

**Powertrain Systems Design for Heavy-Duty Trucks: Integrating Hybrid
Electric Technologies with Low- and Zero-Carbon Fuel Engines to
Reduce Fuel Consumption and Greenhouse Gas Emissions**

by

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Abstract

Class 8 heavy-duty (HD) trucks are major contributors to global greenhouse gas (GHG) emissions, accounting for 50%, 40%, and 25% of transportation-sector emissions in the United States, Canada, and Europe, respectively. This thesis proposes new powertrain architectures and creates a vehicle-level design framework to evaluate the combined effect of hybridization and the use of natural gas (NG) or Hydrogen (H_2) compression ignition (CI) engines on reducing fuel consumption and GHG emissions in Class 8 HD trucks.

First, a conventional baseline truck configuration is modeled and validated against collected on-road test data from a Class 8 Peterbilt diesel truck operated over a real-world regional-haul drive cycle. To create conventional truck models powered by low-carbon NG-CI or near-zero-carbon H_2 -CI engines, the baseline diesel engine is replaced with high-pressure direct injection (HPDI) NG or H_2 engines, along with corresponding modifications to the on-board fuel systems and storage tanks. These HPDI engines use a small amount of diesel ($< 5\%$ of total fuel energy) as a pilot for ignition while maintaining overall brake thermal efficiencies (BTEs) comparable to those of modern diesel engines. The results demonstrate 20.3% to 21.1% reductions in tailpipe CO_2 emissions for the conventional NG model and 97.4% to 98.5% reductions for the conventional H_2 model relative to the baseline diesel configuration over a real-world regional-haul drive cycle.

Second, three full hybrid electric powertrain configurations with high potential for reducing fuel consumption and GHG emissions in Class 8 HD trucks are modeled

and their energy management systems (EMSs) are designed. These hybrid powertrains include: 1) a pre-transmission parallel (P_2) hybrid, 2) a complex power-split hybrid, and 3) a series plug-in hybrid configuration. The hybrid powertrains and EMS strategies are tailored to meet the high torque demands of HD truck applications. A multi-stage component sizing framework is then created to optimize the sizes of the primary hybrid powertrain components, as well as to tune the key EMS-related variables. This framework aims to minimize net powertrain energy consumption over a real-world regional-haul drive cycle while maintaining propulsion system performance comparable to that of the conventional baseline model.

Third, the existing propulsion system performance criteria specifically established for conventional HD trucks are revised to address the testing requirements of hybridized Class 8 trucks equipped with a downsized engine. To achieve this, real-world road grade profiles of 15 steep highways worldwide frequently used by HD trucks are derived. The optimal component sizing of the conventional baseline and P_2 hybrid configurations is then refined using each newly identified steep highway as the gradeability criterion. The results demonstrate the feasibility of engine downsizing in the P_2 hybrid configuration while maintaining propulsion system performance comparable to that of the conventional baseline model and achieving 3.4% to 8.9% lower net powertrain energy consumption over a real-world regional-haul drive cycle.

Fourth, the hybrid electric powertrains are optimized for integration with different types of NG-CI engines, including a diesel-pilot premixed NG engine and technology-enhanced variants of the HPDI NG engine in both downsized and full-size versions. Technology packages are implemented to improve engine BTE, particularly at mid-to-high torque levels where the engines in hybrid configurations operate predominantly. The plug-in hybrid configuration with a downsized HPDI NG engine incorporating a turbo-compound system and a high compression ratio achieves a 56.9% reduction in tailpipe GHG emissions (CO_2 -equivalent), including CH_4 , relative to the conventional

diesel model over a highway drive cycle at full cargo load. These reductions are 27.3% for the power-split hybrid and 26.5% for the P₂ hybrid designs, both equipped with a full-size HPDI NG engine incorporating the same technology package.

Fifth, the P₂ hybrid electric powertrain is optimized for integration with a commercially available HPDI H₂ engine with a maximum BTE of $\sim 52\%$. New model is created to quantify the auxiliary load of the on-board H₂ compression system to maintain a 300 bar hydrogen injection pressure as on-board storage tanks are depleted. Hybridizing the HPDI H₂ engine results in a 17.1% reduction in diesel-equivalent fuel consumption relative to the conventional diesel baseline over a highway drive cycle at full cargo load. The corresponding tailpipe GHG emissions are reduced by 98.9% relative to the diesel baseline, enabling the hybridized configuration to satisfy the European Union (EU) zero-emission vehicle threshold of less than 3 g CO₂/tonne_{cargo}.km on a tank-to-wheel basis.

Preface

This thesis presents original work conducted by Sina Moghadasi. The research was conducted utilizing available computing systems and experimental equipment at the Energy Mechatronics Laboratory (EML) at the University of Alberta, led by Prof. Mahdi Shahbakhti. The conventional and hybrid electric truck models were constructed by the author in the MATLAB-based Autonomie[®] vehicle simulation platform.

