding-horizon control framework is employed to implement the NMPC, where at each discrete time step an optimal sequence of control inputs is computed over a prediction horizon N_p . To establish the predictive control formulation, the nonlinear truck longitudinal dynamics model is discretized with the sampling time Δt using the first-order Euler approximation. The control problem is then posed as an optimization problem subject to constraints on tracking errors and control inputs. Finally, the optimization problem is transformed into a nonlinear programming with a quadratic cost function.

$$\begin{aligned}
\min_{T_{des,i}(k)} J(k) = & \overbrace{\sum_{j=1}^{N_p} w_e \|e_{s,i}(k+j | k)\|^2}^{\text{Platoon Longitudinal Safety}} \\
& + \overbrace{\sum_{j=1}^{N_p} w_v \|v_i(k+j | k) - v_i(k+j-1 | k)\|^2}^{\text{Fuel Saving}} \\
& + \overbrace{\sum_{j=1}^{N_c-1} w_T \|T_{des,i}(k+j | k) - T_{des,i}(k+j-1 | k)\|^2}^{\text{NO}_x \text{ Emissions Reduction}}
\end{aligned} \tag{6.1}$$

subject to

$$\begin{aligned}
& \mathbf{x}_{s,i}(k+j+1 | k) = \mathbf{x}_{s,i}(k+j | k) + f_p(\mathbf{x}_{s,i}(k+j | k), \mathbf{u}_{s,i}(k+j | k))\Delta t & (6.2a) \\
& e_{s,i}^{\min} \leq e_{s,i}(k+j | k) \leq e_{s,i}^{\max} & (6.2b) \\
& v_i^{\min} \leq v_i(k+j | k) \leq v_i^{\max} & (6.2c) \\
& \omega_i^{\min} \leq \omega_{e,i}(k+j | k) \leq \omega_{e,i}^{\max} & (6.2d) \\
& 0 \leq T_{e,i}(k+j | k) \leq T_{e,i}^{\max} & (6.2e) \\
& 0 \leq T_{b,i}(k+j | k) \leq T_{b,i}^{\max} & (6.2f)
\end{aligned}$$

where, w is the tuning weight, k represents the current time step, $e_{s,i}^{\min}$ and $e_{s,i}^{\max}$ are the minimum and maximum inter-vehicle distance error, respectively; $v_i^{\min}$ and $v_i^{\max}$ are the minimum and maximum bounds of the trucks' speed, respectively; $\omega_i^{\min}$ and $\omega_i^{\max}$ denote the minimum and maximum bounds of the engine speed, respectively; $T_{e,i}^{\max}$ and $T_{b,i}^{\max}$ represent the maximum engine and braking torque, respectively.

The first term of the cost function in Eq. (6.1) penalizes the longitudinal spacing error between two consecutive trucks to ensure the safety of the truck platoon. The second term penalizes fluctuations in the follower truck’s speed, promoting a smoother speed profile and consequently further reducing fuel consumption. The third term penalizes abrupt changes in engine torque, which mitigates transient NO_x spikes.

The NMPC problem is subject to the system dynamics in Eq. (6.2a), ensuring that torque commands remain physically feasible. A safety buffer in Eq. (6.2b) prevents potential collisions and maintains stable platooning operation. Eq. (6.2c) bounds the truck’s speed to comply with highway limits and promote smoother speed profiles, while Eq. (6.2d) constrains the engine speed within its feasible operating range. The engine torque is bounded by Eq. (6.2e) to prevent exceeding the rated peak of the Cummins X15 engine used in this study, while Eq. (6.2f) enforces the operational limits of the braking system. Collectively, these constraints ensure that the NMPC formulation is both safe and practically implementable for real-world truck platooning operation.

6.2.2 Lateral Platoon Controller

The lateral platoon controller is developed using an NMPC framework to explicitly regulate lane keeping, platoon lateral alignment, and lateral stability under crosswinds while preserving the aerodynamic benefits associated with wake alignment. The nonlinear lateral dynamics model introduced in Chapter 3 is discretized with sampling time Δt , and the longitudinal velocity planned by the longitudinal controller is treated as a time-varying parameter in the lateral prediction model. This design choice embeds longitudinal–lateral coupling directly into the NMPC prediction, ensuring that lateral tracking performance remains consistent with the planned speed trajectory.

At each time step k , the NMPC computes the steering input $\delta_i(k)$ by minimizing a multi-objective cost that captures four requirements: (i) lane-centering, (ii) yaw-

angle tracking, (iii) lateral-offset regulation between platoon trucks to preserve fuel-saving wake exposure, and (iv) yaw-rate smoothing to improve lateral stability. The optimization problem is formulated as

$$\begin{aligned}
\min_{\delta_i(k)} J(k) = & \overbrace{\sum_{j=1}^{N_p} w_y \|y_i(k+j | k) - y_{ref,i}(k+j | k)\|^2}^{\text{Platoon Lateral Safety}} \\
& + \overbrace{\sum_{j=1}^{N_p} w_a \|\phi_i(k+j | k) - \phi_{ref,i}(k+j | k)\|^2}^{\text{Yaw Angle}} \\
& + \overbrace{\sum_{j=1}^{N_p} w_l \|e_{l,i}(k+j | k)\|^2}^{\text{Fuel Saving}} \\
& + \overbrace{\sum_{j=1}^{N_p} w_r \|\phi_i(k+j | k) - \phi_i(k+j-1 | k)\|^2}^{\text{Lateral Stability}}
\end{aligned} \tag{6.3}$$

subject to

$$\begin{aligned}
& \begin{cases} \mathbf{x}_{l,i}(k+j+1 | k) = \mathbf{x}_{l,i}(k+j | k) + f_p(\mathbf{x}_{l,i}(k+j | k), \mathbf{u}_{l,i}(k+j | k))\Delta t \end{cases} \tag{6.4a} \\
& \begin{cases} e_{l,i}^{\min} \leq e_{l,i}(k+j | k) \leq e_{l,i}^{\max} \end{cases} \tag{6.4b} \\
& \begin{cases} \dot{\phi}_i^{\min} \leq \dot{\phi}_i(k+j | k) \leq \dot{\phi}_i^{\max} \end{cases} \tag{6.4c} \\
& \begin{cases} \delta_i^{\min} \leq \delta_i(k+j | k) \leq \delta_i^{\max} \end{cases} \tag{6.4d}
\end{aligned}$$

where $y_{ref,i}$ and $\phi_{ref,i}$ denote the safe lateral position and yaw angle reference trajectories for follower truck i , respectively, generated in the upper layer using high-definition maps or onboard perception sensors. The variables $e_{l,i}^{\min}$ and $e_{l,i}^{\max}$ define the minimum and maximum allowable lateral offsets, while $\phi_i^{\min}$ and $\phi_i^{\max}$ represent the lower and upper bounds of the truck's yaw angle. Similarly, $\delta_i^{\min}$ and $\delta_i^{\max}$ specify the minimum and maximum steering angle limits.

The first term of the cost function in Eq. (6.3) penalizes deviations of the follower truck's lateral position from the lane-centerline reference, while the second term penalizes yaw angle errors. Together, these two terms ensure lateral safety of the truck

platoon. The third term explicitly regulates the inter-vehicle lateral offset, which is critical for preserving platooning fuel-saving benefits because lateral misalignment can move the follower out of the low-drag wake region. The fourth term penalizes rapid changes in yaw rate, improving lateral stability and mitigating oscillatory steering behavior under crosswinds.

The NMPC formulation incorporates a set of constraints to ensure feasibility and practical implementation, while explicitly accounting for longitudinal–lateral coupling. The lateral vehicle dynamics in Eq. (6.4a) are enforced using the longitudinal velocity provided by the longitudinal controller as a time-varying parameter, thereby embedding the planned speed profile into the lateral controller. This coupling is essential because variations in longitudinal motion directly affect lateral response and tracking performance. The lateral-offset buffer imposed in Eq. (6.4b) preserves the fuel-saving benefits of platooning. The yaw rate is constrained in Eq. (6.4c) to prevent abrupt rotational motion, thereby enhancing lateral stability. In addition, the steering-actuation limits in Eq. (6.4d) ensure that the commanded steering angles remain within the physical capabilities of the truck’s steering system.

For both the developed longitudinal and lateral controllers, the sampling time, Δt , is set to 0.1 sec, representing a practical balance between numerical fidelity and computational load, consistent with widely adopted values in truck dynamics modeling [165]. The prediction horizon, N_p , is chosen as 50, corresponding to a 5-sec prediction window that covers the largest time gap considered in this study. The tuning weights are selected based on trade-offs among safety, fuel consumption, and emissions. A summary of the key NMPC parameters is provided in Table 6.2.

Figure 6.2 illustrates the block diagrams of the integrated NMPC-based longitudinal–lateral control system. At each control step, the SQP solver computes the optimal input sequences $\{T_{\text{des},i}\}$ and $\{\delta_i\}$ over the horizons subject to constraints, and only the first elements are applied in a receding-horizon manner. With this formulation, the longitudinal NMPC plans energy- and emission-aware torque trajectories

Table 6.2: Key parameters of the developed controllers in Figure 6.1.

Parameter	Symbol	Value	Unit
Sampling Time	Δt	0.1	sec
Prediction Horizon	N_p	50	-
Tuning Weight	w_e	10	-
Tuning Weight	w_v	5	-
Tuning Weight	w_T	2	-
Tuning Weight	w_y	10	-
Tuning Weight	w_a	10	-
Tuning Weight	w_l	5	-
Tuning Weight	w_r	2	-
Minimum Spacing error	$e_s^{\min}$	-5	m
Maximum Spacing error	$e_s^{\max}$	5	m
Minimum Truck Speed	$v^{\min}$	20	m/s
Maximum Truck Speed	$v^{\max}$	31	m/s
Peak Engine Torque	$T_e^{\max}$	2272	N·m
Maximum Braking Torque	$T_b^{\max}$	60000	N·m/axle
Minimum Lateral Offset	$e_l^{\min}$	-0.5	m
Maximum Lateral Offset	$e_l^{\max}$	0.5	m
Minimum Yaw Rate	$\dot{\phi}^{\min}$	-0.5	rad/s
Maximum Yaw Rate	$\dot{\phi}^{\max}$	0.5	rad/s
Minimum Steering Angle	$\delta^{\min}$	-0.5	rad
Maximum Steering Angle	$\delta^{\max}$	0.5	rad

that maintain safe spacing, while the lateral NMPC preserves lane keeping and wake alignment under crosswinds to sustain platooning energy benefits.

6.3 Numerical Evaluation and Discussion

6.3.1 Simulation Setup

To evaluate the performance of the designed longitudinal and lateral controllers, this chapter conducts high-fidelity simulations in place of additional on-road platooning experiments, which are logistically and financially demanding. The simulations build

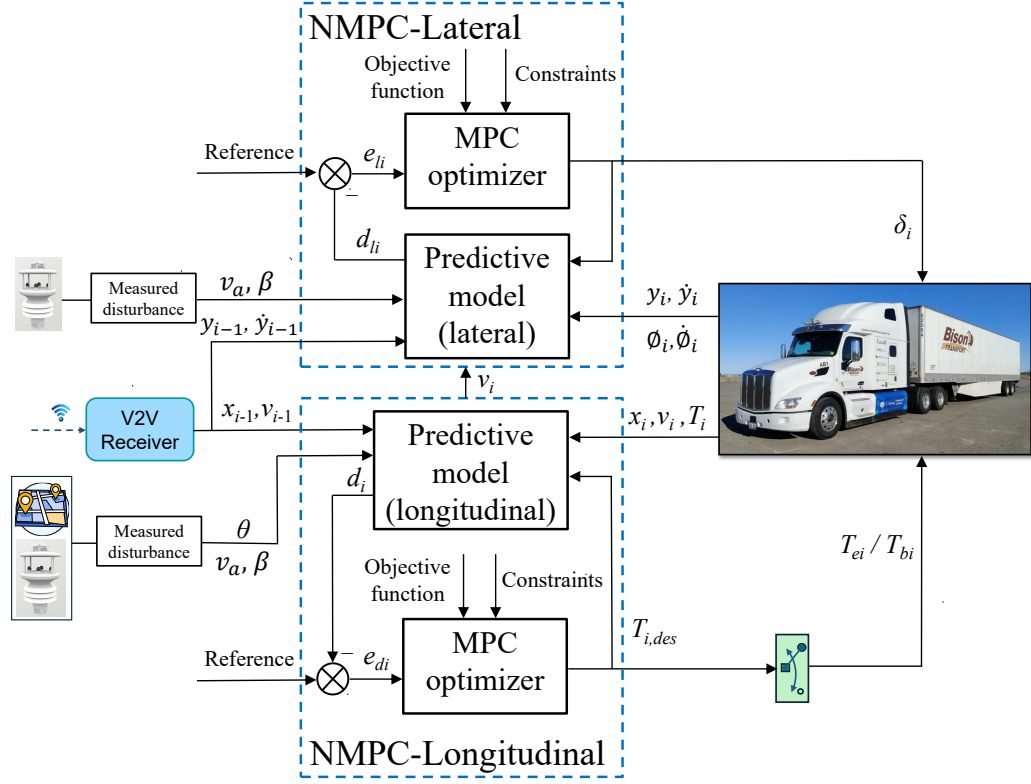


Figure 6.2: Block diagram of the integrated longitudinal and lateral control structure for truck platooning.

upon the experimentally validated truck-platoon models developed in Chapter 3 and leverage real-world measurements collected during the on-road cooperative platooning trials to ensure that the evaluated scenarios remain representative of public-highway operation.

A two-vehicle Class-8 platoon is considered, consisting of a lead truck and a follower truck with identical powertrain and vehicle parameters. The key parameters used in the platoon model are summarized in Table 6.3. The Alberta Highway 2 driving cycle reconstructed from recorded on-road data is adopted as the reference trajectory for evaluation. To reflect realistic operating environments, the measured road-grade profiles and wind conditions are incorporated into the simulation, thereby capturing the dominant exogenous disturbances affecting both longitudinal and lateral motions.

Table 6.3: Key parameters of the platoon trucks.

Parameter	Symbol	Value	Unit
Tractor Mass	m_1	9300	kg
Trailer Mass	m_2	7048	kg
Rotational Mass Coefficient	δ_m	1.05	-
Effective Rolling Radius	r_{eff}	0.5	m
Air Density	ρ	1.2922	kg/m ³
Frontal Area	A	10	m ²
Reference Drag Coefficient	C_{Dref}	0.75	-
Rolling Resistance Coefficient	μ	0.006	-
Gravitational Acceleration	g	9.81	m/s ²
Gear Ratio	i_g	0.77-14.41	-
Final Drive Ratio	i_0	2.85	-
Drivetrain Efficiency	η_t	0.9	-
Yaw Moment of Inertia of Tractor	I_1	31500	kg·m ²
Yaw Moment of Inertia of Trailer	I_2	26300	kg·m ²
Cornering Stiffness 1	C_1	160000	N/rad
Cornering Stiffness 2	C_2	120000	N/rad
Cornering Stiffness 3	C_3	120000	N/rad
Cornering Stiffness 4	C_4	125000	N/rad
Cornering Stiffness 5	C_5	125000	N/rad

The first evaluation scenario is derived from the on-road truck platooning trials, specifically trip 6 in Table 6.1. In this scenario, the lead truck operates at speeds between 85 and 104 km/h, and the follower truck is tracking the lead truck with a time gap of 5 sec. The ambient wind conditions include speeds ranging from 13 to 30 km/h and directions between 61° and 77°. This scenario is particularly representative of real-world highway operations, as it combines moderate-to-high vehicle speeds with varying wind conditions, thereby providing a rigorous test of the designed controllers in maintaining safe platooning, while maximizing fuel savings and minimizing tailpipe NO_x emissions.