The research conducted in this thesis was part of a collaborative project between the University of Alberta, Simon Fraser University (SFU), and the industrial partners, funded by the Natural Sciences and Engineering Research Council of Canada (NSERC). The primary focus of this thesis is to create a vehicle system-level design framework for modeling conventional and hybrid electric Class 8 truck configurations equipped with low- or zero-carbon-fueled CI engines, as illustrated in Figure 1. The contributions of the joint-project partners are summarized as follows:

- **Diesel engine map data:** The fuel consumption and emission maps of the Cummins X15 diesel engine in this thesis were created by the author. Dr. Luo Jiang, a former PhD student in the EML at the University of Alberta, contributed to the collection and analysis of the real-world test truck data used to develop these maps.
- **HPDI NG engine map data:** The fuel consumption and emission maps of the HPDI NG engines in this thesis were created by Dr. Navid Balazadeh-Meresh, a former PhD student at SFU, under the supervision of Prof. Gordon

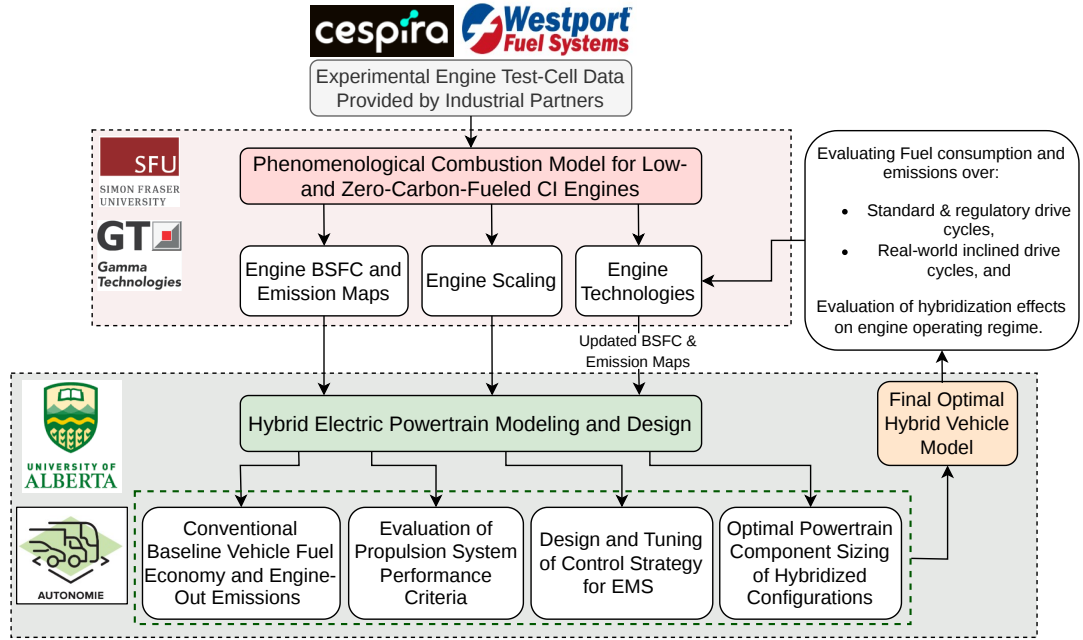


Figure 1: The major development process in the overarching project for designing hybrid electric Class 8 truck configurations equipped with efficient low- or zero-carbon-fueled internal combustion engines to reduce both fuel consumption and GHG emissions. The main focus of this thesis is specified in the flowchart.

McTaggart-Cowan. The experimental test-cell data used to develop the engine models in GT-SUITE[®] software were provided by Westport Inc.

- **HPDI H₂ engine map data:** The fuel consumption and emission maps of the HPDI H₂ engine used in this thesis were created by the author. The engine efficiency map, along with the contribution of diesel and H₂ fuel energy over different engine speeds and torques, was derived from a publicly available dataset provided by David Mumford from Cespira.
- **H₂ compressor map data:** The operating map of the on-board compression system used to maintain the 300-bar H₂ injection pressure in the HPDI H₂ engine was created by Mehdi Nikkhah-Koojehri, a PhD student at SFU, under the supervision of Prof. Gordon McTaggart-Cowan.

The content of Chapters 1 to 7 is fully or partially based on the peer-reviewed

journal papers [1–4] and forthcoming journal papers [5, 6] currently under review. As the lead or second author, I made substantial contributions to all publications referenced throughout this thesis. The specific contributions of co-authors to each paper are summarized below:

- In Chapter 1 [1], Dr. Navid Balazadeh-Meresht contributed to reviewing and organizing the sections related to CI engines advancements and technologies. In addition, Dr. Sandeep Munshi and Prof. Gordon McTaggart-Cowan provided valuable insights into structuring the evaluation of promising near-term pathways for decarbonizing the HD truck sector.
- In Chapter 2 [2, 3], Dr. Luo Jiang contributed to real truck data collection and analysis, and Dr. Sandeep Munshi and prof. Gordon McTaggart-Cowan provided valuable insights into the design of NG-fueled HD truck models.
- In Chapters 3 and 4 [2, 4], Prof. Gordon McTaggart-Cowan and Dr. Sandeep Munshi provided valuable suggestions regarding the design of the hybrid electric powertrain architectures and the development of the optimal component sizing framework.
- In Chapter 5 [3], Amirreza Yasami contributed to digitizing the grade–distance profiles of the identified real-world steep highways, and prof. Gordon McTaggart-Cowan provided valuable insights into classifying these grade profiles based on their steepness levels for gradeability performance assessment.
- In Chapter 6 [2, 4, 5], Dr. Yanxiang Long contributed to the development of fuel consumption and emission maps of the diesel-pilot premixed NG (micro-pilot) engine. Dr. Navid Balazadeh-Meresht provided the fuel consumption and emission maps of the technology-enhanced HPDI NG engines. In addition, Dr. Sandeep Munshi and Prof. Gordon McTaggart-Cowan provided valuable

insights into the development of hybrid electric truck configurations using NG-fueled engines.

- In Chapter 7 [6], Mehdi Nikkhah-Koojehri provided the operating map of the on-board hydrogen compressor for the HPDI H₂ engine. Prof. Gordon McTaggart-Cowan provided valuable insights into the development of the model used to quantify the auxiliary load of the H₂ compression system, as well as modifications to the fuel supply system model for the HPDI engine.

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First and foremost, I would like to express my sincere gratitude to my supervisor, Prof. Mahdi Shahbakhti, for his continuous guidance, unwavering support, and high standards throughout my PhD journey. His deep technical insight, thoughtful mentorship, and encouragement have been invaluable in shaping both the direction and quality of this work. I am truly grateful for the opportunities he provided to pursue meaningful research and to grow both academically and professionally.

I would like to sincerely thank my supervisory committee members, Prof. Gordon McTaggart-Cowan and Prof. Charles (Bob) Koch, for their valuable feedback, insightful discussions, and constructive suggestions. Their expertise and guidance significantly strengthened this work. I also thank Navid Balazadeh-Meresht, Amirreza Yasami, Luo Jiang, Yanxiang Long, and Mehdi Nikkhah-Koojehri for their contributions to our joint publications. Special thanks go to Dr. Sandeep Munshi and Dr. Muhammad Ezzat for sharing their valuable insights throughout my research.

I would like to express my appreciation to David Mumford and Dr. Jim Huang from Cespira for their valuable feedback on this research and for sharing their public presentations and technical data related to HPDI H₂ engine.

At the heart of this journey stands my family, my parents and my sister, to whom I owe my deepest gratitude. Their unwavering love, encouragement, and constant belief in me provided the strength to navigate challenges and persevere. This achievement would not have been possible without them.

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List of Symbols

Latin

$\Delta\text{SOC}_{\text{usable}}$	Usable SOC window (-)
Δt	Time step interval (s)
δ_{mass}	Rotational mass factor (-)
$\dot{m}_{\text{f,de}}$	Diesel-equivalent fuel mass flow rate (g/s)
$\dot{m}_{\text{f,diesel}}$	Diesel fuel mass flow rate (g/s)
$\dot{m}_{\text{f,H}_2}$	Hydrogen fuel mass flow rate (g/s)
$\dot{m}_{\text{f,NG}}$	Natural gas fuel mass flow rate (g/s)
$\text{BSFC}_{\text{e,de}}$	Diesel-equivalent brake-specific fuel consumption (g/kWh)
BTE_{e}	Engine brake thermal efficiency (%)
$\text{FC}_{\text{e,de}}$	Diesel-equivalent fuel consumption (L/tonne _{cargo} .100 km)
$\text{LHV}_{\text{diesel}}$	Lower heating value of diesel fuel (MJ/kg)
LHV_{H_2}	Lower heating value of hydrogen (MJ/kg)
LHV_{NG}	Lower heating value of natural gas (MJ/kg)
SOC	Battery state of charge (%)
SOC_{H}	Upper state of charge threshold (%)
SOC_{in}	Initial battery state of charge (%)
SOC_{L}	Lower state of charge threshold (%)
SOC_{min}	Minimum allowable state of charge (%)
SOC_{ref}	Reference state of charge (%)
ρ_{diesel}	Density of diesel fuel (kg/L)
a	Intermediate haversine parameter (-)
A_{f}	Vehicle frontal area (m ²)

c	Angular distance between two points (rad)
C_d	Aerodynamic drag coefficient (-)
D	Total drive-cycle distance before recharging (m)
d	Horizontal distance between two points (m)
E_{aux}	Auxiliary energy demand (J)
$E_{\text{batt,min}}$	Minimum required battery energy capacity (kWh)
$E_{\text{batt,net}}$	Net battery energy consumption (J)
E_b	Battery energy consumption (kJ)
$E_{\text{e-b}}$	Engine energy used for battery charging (kJ)
E_{elec}	Electrical accessories energy consumption (kJ)
$e_{\text{p,input}}$	Specific compressor work input (MJ/kg)
E_{tr}	Traction energy at wheels (kJ)
$E_{\text{w,brk}}$	Braking energy at the wheels (kJ)
f_{r1}	Rolling resistance coefficient (constant term) (-)
f_{r2}	Rolling resistance coefficient (speed-dependent term) (-)
G	Cumulative elevation gain (m)
H	Hamiltonian cost-function value (-)
I_{batt}	Battery current (A)
k	Ring-to-sun gear teeth ratio (-)
k_{yr}	Kinematic coefficient relating yoke to ring gear (-)
k_{ys}	Kinematic coefficient relating yoke to sun gear (-)
M_{cargo}	Cargo mass (tonne)
M_{diesel}	Mass of diesel fuel consumed (kg)
M_{LCF}	Mass of low-carbon fuel consumed (kg)
$m_{\text{tank},i}$	Hydrogen mass in the tank at time step i (kg)
M_v	Vehicle mass (kg)
N	Number of time steps (-)
$N_{\text{batt,p,min}}$	Minimum number of parallel battery modules (-)
$N_{\text{batt,p}}$	Number of parallel battery strings (-)