6.3.2 Simulation Results and Analyses

Figure 6.3 presents the results of truck platooning using the designed longitudinal and lateral controllers. The longitudinal controller enables the follower truck to effectively track the lead truck, with a maximum speed difference of 0.3 m/s, as shown in Figure 6.3(a). Furthermore, the follower truck exhibits fewer speed fluctuations compared to the lead truck. Speed smoothness is quantified using the normalized curvature-based index $\int_{t_1}^{t_2} C'''(t)^2 dt / C_{max}^2$, where smaller values indicate smoother profiles. The lead truck's speed smoothness is 52.8, while the follower truck's speed smoothness is only 4.7, confirming the effectiveness of the controller in moderating speed oscillations. Figure 6.3(b) shows that the maximum spacing error is only 0.12 m, well within the safety buffer of 5 m. This demonstrates that the designed controller ensures longitudinal safety of the platoon. Figures 6.3(c) and (d) illustrate the lateral performance of the platoon. The follower truck's yaw rate remains bounded between -0.06 and 0.07 rad/s, indicating stable directional maneuvers. In addition, the lateral offset relative to the lead truck is maintained within 0.1 m, demonstrating precise alignment between trucks. These results confirm that the lateral controller effectively enhances stability and ensures consistent tracking performance under crosswind disturbances.

Figures 6.3(e) and (f) present the control inputs of engine torque and steering angle. The results show that torque fluctuations in the follower truck are significantly suppressed, demonstrating the effectiveness of the proposed longitudinal controller in optimizing torque behavior under varying wind conditions. To quantify this effect, the previously defined smoothness index is also applied to engine torque. The lead truck exhibits a torque smoothness index of 37,010, whereas the follower truck's value is substantially lower at 1,740. This reduction confirms the controller's capability to mitigate engine torque transients. In addition, the steering angle of the follower truck exhibits reduced variability, indicating consistent lateral performance.

Given the relatively large inter-vehicle spacing, the aerodynamic influence on the

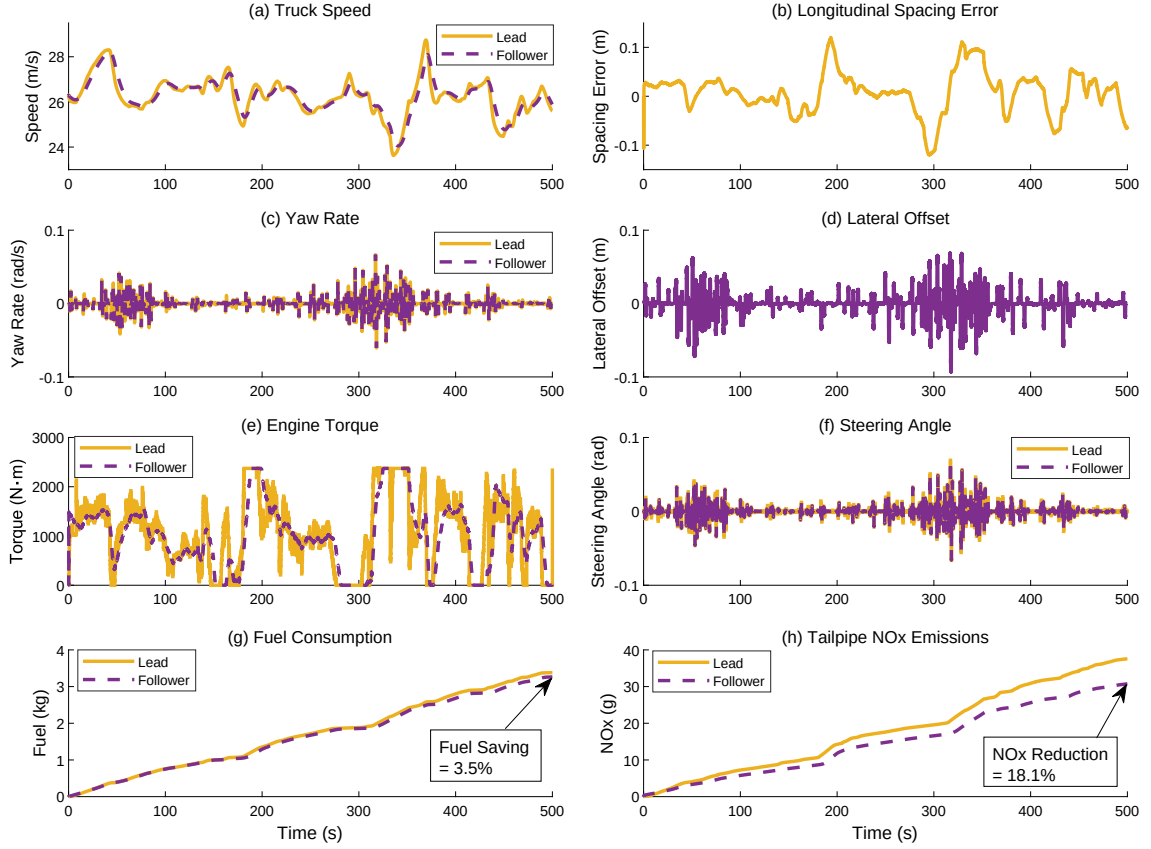


Figure 6.3: Results of 5-sec time gap truck platooning with the designed control system under Alberta Highway 2 driving cycle. The environmental wind speed ranges from 13 to 30 km/h, wind direction varies from 61° to 77° , and the ambient temperature is -13°C .

lead truck is negligible [34]. Therefore, the lead truck is modeled as the control vehicle and serves as the baseline for comparison. At a 5-sec time gap, where aerodynamic drag reduction is marginal, coordinated speed profiling emerges as the dominant source of fuel savings for the follower truck. As shown in Figure 6.3(g), the follower truck achieves a 3.5% reduction in fuel consumption relative to the lead vehicle. In addition, tailpipe NO_x emissions are highly sensitive to rapid engine torque fluctuations, which cause unfavorable combustion conditions and transient air–fuel imbalances in diesel engines [77]. Figure 6.3(h) shows that the follower truck achieves an 18.1% reduction in cumulative tailpipe NO_x emissions compared to the lead truck. This improvement can be attributed to the follower’s smoother engine torque behavior

during platooning, as illustrated in Figure 6.3(e).

These results exceed the performance reported in Table 6.1, where the follower truck operated under a commercial platooning system without optimal coordination between the vehicle dynamics and the platoon controllers. These findings demonstrate the effectiveness of the designed controllers in reducing both fuel consumption and tailpipe NO_x emissions under real-world driving conditions.

In addition to the 5-sec time gap scenario, this chapter also investigates shorter time gaps of 3 and 4 sec using the same driving conditions outlined in Table 6.1. Moreover, a 2-sec time gap is examined to reflect the minimum following distances permitted in certain jurisdictions [134, 166]. The corresponding results are presented in Figures 6.4, 6.5, and 6.6. All simulations were carried out on a PC equipped with an Intel® Core™ i7-9700 CPU @ 3.00 GHz and 32 GB RAM. The average solver computation time for the proposed 2-MPC was 19.3 ms per 100 ms sampling interval, with a maximum of 45.2 ms. The total computation time for the 500-sec test trip was 90.7 sec, demonstrating the real-time feasibility of the proposed control system.

Table 6.4 compares the performance of the proposed NMPC-based coordinated control system with the cooperative truck platooning system (baseline from on-road trials) across different time gaps. For both systems, shorter gaps increase the maximum spacing error due to the higher trajectory-tracking demands. However, NMPC maintains spacing errors within the 5 m safety buffer at all time gaps, with a maximum of 4.5 m at 2 sec, whereas CTPS produces substantially larger errors between 15.7 and 19.1 m. In terms of lateral performance, NMPC limits the maximum lateral offset within 0.1 m consistently across all time gaps, thereby preserving the fuel-saving potential of platooning. By contrast, CTPS yields much larger offsets of 0.5–0.7 m, which erodes aerodynamic benefits.

Fuel consumption results further highlight the advantage of the proposed control system. With NMPC, fuel savings range from 3.5% to 5% across time gaps of 3-5 sec, compared to only 0.3% to 2.1% under CTPS. Notably, the highest saving of 6.6% is

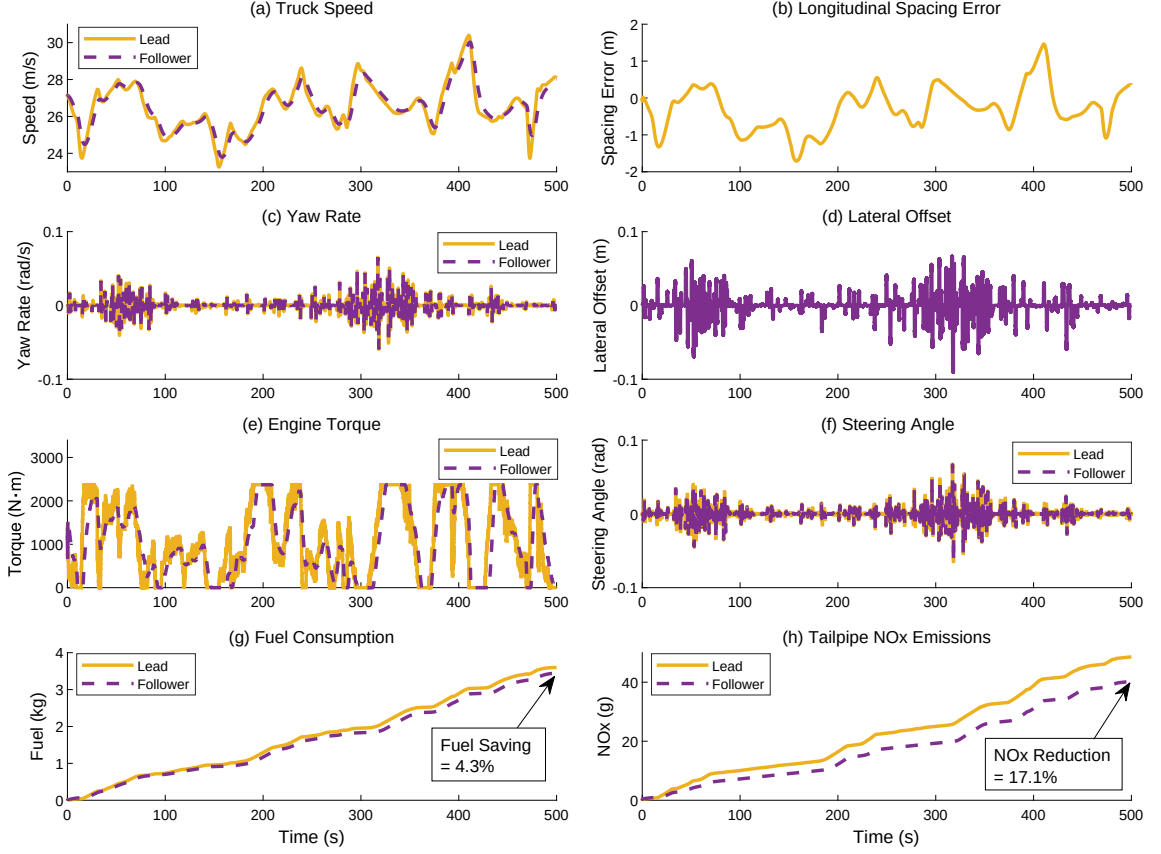


Figure 6.4: Results of 4-sec time gap truck platooning with the designed control system under Alberta Highway 2 driving cycle. The environmental wind speed ranges from 10 to 26 km/h, wind direction varies from 31° to 68° , and the ambient temperature is -7°C .

achieved at a 2-sec time gap due to increased aerodynamic drag reduction.

Furthermore, the NMPC controller achieves consistent reductions in tailpipe NO_x emissions ranging from 17.1% to 20.6%, whereas CTPS results in significantly higher NO_x emissions, with increases between 19.2% and 32.3%. These results indicate that while fuel savings are sensitive to the selected time gaps, NO_x reduction benefits of the designed control system remain robust and persistent.

Overall, the proposed NMPC framework demonstrates superior performance over the commercial CTPS in maintaining safe platoon operation, improving fuel savings, and substantially reducing tailpipe NO_x emissions with varying time gaps under real-world driving conditions.

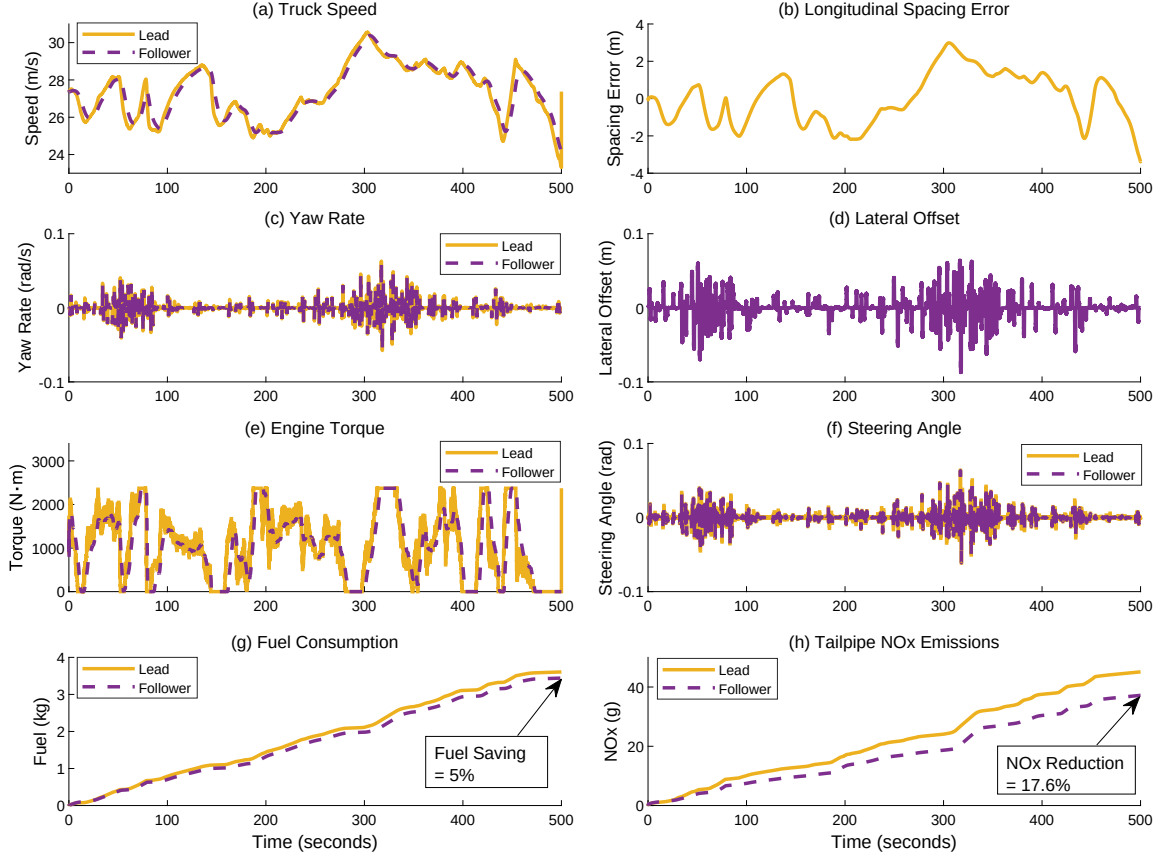


Figure 6.5: Results of 3-sec time gap truck platooning with the designed control system under Alberta Highway 2 driving cycle. The environmental wind speed ranges from 10 to 21 km/h, wind direction varies from 147° to 165° , and the ambient temperature is 2°C .

6.4 Summary of Chapter

This chapter designed a coordinated truck platooning control system that integrates longitudinal and lateral controllers within a nonlinear model predictive control framework. The proposed architecture was developed to maintain safe platooning while enhancing fuel savings and reducing tailpipe nitrogen oxides emissions, particularly in highway scenarios where small time gaps (< 2 sec) are restricted by safety regulations and environmental conditions.

High-fidelity simulations on the Alberta Highway 2 driving cycle, incorporating measured wind and road-grade effects, demonstrated that the proposed control system maintained safe platooning operation in both the longitudinal and lateral direc-

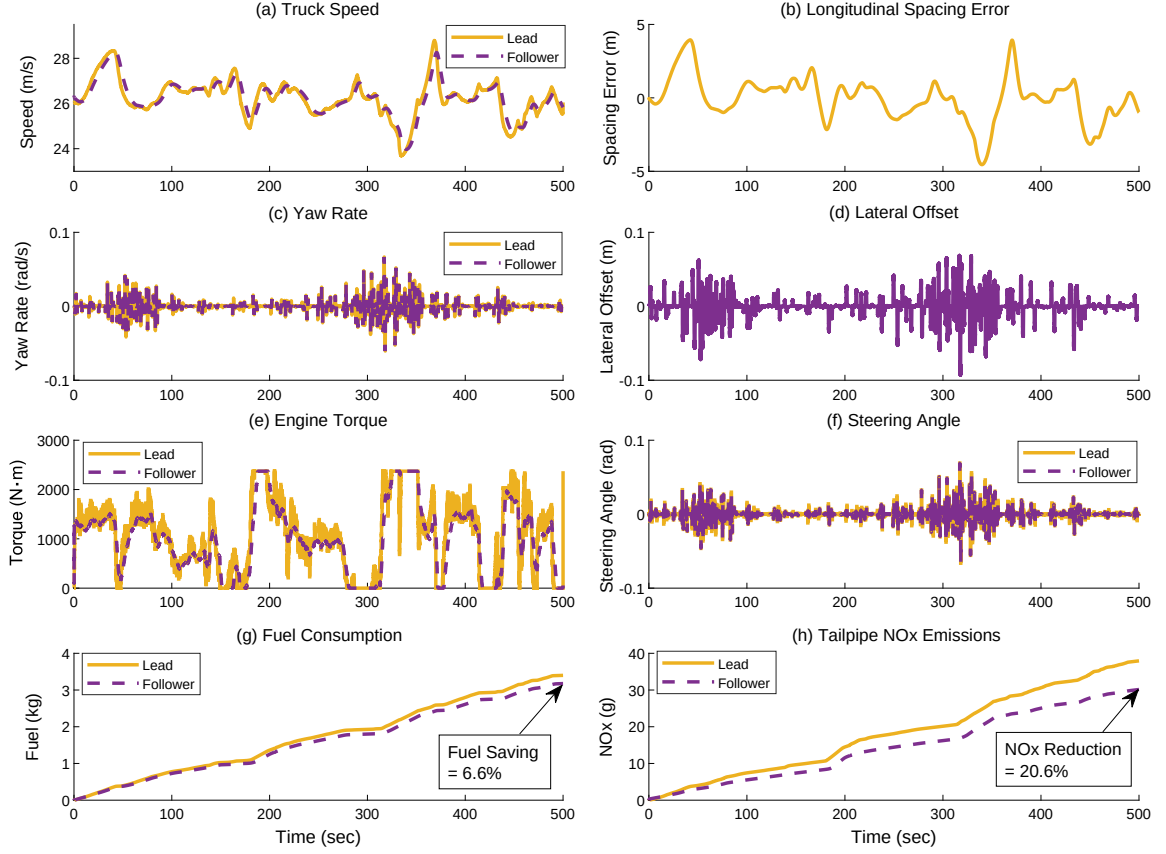


Figure 6.6: Results of 2-sec time gap truck platooning with the designed control system under Alberta Highway 2 driving cycle. The environmental wind speed ranges from 13 to 25 km/h, wind direction varies from 32° to 61° , and the ambient temperature is -5°C .

tions while achieving notable benefits even at large platoon spacings. Fuel savings of up to 6.6% were observed at a 2-sec time gap, with a 3.5% saving still obtained at 5 sec. These results indicated that, beyond aerodynamic drag reduction, coordinated speed profiling is a key mechanism for improving fuel efficiency by suppressing unnecessary accelerations and decelerations. In addition, the integrated lateral controller constrained inter-vehicle lateral offsets (within 0.1 m in the evaluated scenarios), further supporting fuel-saving performance. The NMPC-based controllers also effectively mitigated abrupt engine torque fluctuations, yielding substantial reductions in tailpipe NO_x emissions. At a 5-sec time gap, the follower truck's cumulative tailpipe NO_x emissions decreased by 18.1%, representing a 50.4% improvement compared with

Table 6.4: Comparison of key performance metrics between NMPC and CTPS truck platooning control systems among different time gaps.

Controller	NMPC				CTPS ¹			
Time Gap (sec)	2	3	4	5	2	3	4	5
Maximum Spacing Error (m)	4.5	3.4	1.7	0.1	-	19.1	17.8	15.7
Maximum Lateral Offset (m)	0.1	0.1	0.1	0.1	-	0.5	0.6	0.7
Fuel Saving (%) ²	6.6	5	4.3	3.5	-	2.1	1.6	0.3
NO _x Reduction (%) ³	20.6	17.6	17.1	18.1	-	-19.2	-25.2	-32.3

¹ CTPS stands for the cooperative truck platooning system used in the on-road trials.

² The fuel saving of the follower truck is calculated by (fuel consumption of lead truck - fuel consumption of follower truck)/fuel consumption of lead truck*100%.

³ The NO_x reduction of the follower truck is calculated by (NO_x emissions of lead truck - NO_x emissions of follower truck)/NO_x emissions of lead truck*100%.

the baseline on-road cooperative truck platooning system.

Overall, the results demonstrated that the proposed integrated longitudinal–lateral control system can enhance platooning safety, improve fuel savings, and reduce tailpipe NO_x emissions under crosswinds.

Chapter 7

Electrified Truck Platooning¹

The rapid electrification of Class-8 trucks—spanning hybrid electric, plug-in hybrid electric, and battery-electric architectures—is reshaping the design space of truck platooning. Figure 7.1 classifies truck platooning by powertrain electrification levels. Beyond conventional aerodynamic drag reduction, electrified platoons can exploit fast torque response, regenerative braking, and flexible power-split operation to coordinate speed trajectories more tightly and recover energy that would otherwise be dissipated during braking [41, 81]. In hybrid and plug-in hybrid platoons, the electric path can also buffer transient power demands, enabling smoother internal combustion engine operation and creating new opportunities to reduce fuel consumption and tailpipe NO_x emissions simultaneously. For battery-electric platoons, the absence of tailpipe emissions shifts the focus toward minimizing electrical energy consumption and extending driving range, where coordinated speed smoothing and improved utilization of regenerative braking can be as important as aerodynamic coupling. These electrification-enabled mechanisms complement wake benefits and motivate a unified platooning control approach that simultaneously coordinates inter-vehicle motion and optimizes powertrain operation across multiple electrification levels.

To address this need, this chapter develops a hierarchical energy and emissions management control system that integrates platooning strategies with different powertrain electrification levels. The upper-level controller employs model predictive con-

¹ This chapter is partially based on [6, 8].

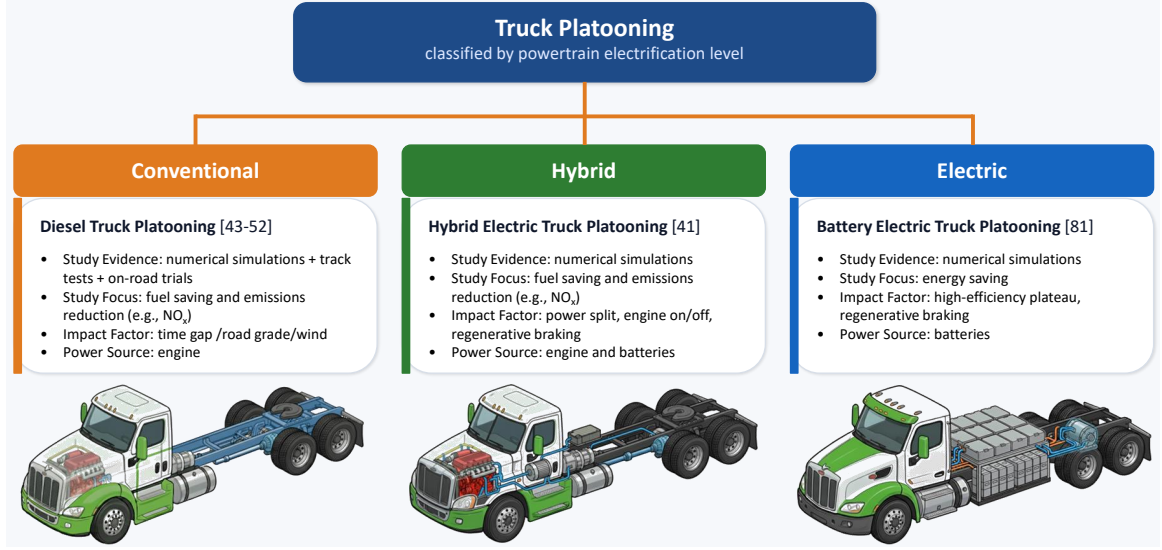


Figure 7.1: Truck platooning classified by powertrain electrification levels.

trol to ensure safe inter-vehicle spacing while generating smooth speed trajectories to suppress unnecessary deceleration–acceleration cycles. The lower-level controller also uses model predictive control to track the upper-level torque demand and optimize propulsion commands to minimize total energy consumption; for hybrid powertrains, it further mitigates engine transients to reduce tailpipe NO_x emissions. The proposed framework is evaluated on homogeneous two-truck platoons with conventional diesel, pre-transmission parallel (P2) hybrid electric, plug-in hybrid electric (PHEV), and battery-electric (BEV) configurations using real-world data collected from on-road truck-platooning trials. The remainder of the chapter presents the electrified powertrain modeling, the proposed hierarchical control design, and a systematic performance comparison across electrification levels and time-gap settings under realistic highway driving conditions.

7.1 Electrified Truck Modelling

The model for a conventional diesel truck (Figure 7.2(a)) was developed in Chapter 3. In this chapter, three representative electrified powertrain configurations are considered for platooning trucks: a pre-transmission parallel hybrid electric vehicle

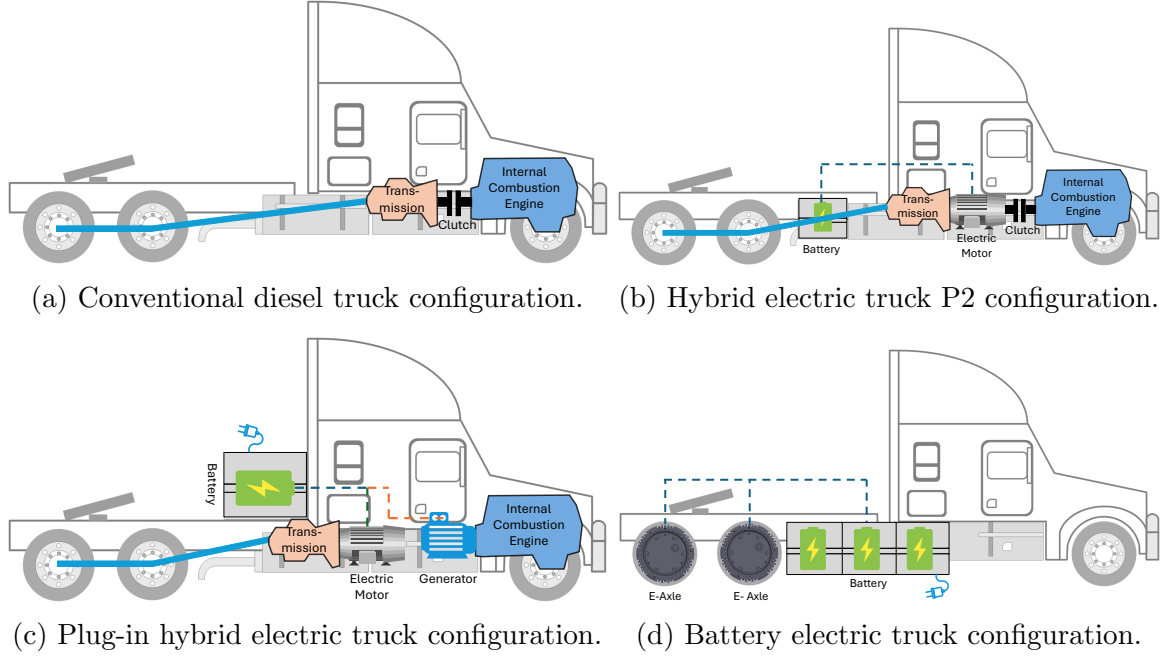


Figure 7.2: Schematics of conventional and electrified truck configurations.

(P2-HEV), a plug-in hybrid electric vehicle (PHEV), and a battery electric vehicle (BEV). Figure 7.2(b) illustrates the P2 hybrid architecture, in which the electric motor and clutch are positioned between the internal combustion engine and transmission. This configuration utilizes the electric driveline to assist during acceleration and peak power demands while also enabling regenerative braking, with minimal changes to the conventional driveline. In contrast, the plug-in hybrid configuration in Figure 7.2(c) employs higher-capacity batteries that can be recharged from the electrical grid, thereby extending electric-only operation and offering greater potential for reducing fuel consumption and tailpipe emissions. Finally, the BEV configuration in Figure 7.2(d) eliminates the internal combustion engine and relies solely on a rechargeable battery, resulting in zero tailpipe emissions during operation. For the BEV, an electric axle (E-Axle) integrates the electric motor, transmission, and inverter within a single housing to convert battery power into tractive effort, while also supporting regenerative braking for energy recovery during deceleration.