$N_{\text{batt,s}}$	Number of battery cells in series (-)
n_{fd}	Final drive gear ratio (-)
n_{g}	Gear ratio (-)
n_{tc}	Torque coupler fixed gear ratio (-)
P_{aux}	Auxiliary power demand (kW)
P_{batt}	Battery power (kW)
P_{brake}	Engine brake power (kW)
P_{chg}	Battery charging power (kW)
$P_{\text{e,min}}$	Minimum required engine power (kW)
P_{elec}	Electrical accessory power (kW)
P_{e}	Engine rated power (kW)
$P_{\text{m,max}}$	Maximum electric motor power (kW)
$P_{\text{m,min}}$	Minimum required electric motor power (kW)
P_{mech}	Mechanical accessory power (kW)
$P_{\text{MG1,chg}}$	Charging power from motor-generator 1 (kW)
$P_{\text{MG2,gen}}^{\text{max}}$	Maximum generating power of motor-generator 2 (kW)
$P_{\text{MG2,mot}}^{\text{max}}$	Maximum motoring power of motor-generator 2 (kW)
P_{MG2}	Power of motor-generator 2 (kW)
P_{m}	Electric motor power (kW)
P_{pump}	Hydraulic pump parasitic power input (kW)
$P_{\text{tank,i}}$	Hydrogen tank pressure at time step i (Pa)
$Q_{\text{batt,cell}}$	Single-cell energy capacity (C)
R	Earth radius (m)
$R_{\text{batt,cell}}$	Equivalent internal resistance of a single cell (Ω)
R_{g}	Gas constant (J/(kg · K))
r_{w}	Wheel radius (m)
t_{a}	Acceleration time (s)
T_{c}	Clutch output torque (N.m)
T_{e}	Engine torque (N.m)

T_{fd}	Final drive output torque (N.m)
$T_{\text{g,out}}$	Transmission output torque (N.m)
T_{gen}	Generator output torque (N.m)
T_{high}	Upper torque boundary (N.m)
T_{low}	Lower torque boundary (N.m)
T_{MG1}	Torque of motor-generator 1 (N.m)
T_{MG2}	Torque of motor-generator 2 (N.m)
T_{m}	Electric motor torque (N.m)
T_{opt}	Optimal engine torque (N.m)
T_{req}	Requested torque at wheels (N.m)
T_{r}	Ring gear torque (N.m)
T_{s}	Sun gear torque (N.m)
T_{y}	Yoke torque (N.m)
u_{c}	Clutch state variable (-)
V_{b}	Base vehicle speed (m/s)
V_{f}	Final vehicle speed (m/s)
V_{H}	Upper vehicle speed threshold (m/s)
v_{i}	Vehicle speed at time step i (m/s)
V_{L}	Lower vehicle speed threshold (m/s)
V_{max}	Maximum vehicle speed (m/s)
$V_{\text{oc,cell}}$	Open-circuit voltage of a single cell (V)
V_{tank}	Hydrogen tank internal volume (m ³)
$X(i)$	Cumulative distance traveled at time step i (m)
Z	Compressibility factor (-)
Z_{r}	Number of teeth of ring gear (-)
Z_{s}	Number of teeth of sun gear (-)

Greek

δ_{e}	Engine efficiency threshold for rule-based control strategies (-)
η_{alt}	Alternator efficiency (%)

$\eta_{\text{batt-m}}$	Efficiency between battery and motor (%)
η_{b}	Battery efficiency (%)
η_{c}	Clutch mechanical efficiency (%)
η_{fd}	Final drive mechanical efficiency (%)
η_{gen}	Generator efficiency (%)
η_{g}	Gearbox mechanical efficiency (%)
$\eta_{\text{m,brk}}$	Electric motor efficiency in braking mode (%)
$\eta_{\text{m,pr}}$	Electric motor efficiency in propulsion mode (%)
η_{m}	Electric motor efficiency (%)
η_{pc}	Power converter efficiency (%)
$\eta_{\text{t,e}}$	Overall driveline efficiency from engine to wheels (%)
$\eta_{\text{t,m}}$	Overall driveline efficiency from motor to wheels (%)
η_{tc}	Torque coupler efficiency (%)
η_{w}	Wheel efficiency (%)
η_{yr}	Efficiency associated with yoke-to-ring power transfer (%)
ω_{c}	Clutch output angular speed (rad/s)
$\omega_{\text{e,max}}$	Maximum efficient engine angular speed (rad/s)
$\omega_{\text{e,min}}$	Minimum efficient engine angular speed (rad/s)
ω_{e}	Engine angular speed (rad/s)
ω_{fd}	Final drive output angular speed (rad/s)
$\omega_{\text{g,in}}$	Transmission input shaft angular speed (rad/s)
$\omega_{\text{g,out}}$	Transmission output angular speed (rad/s)
ω_{gen}	Generator angular speed (rad/s)
ω_{MG2}	Angular speed of motor-generator 2 (rad/s)
ω_{m}	Electric motor angular speed (rad/s)
ω_{r}	Ring gear angular speed (rad/s)
ω_{s}	Sun gear angular speed (rad/s)
ω_{y}	Yoke angular speed (rad/s)
ρ_{a}	Air density (kg/m ³)

θ_i Road grade angle at time step i (rad)

$\Theta_{\text{tank},i}$ Hydrogen tank temperature at time step i (K)

Abbreviations

AER All-Electric Range.

ATIS Automatic Tire Inflation System.

BMEP Brake Mean Effective Pressure.

BSFC Brake Specific Fuel Consumption.

BTE Brake Thermal Efficiency.

CARB California Air Resources Board.

CD Charge Depletion.