The inclusion of P2-HEV, PHEV, and BEV configurations enables a comprehensive

assessment of electrification level and its effects on energy optimization and emissions reduction in truck platooning. The Class-8 electrified-truck simulation models used in this chapter are developed to meet two key objectives: (1) matching the performance criteria of the experimentally validated Class-8 conventional diesel baseline, and (2) minimizing energy consumption over the Alberta Highway 2 drive cycle under full cargo load relative to the conventional baseline.

The electrified truck model is developed by extending the baseline vehicle model to incorporate an electric machine (EM) and a battery model. The EM operates in two modes: as a motor during propulsion and as a generator during regenerative braking. Similar to the ICE representation, the EM is modelled based on torque–speed efficiency maps, which characterize its energy conversion efficiency across different operating points. Figure 7.3 shows representative EM efficiency maps for the hybrid and battery-electric configurations. The electrical power of the EM is computed as:

$$P_{\text{elec}} = \begin{cases} \frac{1}{\eta_m(\omega_m, T_m)} \omega_m T_m & P_{\text{elec}} \geq 0 \text{ (discharging)} \\ \eta_m(\omega_m, T_m) \omega_m T_m & P_{\text{elec}} < 0 \text{ (charging)} \end{cases} \quad (7.1a)$$

where P_{elec} is the electric power, η_m is the electric motor efficiency, ω_m is the motor angular speed, and T_m is the motor torque.

In this thesis, the temperature effect on the batteries is neglected, and the batteries are simplified as a zero order equivalent circuit model [167]. The relationship between the battery power, current, and state of charge (SOC) is expressed as follows:

$$\begin{cases} I_b = \frac{(V_{oc} - \sqrt{V_{oc}^2 - 4R_b P_b})}{2R_b} \end{cases} \quad (7.2a)$$

$$\begin{cases} \text{SOC}(t) = \text{SOC}_{\text{init}} - \frac{1}{Q_{\text{nom}}} \int_0^t I_b(t) dt \end{cases} \quad (7.2b)$$

where I_b is the current, V_{oc} is the open circuit voltage, R_b is the battery resistance, P_b is the battery power, SOC_{init} is initial state of charge of the battery, and Q_{nom} is the nominal battery capacity.

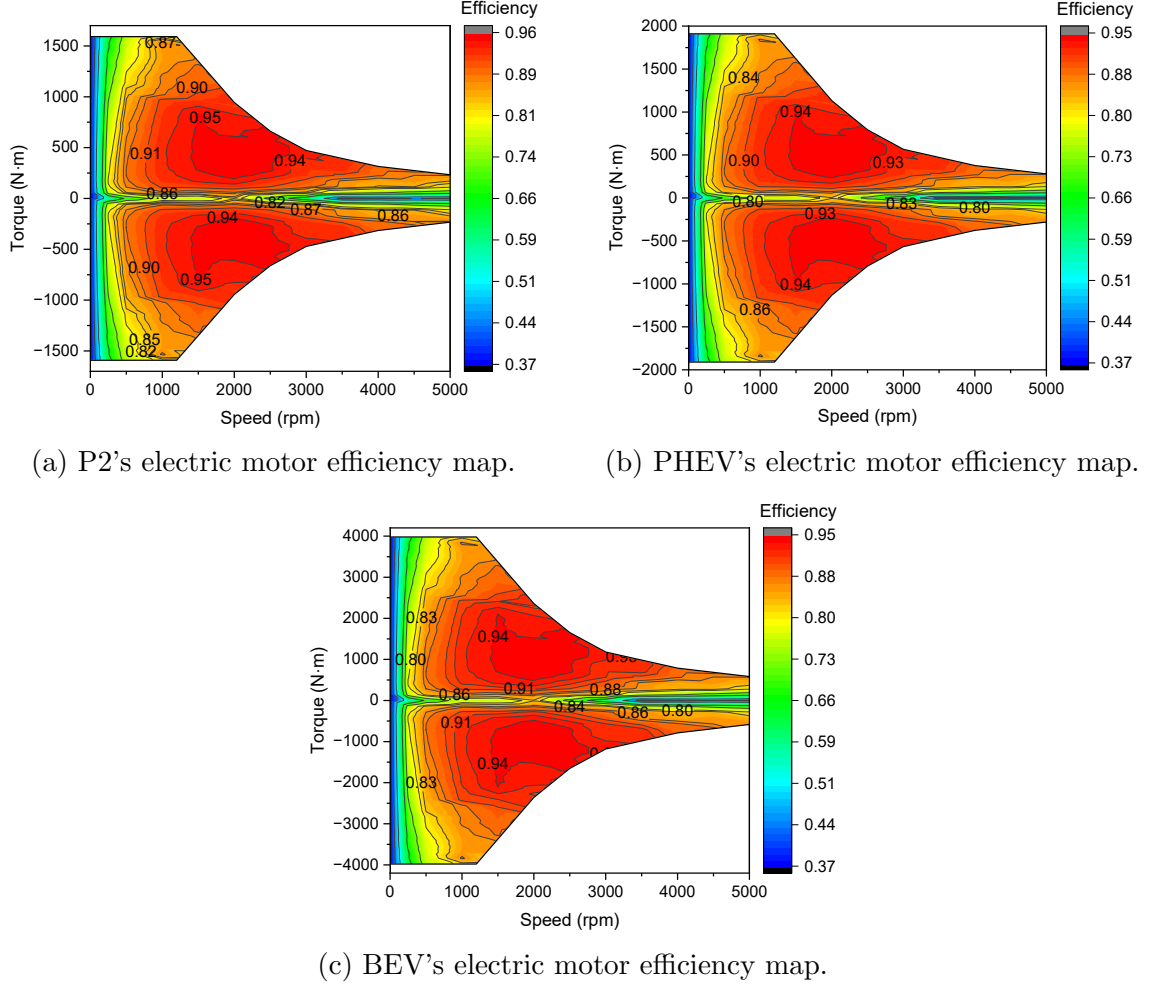


Figure 7.3: Efficiency maps of the electric motors in (a) P2 configuration, (b) PHEV configuration, and (c) BEV configuration.

7.2 Hierarchical Control Strategies for Electrified Truck Platoons

A hierarchical control framework is employed to design the truck platooning control system, consisting of an upper-level controller and a lower-level controller, as illustrated in Figure 7.4. This two-level structure is adopted to decompose the overall control problem into two coordinated subproblems with different control objectives and time scales. The upper-level controller determines the desired total torque using the model predictive control (MPC) method. It relies on driving information from both the lead and follower trucks, including position and speed, which are transmitted via

V2V communication. In this thesis, communication delays are not considered. The upper-level controller performs vehicle-level platoon coordination, including spacing regulation and speed tracking, while the lower-level controller optimally allocates the required torque between the ICE and EM using an MPC-based energy and emission management strategy. By separating vehicle motion control from powertrain control, the proposed framework reduces computational complexity and improves real-time feasibility compared with a single integrated MPC. The lower-level controller ensures that each truck accurately follows the coordinated speed trajectories generated by the upper-level controller while simultaneously optimizing energy conversion efficiency and minimizing tailpipe NO_x emissions. The integration of these two control layers enables the platoon to achieve safe, energy-efficient, and low-emission operation under real-world driving conditions.

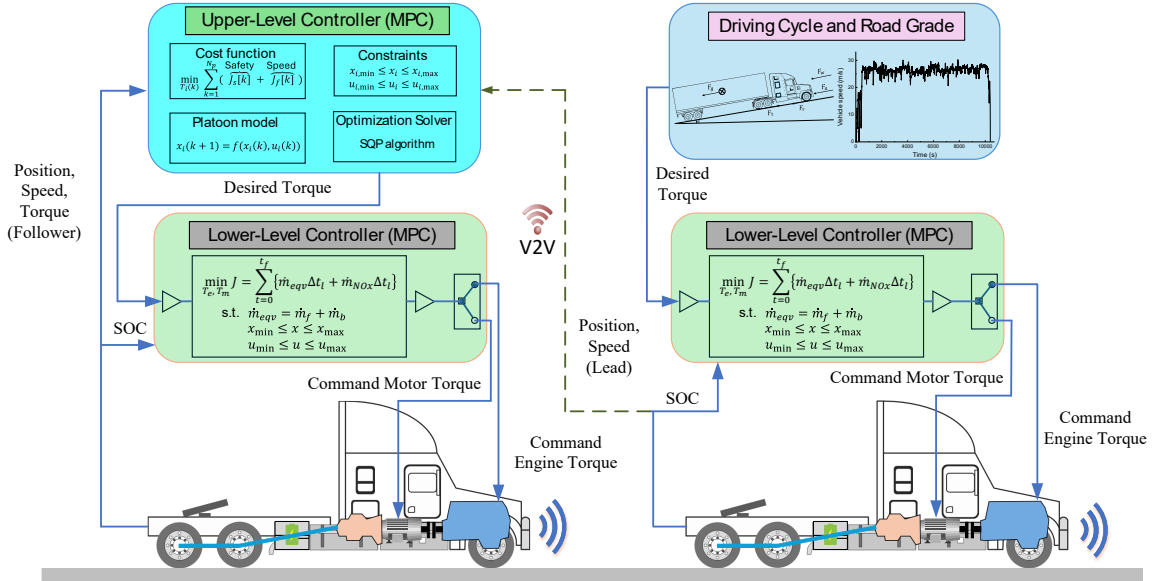


Figure 7.4: Hierarchical control architecture of the designed truck platooning system.

7.2.1 Upper-Level Controller

As shown in Figure 7.4, the lead truck follows a specified driving cycle, while the follower truck adjust its behavior accordingly to maintain the desired spacing and speed

coordination through the developed upper-level controller. The same upper-level controller is applicable to the truck platoons with different powertrain configurations (i.e., diesel, HEV, PHEV and BEV), aiming to ensure safe platooning operation and minimize the total energy consumption by generating smooth speed trajectories through the coordinated operation of all trucks in the platoon. These objectives are formulated as an MPC optimization problem, which determines the optimal desired total torque for each follower truck. To implement the MPC, the platoon dynamics model is discretized using a sampling time, Δt_u , via the first-order Euler approximation. The optimal control problem is formulated as follows:

$$\min_{T_{des,i}(k)} J(k) = \overbrace{\sum_{j=1}^{N_p} \frac{w_1}{s_1} \|e_{di}(k+j|k)\|^2}^{\text{Platoon Safety}} + \overbrace{\sum_{j=1}^{N_p} \frac{w_2}{s_2} \|v_i(k+j|k) - v_i(k+j-1|k)\|^2}^{\text{Smooth Speed Profile}} \quad (7.3)$$

subject to

$$\begin{aligned} \begin{cases} \mathbf{x}_i(k+j+1|k) = \mathbf{x}_i(k+j|k) + f_p(\mathbf{x}_i(k+j|k), \mathbf{u}_i(k+j|k))\Delta t \end{cases} & (7.4a) \\ \begin{cases} e_{di}^{\min} \leq e_{di}(k+j|k) \leq e_{di}^{\max} \end{cases} & (7.4b) \\ \begin{cases} v_i^{\min} \leq v_i(k+j|k) \leq v_i^{\max} \end{cases} & (7.4c) \\ \begin{cases} T_i^{\min} \leq T_i(k+j|k) \leq T_i^{\max} \end{cases} & (7.4d) \end{aligned}$$

where N_p represents the prediction horizon, w and s are the weight and scale factors, respectively. The constraints in Eq. (7.4) ensure safe and efficient platooning: $e_d^{\min}$ and $e_{di}^{\max}$ define the minimum and maximum allowable inter-vehicle distance errors, $v_i^{\min}$ and $v_i^{\max}$ set the lower and upper speed limits for the trucks, and $T_i^{\min}$ and $T_i^{\max}$ specify the permissible torque range.

The MPC optimization problem is solved using the sequential quadratic programming (SQP) method. At each time step k , the optimal control input sequence ($T_{des,i}(k) = [T_{des,i}(k|k), T_{des,i}(k+1|k), \dots, T_{des,i}(k+N_p-1|k)]$) is determined by minimizing the cost function while satisfying system constraints over the prediction horizon. However, only the first element of the computed optimal control input se-

quence, $T_{des,i}(k|k)$, is applied to the system. At the next time step $k+1$, the optimization is resolved over a shifted horizon with updated truck platoon state information $\mathbf{x}_i(k+1)$. Again, only the first element of the computed sequence, $T_{des,i}(k+1|k+1)$, is applied to the system. This receding horizon approach continuously adjusts the control input, ensuring real-time adaptability to changing driving conditions.

7.2.2 Lower-Level Controller

The lower-level controller executes configuration-specific energy and emissions management while tracking the desired total driveline torque command, $T_{des,i}$, generated by the upper-level controller. Depending on the powertrain architecture, the decision variables are the engine torque T_e (diesel), the engine–motor torque split (T_e, T_m) (P2-HEV/PHEV), or the motor torque T_m (BEV). In all cases, the lower-level controller is formulated as an MPC problem solved in a receding-horizon manner to ensure real-time adaptability and constraint satisfaction.

Diesel Truck Platooning

Conventional diesel truck platooning under the proposed hierarchical control architecture serves as the baseline case for energy and emissions management in this chapter. The objectives of the designed lower-level controller for conventional diesel truck platooning are to reduce fuel consumption and lower tailpipe NO_x emissions while accurately tracking the desired total driving torque generated by the upper-level controller. Because the diesel configuration relies exclusively on the ICE for propulsion, an MPC-based strategy is adopted to compute the commanded engine torque that achieves these objectives. The lower-level optimal control problem for truck i at time

step k is formulated as

$$\begin{aligned}
\min_{T_{e,i}(k)} J(k) = & \overbrace{\sum_{j=1}^{N_c} \frac{w_3}{s_3} \|T_{e,i}(k+j | k) - T_{des,i}(k+j-1 | k)\|^2}^{\text{Torque Tracking}} \\
& + \overbrace{\sum_{j=1}^{N_p} \frac{w_4}{s_4} \dot{m}_{f,i}(k+j | k) \Delta t_l}^{\text{Fuel Saving}} + \overbrace{\sum_{j=1}^{N_p} \frac{w_5}{s_5} \dot{m}_{NO_x,i}(k+j | k) \Delta t_l}^{\text{NO}_x \text{ Reduction}}
\end{aligned} \tag{7.5}$$

subject to

$$\begin{cases} \mathbf{x}_{di}(k+j+1 | k) = \mathbf{x}_{di}(k+j | k) + f_d(\mathbf{x}_{di}(k+j | k), \mathbf{u}_{di}(k+j | k)) \Delta t_l & (7.6a) \\ T_e^{\min} \leq T_e \leq T_e^{\max} & (7.6b) \\ \omega_e^{\min} \leq \omega_e \leq \omega_e^{\max} & (7.6c) \end{cases}$$

where $\dot{m}_f$ is the fuel consumption rate, $\dot{m}_{NO_x}$ is the tailpipe NO_x emissions rate, Δt_l is the lower-level sampling time, and $f_d(\cdot)$ represents the nonlinear diesel truck dynamics. The constraints (7.6b) and (7.6c) enforce feasible engine torque and speed operation, where $T_e^{\min}$ and $T_e^{\max}$ specify the allowable engine torque limits, and $\omega_e^{\min}$ and $\omega_e^{\max}$ specify the admissible speed range of the engine.

Hybrid Truck Platooning

The hybrid electric trucks considered in this chapter (P2-HEV and PHEV) utilize both the ICE and the battery–motor subsystem for propulsion. Their efficiency benefits mainly stem from regenerative braking and the ability to optimally coordinate the dual power sources over transient driving conditions. Given the desired total driving torque from the upper-level controller, the lower-level controller determines the optimal torque split between the ICE and the electric machine (EM) to ensure accurate torque tracking while improving energy efficiency. A further objective of this study is to explicitly minimize tailpipe NO_x emissions of hybrid powertrains, which is often neglected in prior platooning studies. The overall formulation of the MPC-based energy and emission management control strategy for each hybrid truck

i is given by:

$$\begin{aligned}
\min_{T_{e,i}(k), T_{m,i}(k)} J(k) = & \overbrace{\sum_{j=1}^{N_c} \frac{w_6}{s_6} \|T_{t,i}(k+j|k) - T_{des,i}(k+j-1|k)\|^2}^{\text{Torque Tracking}} \\
& \underbrace{\sum_{j=1}^{N_p} \frac{w_7}{s_7} \dot{m}_{eqv,i}(k+j|k) \Delta t_l}_{\text{Energy Saving}} + \underbrace{\sum_{j=1}^{N_p} \frac{w_8}{s_8} \dot{m}_{NO_x,i}(k+j|k) \Delta t_l}_{\text{NO}_x \text{ Reduction}}
\end{aligned} \tag{7.7}$$

subject to

$$\begin{cases}
\mathbf{x}_{hi}(k+j+1|k) = \mathbf{x}_{hi}(k+j|k) + f_h(\mathbf{x}_{hi}(k+j|k), \mathbf{u}_{hi}(k+j|k)) \Delta t_l & (7.8a) \\
T_t = T_e + T_m & (7.8b) \\
\dot{m}_{eqv} = \dot{m}_f + \dot{m}_b & (7.8c) \\
\dot{m}_b = \frac{s_b P_b}{Q_{lhv} \eta_{el}} & (7.8d) \\
SOC^{\min} \leq SOC \leq SOC^{\max} & (7.8e) \\
P_b^{\min} \leq P_b \leq P_b^{\max} & (7.8f) \\
T_e^{\min} \leq T_e \leq T_e^{\max} & (7.8g) \\
\omega_e^{\min} \leq \omega_e \leq \omega_e^{\max} & (7.8h) \\
T_m^{\min} \leq T_m \leq T_m^{\max} & (7.8i) \\
\omega_m^{\min} \leq \omega_m \leq \omega_m^{\max} & (7.8j)
\end{cases}$$

where T_t is the total driveline torque (sum of engine and motor torques); $\dot{m}_{eqv}$ is the equivalent fuel consumption rate; $\dot{m}_b$ is the energy consumption rate of batteries; $f_h(\cdot)$ denotes the nonlinear hybrid electric truck dynamics; s_b is the equivalence factor; Q_{lhv} is the lower heating value of diesel fuel (42.6 MJ/kg); η_{el} is the efficiency of the electrical path. In addition, $SOC^{\min}$ and $SOC^{\max}$ denote the permissible SOC bounds of the battery energy storage system; $P_b^{\min}$ and $P_b^{\max}$ represent the allowable battery power range; $T_e^{\min}$ and $T_e^{\max}$ are the minimum and maximum engine torques; $\omega_e^{\min}$ and $\omega_e^{\max}$ are the minimum and maximum engine speeds; $T_m^{\min}$ and $T_m^{\max}$ specify the admissible electric motor torque limits; and $\omega_m^{\min}$ and $\omega_m^{\max}$ denote the corresponding minimum and maximum motor speeds.

The constraint set in Eq. (7.8) enforces physical limits on the power sources of the engine and batteries. For the P2 configuration, the battery SOC is regulated to

sustain close to the reference SOC level. For the PHEV configuration, the SOC is maintained within a specified operating window, where the minimum SOC is set to 20% in this study to promote effective battery utilization while preserving long-term battery health.

Electric Truck Platooning

The battery electric powertrain configuration relies solely on the onboard batteries for propulsion and therefore produces zero tailpipe emissions during vehicle operation. Accordingly, the lower-level controller focuses on minimizing electrical energy consumption while accurately tracking the desired total driving torque command generated by the upper-level controller. For each electric truck i , the MPC-based energy management problem at time step k is formulated as

$$\begin{aligned} \min_{T_{m,i}(k)} J(k) = & \overbrace{\sum_{j=1}^{N_c} \frac{w_9}{s_9} \|T_{m,i}(k+j | k) - T_{des,i}(k+j-1 | k)\|^2}^{\text{Torque Tracking}} \\ & + \overbrace{\sum_{j=1}^{N_p} \frac{w_{10}}{s_{10}} P_{b,i}(k+j | k) \Delta t_l}^{\text{Energy Saving}} \end{aligned} \quad (7.9)$$

subject to

$$\begin{cases} \mathbf{x}_{ei}(k+j+1 | k) = \mathbf{x}_{ei}(k+j | k) + f_e(\mathbf{x}_{ei}(k+j | k), \mathbf{u}_{ei}(k+j | k)) \Delta t_l & (7.10a) \\ \text{SOC}^{\min} \leq \text{SOC} \leq \text{SOC}^{\max} & (7.10b) \\ P_b^{\min} \leq P_b \leq P_b^{\max} & (7.10c) \\ T_m^{\min} \leq T_m \leq T_m^{\max} & (7.10d) \\ \omega_m^{\min} \leq \omega_m \leq \omega_m^{\max} & (7.10e) \end{cases}$$

The constraints in Eq. (7.10) enforce physical limits of the batteries and electric drivetrain by bounding the state of charge within $[\text{SOC}^{\min}, \text{SOC}^{\max}]$, restricting battery power to the allowable range $[P_b^{\min}, P_b^{\max}]$, and limiting the electric-machine torque to $[T_m^{\min}, T_m^{\max}]$ to ensure feasible and safe operation.

All lower-level MPC problems are solved at each control step using the SQP algorithm. Depending on the powertrain configuration, the optimization variables are the diesel engine torque T_e , the engine-motor torque pair T_e and T_m in P2/PHEV, or the motor torque T_m in BEV configuration. At each sampling instant k , SQP computes the optimal control sequences over the control horizon N_c , e.g., $\mathbf{u}(k) = [u(k|k), u(k+1|k), \dots, u(k+N_c-1|k)]$, by minimizing the corresponding cost function while satisfying all system constraints. Consistent with the receding-horizon principle, only the first control move $u(k|k)$ (i.e., $T_e(k|k)$, $(T_e(k|k), T_m(k|k))$, or $T_m(k|k)$ depending on the configuration) is implemented. At the next sampling instant, the optimization problem is solved again using the updated system states.

7.3 Numerical Evaluation and Discussion

7.3.1 Simulation Setup

To evaluate the designed hierarchical control system under realistic and varying driving conditions, two-truck homogeneous platoons (i.e., identical powertrain configurations for the lead and follower trucks) are simulated using a driving cycle derived from Alberta Highway 2, spanning from Edmonton to Calgary. This corridor includes hilly terrain segments, with road grades ranging from -5.1% to 5.6%, providing a representative profile for real-world long-haul operations. The maximum speed along the route is 110 km/h, reflecting typical highway driving conditions encountered by heavy-duty trucks in Alberta, Canada.

Environmental wind conditions are incorporated to capture realistic aerodynamic loading. Wind measurements are obtained from the mobile weather station mounted on the lead truck. A representative wind profile is used in the simulations, with wind speed varying from 6 to 32 km/h and wind directions ranging from 12° to 65° relative to the vehicle heading. The platoon trucks with electrified configurations of P2-HEV, PHEV and BEV are simulated separately under the identical driving and

environmental conditions and their performance is compared against the conventional diesel baseline. The key component specifications of the electrified configurations in this study are summarized in Table 7.1. All the simulations are conducted on a PC equipped with an Intel® Core™ i7-9700 CPU @ 3.00 GHz and 32 GB RAM, using the real-world data collected during the on-road cooperative truck platooning trials. No computational or feasibility issues were observed throughout the simulations.

Table 7.1: Key component specifications of the hybrid configurations in this study.

Component	Parameter	P2	PHEV	BEV
Engine	$P_{\max}$ (kW)	336	231	-
	$T_{\max}$ (N·m)	2273	1226	-
Electric Machine	$P_{\max}$ (kW)	200	240	500
	$T_{\max}$ (N·m)	1592	1910	3979
Battery	Type	Li-ion	Li-ion	Li-ion
	Capacity (kWh)	40	312	661

7.3.2 Simulation Results and Analyses

Hybrid Electric Truck Platooning

A two-truck homogeneous platoon with P2 hybrid powertrains is first simulated under the Alberta Highway 2 driving cycle at a large time gap of 5 sec. This case is used to verify the coordination effectiveness of the designed hierarchical control system under conditions where aerodynamic platooning benefits are expected to be minimal, such that any performance differences primarily reflect control-induced improvements in speed coordination and powertrain operation optimization. Given there is insignificant aerodynamic influence of platooning on the lead truck [34], it serves as the control vehicle for performance comparison within the platoon. The simulation results are presented in Figure 7.5.

Figure 7.5(a) presents the speed profiles of the lead and follower trucks, indicating that the follower accurately tracks the lead truck over the entire driving cycle. The

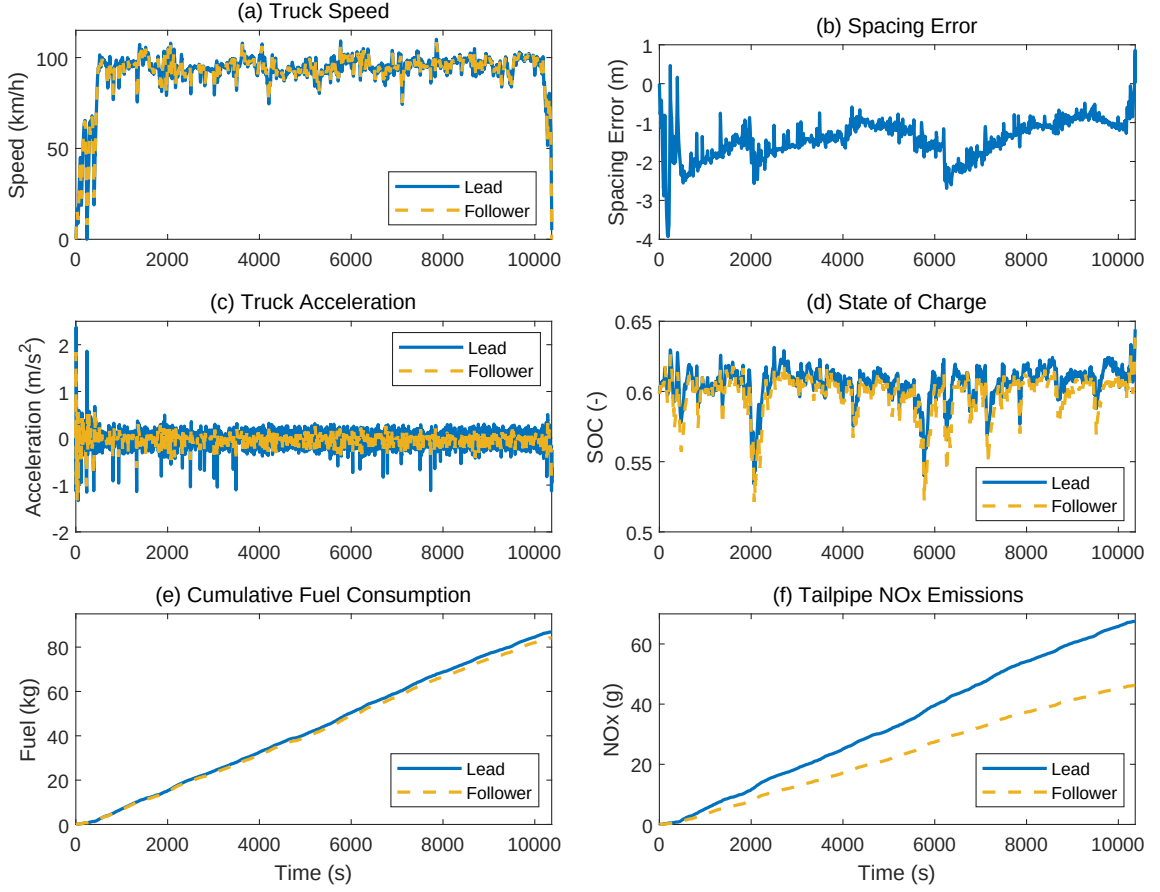


Figure 7.5: Results of truck platooning with P2 hybrid configuration under Alberta Highway 2 driving cycle.

speeds vary between 0 and 110 km/h, with a maximum speed deviation of 2.4 km/h between the two vehicles. Furthermore, the follower truck exhibits a smoother speed profile, which is supported by the acceleration curves shown in Figure 7.5(c). The lead truck's acceleration ranges from -1.3 to 2.4 m/s², while the follower's acceleration varies within -1.3 to 1.8 m/s², demonstrating the controller's ability to generate smooth speed trajectories. Figure 7.5(b) further demonstrates that the inter-vehicle spacing error remains well within the 5 m safety buffer throughout the entire operation, thereby confirming the effectiveness of the controller in ensuring safe platooning behavior.

As shown in Figure 7.5(d), both trucks start with an initial SOC of 60%. The lead truck's SOC varies between 53.4% and 63.9%, whereas the follower truck's SOC

ranges from 52.1% to 64.4%. By the end of the route, the SOC of the follower truck remains 0.5% higher than that of the lead truck, suggesting enhanced energy recuperation through regenerative braking under the proposed coordination strategy and more effective utilization of the battery through optimized torque-split management. Figure 7.5(e) presents the cumulative fuel consumption of the two trucks. The lead truck consumes approximately 3 kg more diesel over the selected route. When the equivalent fuel consumption associated with battery energy, calculated using Eq. (7.8d), is taken into account, the follower truck achieves a total energy saving of 3.6% relative to the lead truck. These results demonstrate that the developed control system improves energy conversion efficiency by optimally coordinating the torque contribution of the engine and electric motor, thereby reducing overall fuel consumption.

To better illustrate this effect, Figure 7.6 compares the operating-point distributions of the lead and follower trucks on the diesel-engine and electric-motor efficiency maps over the selected drive cycle. The follower truck's operating points shift toward higher-efficiency regions for both the engine and the electric motor. This trend indicates that the smoother speed trajectory enabled by the proposed coordination strategy allows the follower truck to operate in regions of the engine and motor efficiency maps that maximize energy conversion efficiency and reduce the reliance on the internal combustion engine. For example, the follower truck is propelled approximately 3.3% more by the electric motor than the lead truck, further enhancing its energy recovery. Moreover, the follower's engine operating points are 2.5% more concentrated in the highest-efficiency region (with the brake-specific fuel consumption (BSFC) less than 190 g/kW·h), while its electric-motor operating points are 5.2% more concentrated in the corresponding highest-efficiency region. These operational advantages contribute to the follower truck's improved energy conversion efficiency and fuel savings.