CH₄ Methane.

CI Compression Ignition.

CNG Compressed Natural Gas.

CO₂ Carbon Dioxide.

CR Compression Ratio.

CS Charge Sustaining.

CSHVR City-Suburban Heavy Vehicle Route.

DC Direct Current.

DDR Daily Driving Range.

DEM Digital Elevation Model.

DI Direct Injection.

DOC Diesel Oxidation Catalyst.

DOE Design Of Experiments.

DP Dynamic Programming.

DPF Diesel Particulate Filter.

EA Electrical Accessories.

EATS Engine Aftertreatment System.

EC European Commission.

ECMS Equivalent Consumption Minimization Strategy.

ECU Engine Control Unit.

eCVT Electronic Continuously Variable Transmission.

EGR Exhaust Gas Recirculation.

EMS Energy Management System.

EPA Environmental Protection Agency.

EU European Union.

FHWA Federal Highway Administration.

FTE Freight Transport Efficiency.

GA Genetic Algorithm.

GHG Greenhouse Gas.

GPS Global Positioning System.

GVWR Gross Vehicle Weight Rating.

GWP Global Warming Potential.

H₂ Hydrogen.

HD Heavy-Duty.

HHDDT Heavy Heavy-Duty Diesel Truck.

HPDI High-Pressure Direct Injection.

HVAC Heating, Ventilation, And Air Conditioning.

ICE Internal Combustion Engine.

IM Induction Motor.

IMEP Indicated Mean Effective Pressure.

ISG Integrated Starter Generator.

ITE Institute Of Transportation Engineers.

LCFS Low Carbon Fuel Standard.

LFP Lithium Iron Phosphate.

LHV Lower Heating Value.

LNG Liquefied Natural Gas.

LTO Lithium Titanate Oxide.

MAE Mean Absolute Error.

MAPE Mean Absolute Percentage Error.

MG Motor/Generator.

MY Model Year.

NG Natural Gas.

NMC Nickel Manganese Cobalt.

NO_x Nitrogen Oxides.

NREL National Renewable Energy Laboratory.

OEM Original Equipment Manufacturer.

ORC Organic Rankine Cycle.

P₂ Pre-Transmission Parallel.

PC Power Converter.

PCP Peak Cylinder Pressure.

PER Pilot Energy Ratio.

PFI Port Fuel Injection.

PM Particulate Matter.

PMSM Permanent Magnet Synchronous Motor.

PSH Power-Split Hybrid.

PTO Power Take-Off.

RBC Rule-Based Control.

RK Redlich-Kwong.

RMSE Root Mean Square Error.

SAE Society Of Automotive Engineers.

SCR Selective Catalyst Reduction.

SI Spark Ignition.

SMR Steam Methane Reforming.

SOC State Of Charge.

SOI Start Of Injection.

SRM Switched Reluctance Motor.

TC Turbo-Compounding.

TDC Top Dead Center.

THS Toyota Hybrid System.

TWC Three-Way Catalyst.

U.S. United States.

VGT/C Variable Geometry Turbocharger.

VVA Variable Valve Actuation.

WBS Wide Base Single.

WCC Wheel Cavity Cover.

WHR Waste Heat Recovery.

WHTC World Harmonized Transient Cycle.

XPI Extreme Pressure Injection.

ZEV Zero-Emission Vehicle.

Chapter 1

Introduction and Problem Statement¹

The rise in global temperatures over the past century has been closely linked to increasing anthropogenic greenhouse gas (GHG) emissions, prompting international efforts such as the Paris Agreement (2015), which aims to limit global warming to below 2 °C [7]. Despite these efforts, global emissions continue to increase, underscoring the urgency of addressing climate change across major energy-consuming sectors. Among these, the transportation sector exhibits the fastest growth rates in GHG emissions, with an average annual increase of approximately 1.5–2% over recent decades [8]. The transportation sector currently contributes to 23% of global energy-related carbon dioxide (CO₂) emissions [9]. Road vehicles account for nearly 75% of this share, primarily due to the widespread use of internal combustion engines (ICEs) [10]. In addition, transportation relies heavily on petroleum-based fuels, consuming more than half of global oil production, raising concerns related to energy security and supply stability [11]. These trends highlight the significant role of the transportation sector in the global GHG challenge and emphasize the need for effective and scalable mitigation strategies to reduce its environmental impact.

Building on the significant contribution of the transportation sector to global GHG emissions, freight transportation, particularly road freight, plays a critical and growing role. In North America, the road freight market is valued at approximately USD 660 billion in 2025 and is projected to reach nearly USD 800 billion by 2030, reflecting increasing demand for goods movement [12]. Heavy-duty (HD) trucks, defined in the United States (U.S.) and Canada as vehicles with a gross vehicle weight rating (GVWR) exceeding 11.8 tonne (26,000 lb), transport more than 70% of domestic freight by volume [13]. Despite this, they contribute disproportionately to

¹This chapter is partially based on the published review paper in [1].

emissions, accounting for nearly 25% of on-road GHG emissions in these regions [14, 15], highlighting the need for improved efficiency and emission reduction strategies.