Figure 7.5(f) compares the tailpipe NO_x emissions for both trucks. The follower truck emits 21 g less NO_x than the lead truck, corresponding to a 31.6% reduction

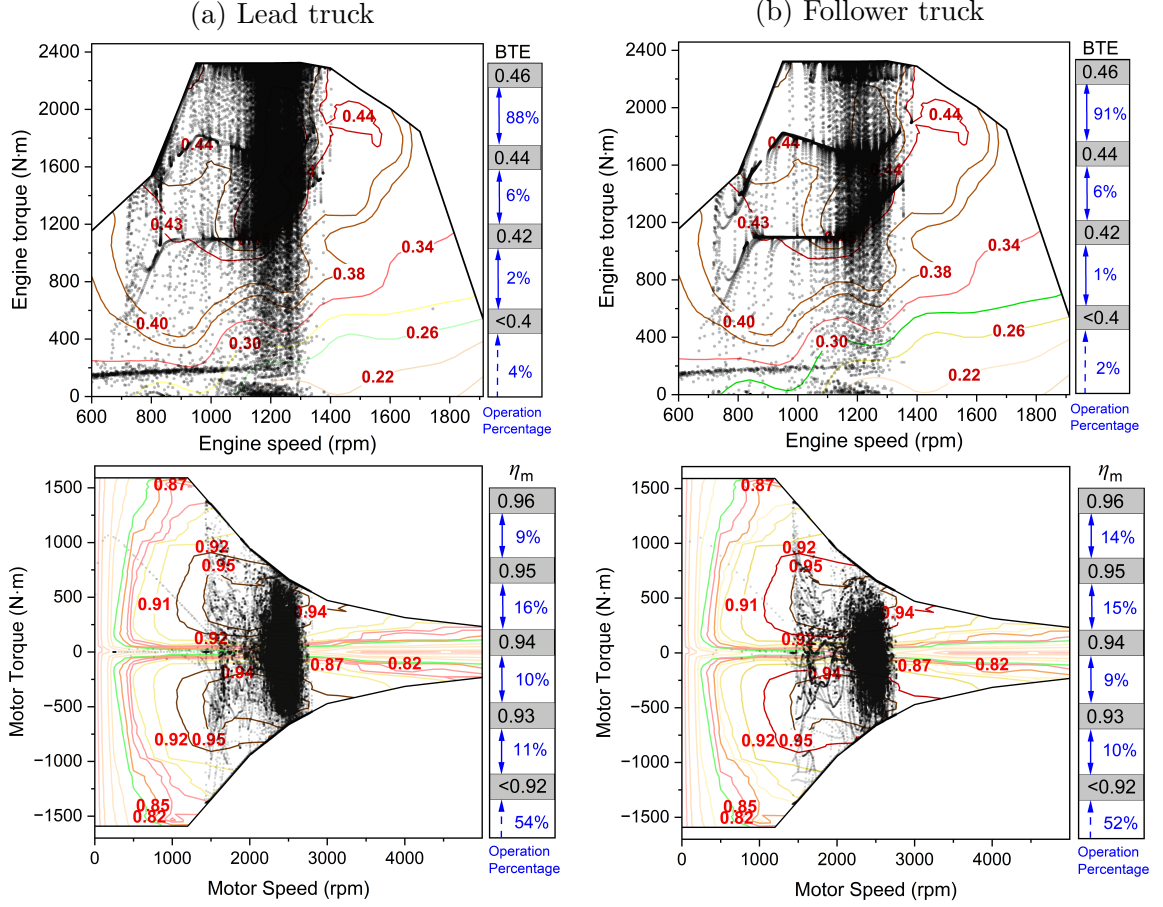


Figure 7.6: Operating points on diesel engine and electric motor efficiency maps over Alberta Highway 2 driving cycle for the lead and follower trucks with P2 hybrid configurations. The percentage tables shown beside each sub-figure indicate the fraction of engine/motor operating time spent within the corresponding ranges. BTE stands for brake thermal efficiency.

in total tailpipe NO_x emissions. This improvement is primarily attributed to the smoother engine torque profile produced by the proposed control strategy, which reduces transient operation that is typically associated with elevated NO_x formation. To quantitatively evaluate torque smoothness, a smoothness index is defined as the integral of the squared second-order derivative of the torque profile, normalized by the square of the maximum torque ($\int_{t_1}^{t_2} C''(t)^2 dt / C_{max}^2$) [160]. Lower values indicate smoother engine operation. The computed smoothness index for the lead truck is 153, while that of the follower truck is only 11, confirming that the follower ex-

hibits a significantly smoother torque profile and, consequently, lower transient NO_x emissions.

Plug-In Hybrid Electric Truck Platooning

A two-truck homogeneous platoon with PHEV hybrid configurations is then simulated under the identical driving and environmental conditions. The results are presented in Figure 7.7. Figures 7.7(a) and 7.7(b) demonstrate that the follower truck accurately tracks the lead truck's speed trajectory while maintaining safe spacing throughout the route. As indicated by the acceleration profiles in Figure 7.7(c), the follower truck exhibits smoother longitudinal motion, further confirming the coordination effectiveness of the proposed control framework in coordinating platoon operation.

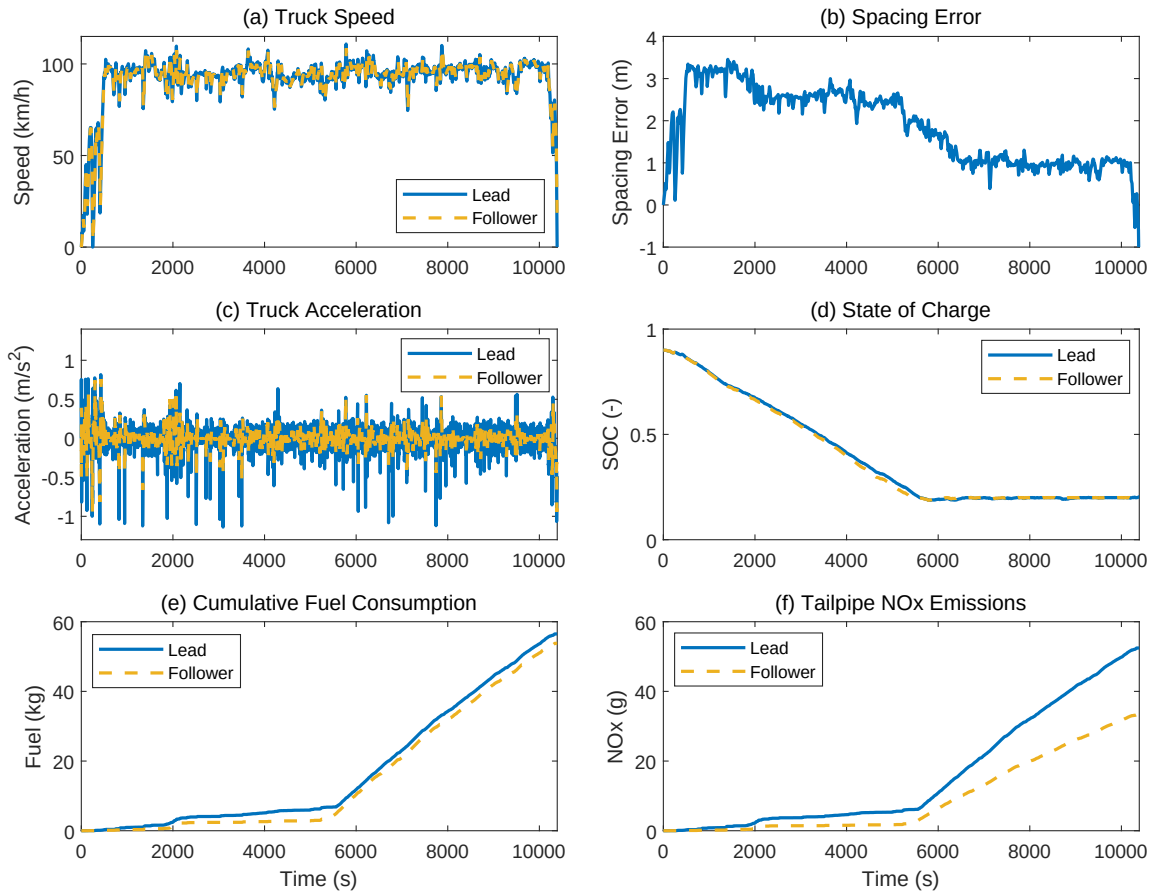


Figure 7.7: Results of truck platooning with PHEV hybrid configuration under Alberta Highway 2 driving cycle.

Both trucks start with an initial SOC of 90% and experience a gradual depletion to 20%, as shown in Figure 7.7(d). During this depletion phase, the follower truck leverages its battery more extensively to provide electric assistance under the varying operating conditions of Alberta Highway 2. This behavior is consistent with the coordination mechanism: the follower can anticipate upcoming torque demands and generate a smoother speed and traction-power trajectory, thereby reducing unnecessary acceleration–braking cycles and enabling more efficient torque-split decisions.

This effect is further supported by Figure 7.8, which compares the operating-point distributions on the diesel engine and electric motor efficiency maps for both the lead and follower trucks. Relative to the lead truck, the follower’s powertrain operation shifts toward higher-efficiency regions for both power sources. Specifically, the proposed control system increases the fraction of operating points in the highest-efficiency region ($\text{BSFC} < 190 \text{ g/kW}\cdot\text{h}$) by 6.7% and increases the reliance on the electric machine by 7.9% compared with the lead truck. These shifts indicate that the follower more frequently allocates traction demand to the higher-efficiency electric path and avoids inefficient engine operating zones, which directly contributes to the reduced diesel fuel consumption observed in Figure 7.7(e). Moreover, the follower’s electric-machine operating points are 8.5% more concentrated in the highest-efficiency region than those of the lead truck, implying improved utilization of high-efficiency motor operating zones and reduced electrical losses for a given level of electric assistance. Consequently, the follower truck achieves a 5.8% total energy saving over the route.

In addition to reduced engine usage, the follower exhibits substantially attenuated engine-torque transients, reflecting smoother torque delivery enabled by the coordination strategy. For instance, the computed smoothness index of the follower truck is 7, compared with 171 for the lead truck. This reduction in torque fluctuations mitigates rapid fueling and air-path excursions that are known to elevate NO_x formation [162], thereby contributing to the 36.2% reduction in tailpipe NO_x emissions shown in Figure 7.7(f).

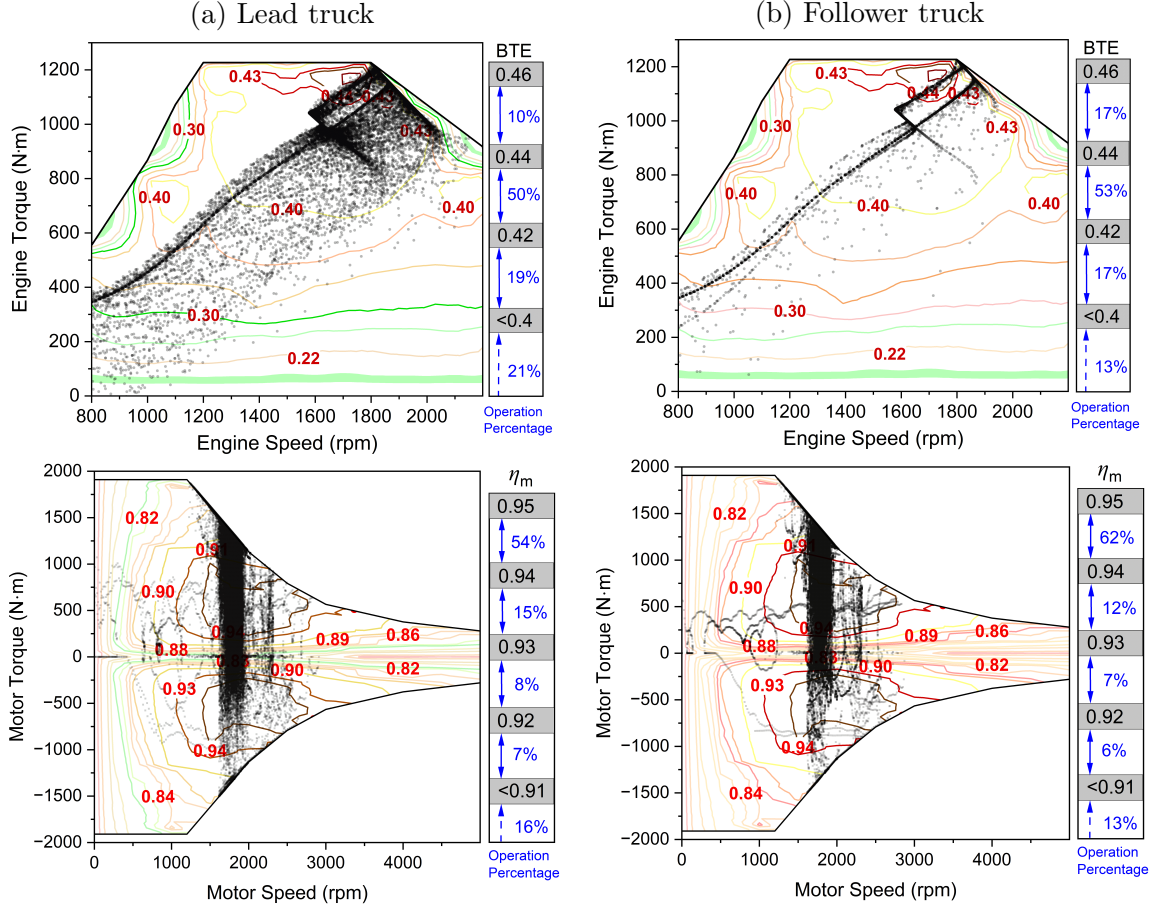


Figure 7.8: Operating points on diesel engine and electric motor efficiency maps over Alberta Highway 2 driving cycle for the lead and follower trucks with PHEV hybrid configurations. The percentage tables shown beside each sub-figure indicate the fraction of engine/motor operating time spent within the corresponding ranges. BTE stands for brake thermal efficiency.

Electric Truck Platooning

A two-truck homogeneous platoon with battery electric powertrains is simulated under the same driving and environmental conditions, and the corresponding results are shown in Figure 7.9. Figure 7.9(a) indicates that the follower accurately tracks the lead truck's speed trajectory, while Figure 7.9(b) indicates that the inter-vehicle spacing error remains within the prescribed safety buffer over the entire route, and the maximum spacing error is only 2.6 m. This peak error is smaller than that observed for the other configurations, which is attributed to the faster torque response

of the electric drivetrain and the absence of engine dynamics.