Within the HD truck segment, large tractor-trailer trucks with a GVWR exceeding 15 tonne (33,000 lb) are classified as Class 8 in North America and as category N3 in Europe [16, 17]. These vehicles can be further categorized into three groups based on their daily driving range (DDR): 1) long-haul (sleeper-cab) trucks, with a DDR of 600-800 km; 2) regional-haul (day-cab) trucks, with a DDR of 400-600 km; and 3) drayage trucks, with a DDR of less than 400 km [18, 19]. Although Class 8 (or category N3) trucks constitute a small portion of the HD fleet (less than 16%), they are responsible for a significant share of emissions, contributing up to 50% of the fleet’s GHG and nitrogen oxides (NO_x) emissions [15, 20]. In response to growing environmental concerns, the transportation sector is undergoing a transition toward more energy-efficient and low-carbon technologies, supported by increased investment in advanced powertrain systems and alternative fuels [21]. While powertrain electrification and the use of alternative fuels such as hydrogen are gaining attention, ICEs are expected to remain dominant in both light and HD fleets in the near future, accounting still for ~69% of the fleet by 2050 [15]. This thesis presents a near-term pathway for decarbonizing Class 8 HD trucks through the integration of powertrain electrification and low- or near-zero-carbon, high-efficiency ICEs.

1.1 Heavy-Duty Transportation and Its Impact on Greenhouse Gas Emissions

Although Class 8 tractor-trailer trucks represent a small share of commercial HD fleets across different countries, their high DDR (typically exceeding 300 km) leads to a significant contribution to tailpipe emissions, as shown in Table 1.1. The share of Class 8 trucks within total commercial HD vehicles is relatively low, at 9% in the U.S. [22], 6.4% in Canada [23], 16% in the European Union (EU) [24], and 7.8% in China [25]. However, their contribution to GHG emissions is substantial, accounting for 50%, 40%, 25%, and 23% of total emissions in these regions, respectively [25, 26].

The GHG emissions from commercial HD trucking in developed economies have increased in recent decades due to the increasing demand for road freight transport. In North America, GHG emissions from HD trucks have doubled over the past 25 years, despite significant improvements in vehicle fuel economy [27]. A similar trend is observed in the EU, where emissions from HD trucks have increased by approximately 20–30% since 1990, primarily due to rising freight activity and transport demand [24,

Table 1.1: Share of Class 8 tractor-trailer trucks in tailpipe emissions from commercial HD vehicles in the United States (U.S.), Canada, the European Union (EU), and China [22–26].

Jurisdiction	United States	Canada	European Union	China
Class 8 truck share in commercial HD vehicles	9%	6.4%	16%	7.8%
Class 8 truck emissions share in commercial HD vehicles	50% of NO _x and 50% of GHGs	40% of GHGs	25% of GHGs	74% of NO _x and 23% of GHGs and 52.4% of PM ¹

¹ PM: Particulate matter, referring to airborne particles with diameters typically less than 10 μm (PM₁₀) or 2.5 μm (PM_{2.5}), associated with combustion processes.

25]. Eliminating these emissions is essential to achieve global net-zero targets, while ensuring that freight transportation continues to support economic activity in a cost-effective manner.

The pressing need to reduce GHG emissions has led to stricter regulations for Class 8 HD trucks. In response to their high contribution to emissions, the European Commission (EC), the U.S. Environmental Protection Agency (EPA), and the California Air Resources Board (CARB) have implemented increasingly stringent regulations for GHG emissions for Class 8 trucks. In the U.S., Phase-2 GHG regulations require a 15% reduction in tailpipe CO₂ emissions from 2021 to 2027, followed by Phase-3, mandating a 25% reduction from 2027 to 2032 [28–30]. In 2023, the EU established targets to cut CO₂ emissions by 45% by 2030, 65% by 2035, and 90% by 2040, relative to a 2019 baseline [24, 31]. In China, under China-VI (Stage III) regulations, a CO₂ reduction of at least 30% has been set by the end of 2030 compared to a 2012 baseline [32].

In addition to CO₂ reduction targets, interim zero-emission vehicle (ZEV) adoption is promoted for HD truck decarbonization. In the U.S., state-level policies led by CARB require 25% ZEV sales by 2030 and 40% by 2035 for HD trucks [33, 34]. In contrast, the EU does not impose a direct ZEV mandate for HD trucks; however, its CO₂ reduction target of about 90% by 2040 effectively necessitates a substantial share of ZEVs in the fleet [35]. ZEVs in the HD sector primarily include battery electric and fuel-cell electric trucks, which produce no tailpipe emissions [36, 37]. However, HD trucks equipped with near-zero-carbon-fueled ICEs, such as those powered by hydrogen, may also qualify as ZEVs under specific regulatory thresholds defined by the EU [38], as summarized in Table 1.2. These thresholds vary depending on the engine type, spark ignition (SI) or compression ignition (CI), as well as the method

of hydrogen production.

Table 1.2: Equivalent CO₂ emission thresholds regulated by the European Union (EU) for classifying hydrogen-fueled HD trucks as zero-emission vehicles based on engine type [38, 39].

Zero-Emission Thresholds for HD Trucks (g CO₂/tonne_{cargo}·km)	Spark Ignition (SI) Engines	Compression Ignition (CI) Engines
Tank-to-Wheel (or Tailpipe)	< 1	< 3
Well-to-Wheel (Green Hydrogen)	< 9	< 8
Well-to-Wheel (Blue Hydrogen)	< 18	< 16

The majority of the global Class 8 fleet still operates on conventional diesel engines that contribute to GHG emissions and are major sources of criteria pollutants such as NO_x and particulate matter (PM) [38]. In the U.S., the EPA requires a 82.5% reduction in tailpipe NO_x emissions for model year (MY) 2027 engines relative to the current federal standard of 0.27 g/kWh [40]. In addition, CARB introduced the HD Low NO_x Omnibus regulation in 2020, applicable from 2024, which requires up to a 90% reduction in tailpipe NO_x emissions [41]. In Europe, the Euro-VII standard is expected to further tighten emission limits, requiring a 24% reduction in tailpipe NO_x emissions over the World Harmonized Transient Cycle (WHTC) compared to the Euro-VI standard [31]. These substantial reductions are expected to be achieved through advancements in engine and powertrain technologies, along with significant improvements in exhaust aftertreatment systems [42].

Reducing GHG emissions and complying with emission regulations in Class 8 trucks can be achieved through three primary approaches [43]: 1) lowering transportation activity, measured in tonne-km; 2) improving transportation energy efficiency, defined as the energy consumed per tonne-km, through advancements in engine and vehicle technologies; and 3) utilizing fuels with lower carbon intensity, characterized by the amount of CO₂ emitted per unit of energy. Given the growing demand for road freight transportation, the first approach is not a practical option due to its strong linkage with economic development. Therefore, technology improvements and transitioning to low-carbon fuels represent the most feasible near-term solutions.

1.2 Near-Term Solutions for Decarbonizing Heavy-Duty Trucks

Current HD truck technologies are advancing rapidly in response to regulatory pressures, rising fuel costs, and increasing concerns over environmental sustainability. These advancements include improvements in engine efficiency, developments in powertrain systems, and enhancements in vehicle technologies aimed at reducing road resistance forces. Experimental studies [44, 45] conducted by EU-funded consortia in collaboration with major vehicle original equipment manufacturers (OEMs) show that the combined application of these advancements to Class 8 diesel trucks can achieve 18% to 27% reductions in tailpipe CO₂ emissions. These reductions are relative to a 2019 baseline truck. The range of reduction corresponds to the application of a 48 V mild hybrid electric powertrain rather than a full hybrid configuration, as shown in Figure 1.1.

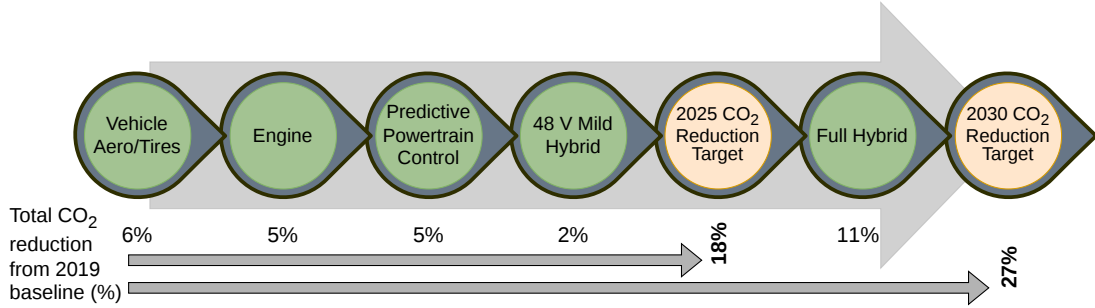


Figure 1.1: Technology advancement packages proposed by EU-funded consortia in collaboration with major OEMs to reduce tailpipe CO₂ emissions of Class 8 diesel HD trucks and meet regulated reduction targets for 2025 and 2030. These advancements cover engine, powertrain, and vehicle technologies aimed at reducing road resistance forces, with an estimated payback period of two years [44, 45].

Although the proposed European technology advancement package meets the regulated CO₂ reduction targets for 2025, it is not sufficient to achieve the 2030 target of a 45% reduction relative to a 2019 baseline [31]. To address this gap, a feasible near-term pathway has been proposed based on transitioning to low- or zero-carbon-fueled ICEs that maintain brake thermal efficiency (BTE) comparable to conventional diesel engines [31, 46]. Figure 1.2 illustrates the impact of integrating low-carbon natural gas (NG)-fueled ICEs with the aforementioned technology advancements to meet the EU 2030 CO₂ reduction target for Class 8 HD trucks.

A further step is required to achieve the more stringent EU CO₂ reduction targets of 65% by 2035 and 90% by 2040, as NG with 20–30% lower carbon intensity than

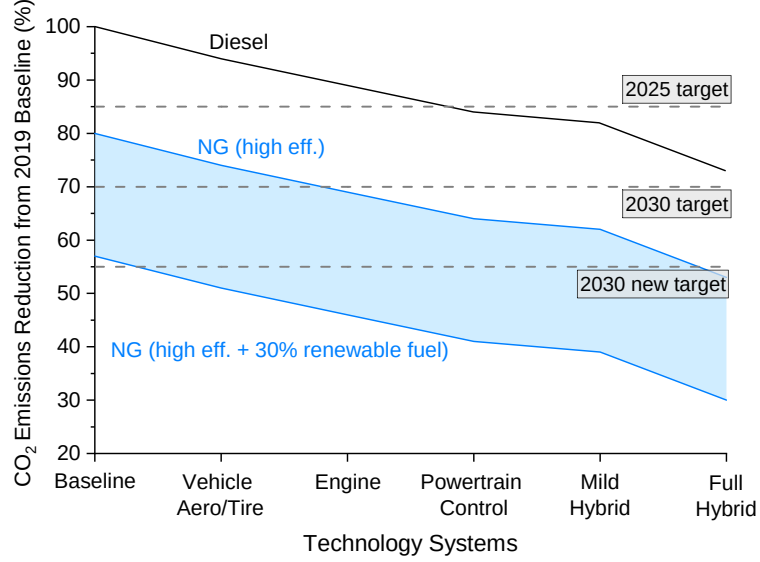


Figure 1.2: Impact of integrating low-carbon NG-fueled ICEs with advanced technologies to meet the 2030 tailpipe CO₂ reduction target for Class 8 HD trucks in the EU [31, 46].