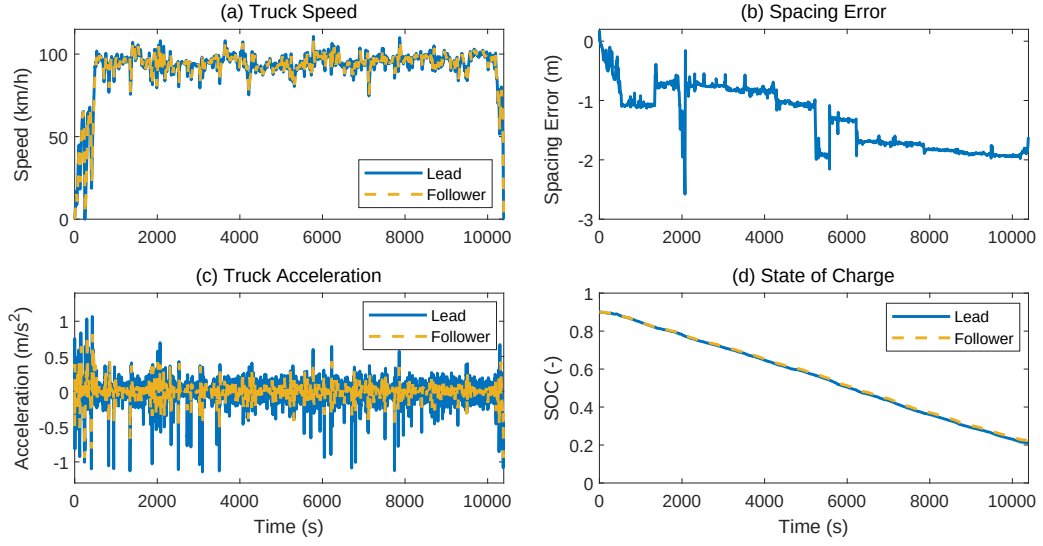


Figure 7.9: Results of truck platooning with BEV configuration under Alberta Highway 2 driving cycle.

As illustrated in Figure 7.9(c), the follower truck exhibits smoother longitudinal motion than the lead truck, reflecting the effectiveness of the proposed coordination strategy in minimizing unnecessary speed fluctuations. The follower's smoother motion results in improved energy conversion efficiency, as it can avoid sudden braking and acceleration events, contributing to overall energy savings. From a powertrain perspective, Figure 7.10 further shows that the follower's electric motor operates more frequently in the highest-efficiency region than that of the lead truck (67% vs 60%). In addition, the follower truck spends 3.9% less time in braking than the lead truck. The combined effect of a higher operating percentage in high-efficiency electric regions and a shorter braking duration contributes to a 3.1% energy saving, as reported in Figure 7.9(d).

Truck Platooning Performance across Powertrains

Lastly, to span the representative inter-vehicle spacings relevant to real highway operations, additional simulations are conducted under time gaps ranging from 1 to 4 sec. Table 7.2 compares the homogeneous-platoon performance of the P2 hybrid,

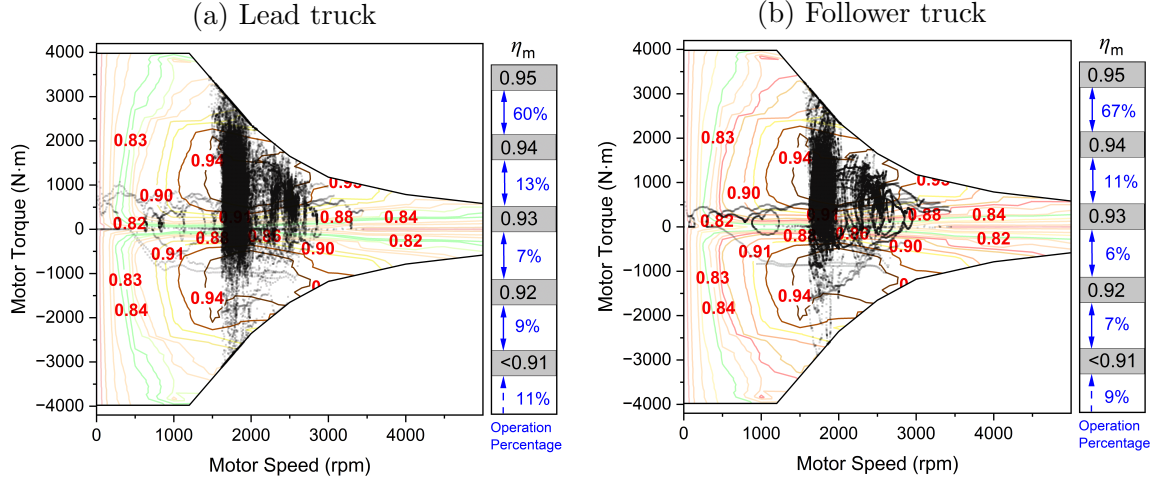


Figure 7.10: Operating points on electric motor efficiency maps over Alberta Highway 2 driving cycle for the lead and follower trucks with BEV configurations. The percentage tables shown beside each sub-figure indicate the fraction of motor operating time spent within the corresponding ranges.

plug-in hybrid, and battery electric configurations against the diesel baseline under the selected time gaps, reporting energy savings and tailpipe NO_x reductions, along with the engine on percentage for the hybrid cases. For the diesel baseline, energy savings remain modest, reaching a maximum of 8.1% at a 1-sec time gap, primarily driven by reduced aerodynamic drag at small spacings. Notably, the follower diesel truck achieves remarkable NO_x reductions (27.5%-32.6%), suggesting that the proposed control strategies can mitigate engine transients that are closely associated with elevated NO_x formation. Overall, all electrified configurations consistently outperform the conventional diesel system in both energy and emissions performance, with the benefits becoming more pronounced at smaller time gaps. As the time gap decreases, aerodynamic coupling strengthens and the attainable savings typically increase, whereas at larger time gaps the diminished aerodynamic benefit is partly offset by greater opportunity for coordinated speed smoothing.

Hybridization introduces additional degrees of freedom via motor assist, regenerative braking, and flexible engine operation. As a result, the P2 hybrid truck platoon

Table 7.2: Platooning performance comparison of different powertrain configurations under 1–5 sec time gaps. Energy saving and NO_x reduction are reported for the follower truck relative to the lead truck.

Powertrain Configuration	Time gap (sec)	Engine on percentage Lead vs. Follower (%)	Energy saving (%)	NO _x reduction (%)
Diesel	1	100 vs. 100	8.1	32.6
	2	100 vs. 100	6.4	31.8
	3	100 vs. 100	5.0	30.9
	4	100 vs. 100	3.8	28.9
	5	100 vs. 100	2.8	27.5
P2 Hybrid	1	86 vs. 83	9.6	35.9
	2	88 vs. 85	7.8	34.7
	3	89 vs. 87	6.2	33.7
	4	92 vs. 90	4.8	32.6
	5	94 vs. 93	3.6	31.6
Plug-in Hybrid	1	54 vs. 48	12.4	40.9
	2	55 vs. 50	10.4	39.5
	3	56 vs. 52	8.7	38.2
	4	57 vs. 54	7.2	37.1
	5	59 vs. 56	5.8	36.2
Battery Electric	1	–	9.0	–
	2	–	7.2	–
	3	–	5.6	–
	4	–	4.2	–
	5	–	3.1	–

outperforms the diesel platoon across all examined time gaps, with total energy savings increasing from 3.6% (5 sec) to 9.6% (1 sec). Meanwhile, the follower truck's engine on percentage decreases from 93% at 5 sec to 83% at 1 sec, indicating greater reliance on the electric motor as shorter spacing leads to more frequent transient power demands. Tailpipe NO_x reductions remain consistently high (31.6%-35.9%), reflecting the combined effects of smoother engine torque trajectories and partial load shifting to the electric path.

Among all powertrain configurations, the plug-in hybrid follower truck exhibits the largest performance gains. Energy savings increase from 5.8% at 5 sec to 12.4% at 1 sec, while the engine on percentage drops from 56% to 47%, demonstrating extensive electric propulsion enabled by grid-recharged energy. Correspondingly, it achieves the greatest tailpipe NO_x mitigation, peaking at 40.9% at 1 sec and remaining above 36% across all time gaps. These results indicate that higher battery capacity and greater electric-propulsion authority not only reduce fuel use but also suppress NO_x-intensive engine transients by reducing both the magnitude and frequency of aggressive engine torque excursions.

The BEV configuration also benefits from platooning, with energy savings increasing from 3.1% at 5 sec to 9% at 1 sec. Compared to hybrids, the incremental benefit is slightly smaller at BEVs, which is consistent with the BEV drivetrain operating over a broad high-efficiency region and therefore offering less headroom for further energy conversion efficiency improvement. As a result, BEV platooning gains are dominated by aerodynamic drag reduction and coordinated speed smoothing that reduces unnecessary braking events and improves the utilization of regenerative braking.

In summary, Table 7.2 highlights two consistent trends: (i) decreasing time gap generally strengthens energy savings due to increased aerodynamic coupling, and (ii) increasing electrification level amplifies both energy savings and NO_x reductions for platoons with hybrid electric powertrains through smoother engine operation and reduced ICE utilization in hybrid electric trucks. Importantly, even at regulation-

compliant time gaps (i.e., ≥ 2 sec), electrified platoons using the designed control system retain meaningful benefits, supporting the feasibility of simultaneous energy and emissions improvements under realistic long-haul operating constraints.

7.4 Summary of Chapter

This chapter developed a two-level hierarchical energy and emissions management control framework for homogeneous truck platoons spanning multiple powertrain electrification levels, including conventional diesel, P2 hybrid, plug-in hybrid, and battery-electric configurations. The upper-level controller, formulated using model predictive control, enforced safe following distances while coordinating platoon motion to generate smooth speed trajectories. The lower-level controller also used model predictive control to track the upper-level torque demand while minimizing energy consumption; for hybrid configurations, it further reduced tailpipe NO_x emissions by shaping engine operating conditions and suppressing transient torque excursions.

High-fidelity simulations on the Alberta Highway 2 driving cycle, incorporating realistic road grades and measured wind conditions, showed that the proposed framework improved follower-vehicle operation through coordinated torque planning. Specifically, the controller reduced unnecessary deceleration–acceleration cycles and enhanced regenerative energy recovery, yielding meaningful benefits even at regulation-compliant time gaps where aerodynamic drag reduction was limited. For example, at a 5-sec time gap, the plug-in hybrid follower achieved up to 5.8% equivalent energy saving relative to the lead truck, highlighting the contribution of control-enabled speed smoothing and powertrain-aware energy management beyond aerodynamic effects. In parallel, substantial tailpipe NO_x mitigation was obtained for hybrid-electric platoons; a representative case achieved a 36.2% reduction, consistent with smoother engine torque trajectories and reduced engine-on utilization enabled by electric propulsion support.

Across the evaluated time gaps, higher degrees of electrification systematically am-

plified platooning benefits for hybrid powertrains. In particular, plug-in hybrid operation leveraged grid-charged energy to increase electric-propulsion authority, which simultaneously improved energy conversion efficiency and suppressed NO_x -intensive engine transients. Battery-electric platoons also exhibited consistent energy benefits, primarily driven by aerodynamic interaction and coordinated speed smoothing, with additional gains attributable to improved utilization of regenerative braking.

Overall, these findings indicated that combining cooperative platoon coordination with powertrain-aware energy and emissions optimization provided a practical and scalable pathway to energy-efficient, low-emission long-haul freight operations.

PART V: Conclusions

Chapter 8

Conclusions and Future Work

8.1 Conclusions

This thesis designed and validated the cooperative truck platooning system for real-world platooning operations. The primary objectives were to (i) ensure safe platoon operation under environmental disturbances with varying winds and road grades, (ii) improve energy savings beyond what can be achieved by aerodynamic drag reduction alone under regulation-compliant time gaps, and (iii) mitigate tailpipe NO_x emissions by integrating powertrain dynamics into the platoon control. The main conclusions of this thesis are summarized below.

8.1.1 On-Road Cooperative Truck Platooning Trials

Canada's first cooperative truck platooning trials were conducted on public highways with commercial operations, and the results were presented in Chapter 2. The on-road cooperative truck platooning trials provided direct evidence of how real-world disturbances and operational variability shape platoon performance. First, the trials confirmed the feasibility of maintaining safe platoon operation on public highways at regulation-compliant time gaps, while also demonstrating that achievable fuel-saving benefits are highly sensitive to the surrounding environment. In particular, wind conditions and road geometries materially influenced the longitudinal response of the platoon.

Second, the trials revealed that lateral alignment is a critical yet under-managed factor in practical platooning. Crosswinds produced non-negligible lateral offsets between the lead and follower trucks, partially displacing the follower from the lead truck’s wake and reducing the expected aerodynamic shielding benefits. These lateral offsets were generally larger under high wind conditions and could persist even when lane keeping was maintained, highlighting that lane centering alone is insufficient to preserve inter-vehicle wake alignment.

Third, the measured powertrain behavior indicated that inadequate coordination between the platooning controller and the vehicle powertrain system can induce oscillatory torque responses. Such transient torque excursions are undesirable from both an efficiency and an emissions standpoint: they increase fuel consumption through unnecessary acceleration–deceleration cycles and can elevate tailpipe NO_x emissions due to transient engine operation.

8.1.2 Truck Platoon Modelling and Validation

In Chapter 3, an integrated modelling framework for Class-8 truck platoons was developed and validated to support controller design and high-fidelity simulation. The modelling framework unified (i) longitudinal platoon dynamics and driveline power balance, (ii) a control-oriented tractor–trailer lateral dynamics model, (iii) aerodynamic interaction mechanisms induced by both inter-vehicle spacing and lateral alignment, and (iv) fuel and tailpipe-emissions estimation modules that are compatible with transient, real-world operation.

To improve tailpipe NO_x prediction fidelity under transient driving, this thesis further established the strong dependence of tailpipe NO_x emissions on engine-torque transient behavior and leveraged this insight to develop a data-driven model. The LSTM was trained using the real-world trial dataset to capture transient aftertreatment behavior beyond conventional map-based approaches, thereby enabling more credible evaluation of control strategies that intentionally shape torque trajectories.

The developed models were validated against independent datasets from the on-road trials by applying measured control inputs and incorporating recorded road grade and wind conditions. The validation results demonstrated close agreement between simulated and measured trajectories and outcomes, with low errors in speed, longitudinal spacing, and lateral offset, as well as small deviations in cumulative fuel consumption and tailpipe NO_x emissions. Overall, these results confirmed that the modelling framework can reliably reproduce coupled longitudinal–lateral platoon dynamics and provide trustworthy energy and emissions estimates under realistic disturbances, forming a high-fidelity foundation for the controller designs.

8.1.3 Cooperative Platoon Control under Disturbances

In Chapters 4 and 5, this thesis developed hierarchical cooperative control systems for large-spacing truck platooning under two dominant real-world disturbance classes: road-grade variations and varying winds. Explicitly modelling and compensating these disturbances enables performance gains that persist even when aerodynamic wake benefits are weak at regulation-compliant time gaps. By coordinating speed trajectories to suppress unnecessary deceleration–acceleration cycles and by shaping engine-torque trajectories to avoid abrupt transients, the proposed disturbance-aware controllers improve energy conversion efficiency and mitigate NO_x intensive operating regimes. These control-induced benefits complement aerodynamic coupling, thereby improving follower-vehicle operation under realistic highway conditions.

Road grade primarily affects platooning through longitudinal load disturbances that change tractive and braking demands along the route. Without anticipatory coordination, grade transitions can propagate downstream as amplified speed oscillations and torque excursions, degrading fuel economy and increasing emissions. The developed hierarchical control strategy counteracts these effects by moderating follower speed changes and smoothing torque commands during uphill and downhill segments. Numerical evaluations on Alberta Highway 2 demonstrate that the fol-

lower truck achieves fuel savings of up to 6.6% and tailpipe NO_x reductions of up to 23.2% under grade-varying conditions, confirming that coordinated speed profiling and powertrain-aware optimization deliver measurable benefits beyond drag reduction.

Wind disturbances affect platooning through time-varying aerodynamic loads that directly perturb drag. The designed controller in this thesis mitigates these effects by coordinating follower response and penalizing aggressive torque changes, thereby stabilizing speed tracking while reducing powertrain transients. Under windy conditions, simulations show fuel savings of up to 5.5% and tailpipe NO_x reductions of up to 32%, demonstrating that wind-aware cooperative platoon control can maintain meaningful energy and emissions benefits.

8.1.4 Integrated Longitudinal–Lateral Platooning Control

This thesis further demonstrated that explicitly coordinating longitudinal and lateral control is essential for preserving platooning benefits in real-world operation, where crosswinds can introduce lateral misalignment and degrade aerodynamic coupling. In Chapter 6, a coordinated platooning control framework was developed that integrated longitudinal spacing regulation and lateral alignment regulation within an NMPC-based formulation. Simulation studies on Alberta Highway 2 using measured wind profiles and realistic road grades showed that the coordinated controllers simultaneously maintained safe longitudinal spacing and minimized lateral offsets, while producing smoother speed and torque trajectories.

Across regulation-compliant time gaps, the integrated framework achieved consistent improvements over the on-road CTPS baseline. In particular, fuel savings increased by up to 3.2% relative to CTPS, and tailpipe NO_x outcomes improved by up to 50.4%, shifting from NO_x increases under CTPS to sustained NO_x reductions under the proposed NMPC framework. These gains were primarily enabled by disturbance-aware coordination that suppresses unnecessary deceleration–acceleration cycles, pre-

serves wake alignment through tighter lateral control, and mitigates NO_x -intensive engine torque transients through optimized powertrain-aware longitudinal control.

8.1.5 Electrified Truck Platooning across Powertrain Architectures

In Chapter 7, this thesis further extended platooning control to electrified Class-8 powertrains, where additional actuation degrees of freedom (motor assist, regenerative braking, and torque-split flexibility) create new opportunities for energy optimization and emissions reduction. A two-level hierarchical energy and emission management framework was developed and evaluated for homogeneous two-truck platoons spanning conventional diesel, P2 hybrid, plug-in hybrid, and battery-electric configurations. The upper-level MPC coordinated platoon motion to maintain safety while generating smooth speed trajectories, and the lower-level MPC optimized powertrain actuation to track the demanded torque while minimizing energy consumption; for hybrid configurations, the controller additionally reduced tailpipe NO_x by mitigating engine transients and shifting load to the electric path when beneficial.

Simulation results under Alberta Highway 2 driving showed that electrified powertrains systematically amplified platooning benefits. Plug-in hybrid platoons achieved the largest combined gains, reflecting higher battery capacity and increased electric-propulsion authority that reduced both fuel use and NO_x intensive engine transients. P2 hybrid platoons also consistently outperformed the diesel baseline through improved regenerative energy recovery and more efficient engine operating-point utilization. Battery-electric platoons achieved consistent energy reductions driven primarily by aerodynamic interaction, coordinated speed smoothing, and improved utilization of regenerative braking. Importantly, the thesis demonstrated that meaningful benefits persisted at larger, regulation-compliant time gaps where aerodynamic drag reduction is limited, highlighting coordinated speed profiling and powertrain-aware energy management as dominant mechanisms in these regimes.

8.2 Future Work

Potential avenues for extending this work include:

- Implement the designed control systems on instrumented Class-8 trucks and conduct highway experiments to validate platoon safety, energy-saving performance, and tailpipe NO_x outcomes.
- Investigate the combined effects of truck load, road grade, and wind in practical platooning applications by integrating intelligent tires for vehicle mass estimation, GPS/RTK and digital-map information for road grade identification, and on-board sensing or sensor-fusion-based approaches for wind estimation.
- Incorporate transient dynamics of the SCR system into the emission analysis and control framework could improve the accuracy of tailpipe NO_x prediction and enable more comprehensive assessment of its behavior in truck platooning.
- Enhance the practical feasibility of the hierarchical two-MPC framework. In particular, slack variables may be introduced to soften selected constraints and avoid infeasibility under actuator limits, powertrain limitations, and uncertain operating conditions.
- Quantify uncertainty propagation within the proposed control framework and assess its effects on closed-loop performance and robustness.
- Evaluate upstream emissions associated with electricity generation, transmission, and charging in electrified truck platooning using a well-to-wheel or life-cycle assessment framework.
- Extend the proposed frameworks toward higher SAE levels of automation, such as SAE Level 4 truck platooning, where vehicles operate without driver intervention. This future direction will require AI-enabled integrated perception–planning–control validation for unattended highway operation.

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Appendix A: Ph.D. Publications

A.1 Peer Reviewed Journal Papers

A.1.1 Published Journal Papers

1. **L. Jiang**, and M. Shahbakhti, “Optimizing large-spacing truck platooning in windy environments: Minimizing fuel consumption and tailpipe nitrogen oxides emissions”, *IEEE/ASME Transactions on Mechatronics*, early access, pp. 1–11, 2026.
2. **L. Jiang**, and M. Shahbakhti, “Reducing fuel consumption and tailpipe nitrogen oxides emissions through large-spacing truck platooning”, *Transportation Research Part D: Transport and Environment*, vol. 147, 104917, 2025.
3. **L. Jiang**, J. Kheyrollahi, C.R. Koch, and M. Shahbakhti, “Cooperative truck platooning trial on Canadian public highway under commercial operation in winter driving conditions”, *Proc IMechE Part D: Journal of Automobile Engineering*, vol. 239, no. 7, pp. 2272-2284, 2024.
4. **L. Jiang**, J. Kheyrollahi, C.R. Koch, and M. Shahbakhti, “Performance analysis of cooperative truck platooning under commercial operation during Canadian winter season”, *SAE International Journal of Connected and Automated Vehicles*, vol. 7, no. 2, pp. 183-198, 2024. (**High-Impact Article**)
5. S. Shahpouri, **L. Jiang**, C.R. Koch, and M. Shahbakhti, “Tailpipe NO_x emissions modeling of a heavy duty diesel truck using deep learning methods”, *SAE International Journal of Engines*, vol. 18, no. 8, pp. 915-939, 2025.

6. S. Moghadasi, Y. Long, **L. Jiang**, S. Munshi, G. McTaggart-Cowan, and M. Shahbakhti, “Design and performance analysis of hybrid electric class 8 heavy-duty regional-haul trucks with a micro-pilot natural gas engine in real-world highway driving conditions”, *Energy Conversion and Management*, vol. 309, 118451, 2024.

A.1.2 Submitted Journal Paper

1. **L. Jiang**, and M. Shahbakhti, “Coordinated longitudinal-lateral control for truck platooning: Minimizing fuel consumption and NO_x emissions”, 28 pages, submitted to *ISA Transactions*, under review, 2026.
2. **L. Jiang**, and M. Shahbakhti, “Platooning electrified trucks: Benefits of energy and emission optimization”, 32 pages, submitted to *Energy*, under review, 2026.

A.2 Peer Reviewed Conference Proceeding Papers

1. **L. Jiang**, and M. Shahbakhti, “Large-spacing truck platooning under windy conditions”, *2025 American Control Conference (ACC)*, 6 pages, July 8-10, 2025, Denver, CO, USA.
2. **L. Jiang**, and M. Shahbakhti, “Fuel-efficient large-spacing truck platooning in windy conditions considering longitudinal and lateral motions”, *2025 International Conference on Intelligent Transportation and Future Mobility (ITFM)*, 6 pages, SAE Paper No. 2025-99-0419, April 11-13, 2025, Guilin, Guangxi, China.
3. **L. Jiang**, D. Gordon, and M. Shahbakhti, “Fuel-efficient control system design for cooperative truck platooning at large separation distances”, *27th IEEE International Conference on Intelligent Transportation Systems (ITSC)*, 6 pages, September 24-27, 2024, Edmonton, AB, Canada.

4. **L. Jiang**, and M. Shahbakhti, “Cooperative truck platooning on Canadian public roads during a winter season”, *28th IAVSD International Symposium on Dynamics of Vehicles on Roads and Tracks (IAVSD)*, 9 pages, August 21-25, 2023, Ottawa, ON, Canada.
5. A. Yasami, T. Mohamadali, **L. Jiang**, H. Abediasl, C.R. Koch, and M. Shahbakhti, “Data-efficient vehicular fuel rate estimation via transfer learning using a nonlinear-autoregressive-exogenous neural network”, *11th Symposium on Advances on Automotive Control (AAC)*, 6 pages, June 16-18, 2025, Eindhoven, North Brabant, Netherlands.

A.3 Technical Presentations (Refereed Abstracts)

1. **L. Jiang**, J. Kheyrollahi, C.R. Koch, and M. Shahbakhti, “Cooperative truck platooning trials on Canadian public roads under a commercial operation in winter driving conditions”, *Canadian Society of Mechanical Engineers (CSME) International Congress 2022*, June 5-8, 2022, Edmonton, AB, Canada.
2. Y. Long, **L. Jiang**, S. Moghadasi, S. Munshi, G. McTaggart-Cowan, and M. Shahbakhti, “Performance analysis of a heavy-duty hybrid-electric truck with a micro-pilot natural gas engine in real-world highway driving conditions”, *Canadian Society of Mechanical Engineers (CSME) International Congress 2022*, June 5-8, 2022, Edmonton, AB, Canada.

Appendix B: Research Source Files

B.1 Chapter 1 Source Files

Table B.1: Source files of chapter 1.

File name	File description
Fuel saving strategies on a tractor-trailer truck.pptx	Figure 1.1
Long combination vehicle.png	Figure 1.2
Truck platoon demonstration.png	Figure 1.3
Aftertreatment system.drawio	Figure 1.4
Five-layer control architecture.pptx	Figure 1.5
Three-layer control architecture.pptx	Figure 1.6
Schematic of the thesis organization.pptx	Figure 1.8

B.2 Chapter 2 Source Files

Table B.2: Source files of chapter 2.

File name	File description
Hardware components in the truck.pptx	Figure 2.1
Experimental sensor configuration.drawio	Figure 2.2
Test route and topographic map.eps	Figure 2.3
Test procedure.pptx	Figure 2.4

Continued on next page

Table B.2 (continued)

File name	File description
Schematic of the CTPS trials.JSON	Figure 2.5
Gross vehicle weight of trucks.xlsx	Figure 2.6
Elevation changes between Edmonton and Calgary.png	Figure 2.7
Apparent wind speeds.xlsx	Figure 2.8
Truck speed profiles.xlsx	Figure 2.9
Engine BSFC map.opju	Figure 2.10
Freight transportation specific fuel consumption.xlsx	Figure 2.11
Specific engine-out NO _x emissions.xlsx	Figure 2.12
Specific tailpipe NO _x emissions.xlsx	Figure 2.13
Engine torque and NO _x emissions.xlsx	Figure 2.14

B.3 Chapter 3 Source Files

Table B.3: Source files of chapter 3.

File name	File description
Truck model.slx	System modeling build
Truck parameters.m	System modeling parameters
Longitudinal forces on truck.pptx	Figure 3.1
Schematic of powertrain system.pptx	Figure 3.2
Fuel consumption map.m	Figure 3.3
LSTM model structure.pptx	Figure 3.4
Comparison of tailpipe NO _x emissions.m	Figure 3.5
Schematic of longitudinal platoon.vsd	Figure 3.6
Drag coefficient vs spacing.m	Figure 3.7
Lateral forces on truck.pptx	Figure 3.8

Continued on next page

Table B.3 (continued)

File name	File description
Schematic of n -truck platoon.vsd	Figure 3.9
Drag coefficient vs offset.m	Figure 3.10
Validation procedures.pptx	Figure 3.11
Validation results.fig	Figure 3.12

B.4 Chapter 4 Source Files

Table B.4: Source files of chapter 3.

File name	File description
Truck platoon controller (grade).slx	Controller design
Designed control architecture (grade).vsd	Figure 4.1
Block diagram (grade).ppt	Figure 4.2
Road grade specifications.xlsx	Figure 4.3
Measured wind speed and angle.xlsx	Figure 4.4
Results of 5-sec time gap (grade).fig	Figure 4.5
EGR system's dynamic response.xlsx	Figure 4.6
Results of 4-sec time gap (grade).fig	Figure 4.7
Results of 3-sec time gap (grade).fig	Figure 4.8
Results of 2-sec time gap (grade).fig	Figure 4.9
Coordination vs aero-reduction.xlsx	Figure 4.10
Impact of platooning.pptx	Figure 4.11

B.5 Chapter 5 Source Files

Table B.5: Source files of chapter 5.

File name	File description
Truck platoon controller (wind).slx	Controller design
Fuel consumption (wind vs no wind).xlsx	Figure 5.1
Schematic of the wind vectors.vsd	Figure 5.2
RWIS site map.pdf	Figure 5.3
Mobile weather station calibration results.xlsx	Figure 5.4
Designed control architecture (wind).vsd	Figure 5.5
Headwind and tailwind speed and angle.xlsx	Figure 5.6
Results of 4-sec time gap (headwind).fig	Figure 5.7
Comparison of operating points in BSFC map.opj	Figure 5.8
Results of 4-sec time gap (tailwind).fig	Figure 5.9

B.6 Chapter 6 Source Files

Table B.6: Source files of chapter 6.

File name	File description
Truck platoon controller (integrated).slx	Controller design
Designed control architecture (integrated).vsd	Figure 6.1
Block diagram (integrated).fig	Figure 6.2
Results of 5-sec time gap (integrated).fig	Figure 6.3
Results of 4-sec time gap (integrated).fig	Figure 6.4
Results of 3-sec time gap (integrated).fig	Figure 6.5
Results of 2-sec time gap (integrated).fig	Figure 6.6

B.7 Chapter 7 Source Files

Table B.7: Source files of chapter 7.

File name	File description
Truck platoon controller (electrified).slx	Controller design
Truck platooning classified by powertrains.pptx	Figure 7.1
Schematics of different powertrain configurations.pptx	Figure 7.2
Efficiency maps of the electric motors.opju	Figure 7.3
Designed control architecture (electrified).vsdx	Figure 7.4
P2-HEV platooning results.fig	Figure 7.5
Operating points on engine/motor efficiency maps (P2).opju	Figure 7.6
PHEV platooning results.fig	Figure 7.7
Operating points on engine/motor efficiency maps (PHEV).opju	Figure 7.8
BEV platooning results.fig	Figure 7.9
Operating points on motor efficiency maps (BEV).opju	Figure 7.10