diesel, is no longer sufficient to meet these targets. Given the current technological, infrastructural, and cost-related challenges in fully phasing out ICEs as part of the ZEV transition by 2040, a viable intermediate pathway is the use of high-efficiency ICEs powered by zero-carbon fuels such as hydrogen (H₂) [47, 48]. Concurrent advancements in powertrain and vehicle technologies are still required to reduce fuel consumption, in addition to lowering tailpipe GHG emissions.

On the other hand, the U.S. Department of Energy has collaborated with major vehicle OEMs through three phases of the ‘SuperTruck’ program to improve efficiency and reduce GHG emissions in Class 8 HD trucks. As the U.S. CO₂ reduction targets are less stringent than those in the EU, the first two phases of the SuperTruck program focused on diesel-based powertrains, with advancements in engine, powertrain (including mild and full hybridization), and vehicle-side technologies. These advancements have been reported to achieve a peak engine BTE of up to 50%, while enabling 25% reductions in tailpipe CO₂ emissions relative to a MY2009 baseline, with a payback period of two years [49, 50]. The third phase of the program, which is currently ongoing, shifts focus toward zero-emission pathways, including hydrogen utilization in fuel-cell and battery electric HD truck platforms [51].

However, despite their zero-tailpipe-emission potential, battery electric and hydrogen fuel-cell trucks still face important barriers for broad near-term deployment in Class 8 long-haul and high-utilization applications. Battery electric trucks are promising for urban, drayage, and depot-based routes, but long-haul operation remains con-

strained by limited driving range, high battery mass and cost, charging time, battery degradation and residual-value uncertainty, and the need for high-power charging infrastructure [37, 52]. Hydrogen fuel-cell trucks can offer longer range and faster refueling, but their deployment remains limited by high vehicle and hydrogen fuel costs, insufficient refueling infrastructure, and long-term fuel-cell durability concerns under HD operation [53, 54]. Therefore, hybridized low- or zero-carbon-fueled ICE-based powertrains remain a relevant near-term pathway for reducing fuel consumption and GHG emissions in Class 8 HD trucks.

1.2.1 Engine Technology Advancements

Engine technology advancements range from increasing the compression ratio (CR) to enhancing the recovery of dissipated energy through waste heat recovery (WHR) systems [15]. The different technological features are summarized in Table 1.3 for HD trucks, along with their corresponding reductions in fuel consumption, which are directly proportional to reductions in tailpipe CO₂ emissions. Understanding how the different technologies can be combined to achieve real CO₂ reductions is important. In the European context, the combination of engine technologies required to meet the 2030 GHG reduction target, as previously shown in Figure 1.1, is as follows: friction losses reduction, higher CR, downspeeding, WHR system, enabling peak BTE to reach almost 54% [44, 45].

In Phase two of the SuperTruck program in the U.S., the combination of advanced engine technologies enabled peak BTE exceeding 50% for Class 8 HD truck applications [49]. These technologies include higher CR, friction reduction, optimized two-stage turbocharging, engine downspeeding, variable valve actuation (VVA) to enable Miller cycle operation, and two-stage EGR cooling [50]. Regarding aftertreatment system development, all teams have adopted closed-coupled selective catalytic reduction (SCR) systems in combination with 48 V electrically heated catalysts. A series of significant improvements, based on improving the overall in-cylinder combustion process while reducing losses in the engine system, can produce significant net additive benefits [51].

1.2.2 Vehicle Side Technology Advancements

In addition to advancing engine-side technologies aimed at reducing fuel consumption and emissions in Class 8 HD trucks, various vehicle-side modifications can be implemented. These vehicle-side technologies encompass a range of enhancements,

Table 1.3: Advanced engine modifications and technology systems in commercial HD trucks [15, 46, 47, 55–64].

Technology	Description	Benefits
Common rail injection	• Higher injection pressure, maximum around 2700 to 3000 bar	• Smaller and faster fuel droplets improved air entrainment and mixing • Reduces soot emission • Offset the drawbacks of a high EGR¹ rate • 1.1% FC² reduction
Higher compression ratio	• Optimization of combustion chamber design affects the combustion process	• 2% to 15% higher BTE in HD engines
Advanced turbocharging and downsizing	• Variable geometry turbine (VGT) • Multi-stage turbocharging • Twin scroll housing • Double scroll housing	• 2.9% FC reduction • Improve transient and part load conditions
Turbo-compounding	• Using a turbine to recover the exhaust energy • Connected directly to the crankshaft or powers a generator	• 1.8% to 4% FC reduction
Waste Heat Recovery (WHR)	• Conversion of dissipated engine heat to mechanical or electrical energy • Closed Rankine cycle • Thermo-electric generators	• BTE improvement • 3% to 4% FC reduction
Engine friction reduction	• Piston skirt and piston ring friction reduction • Advanced materials/lubricants	• 1.5% FC reduction
Engine downsizing	• Not receiving much attention in HD engines as it is disruptive to implement and no significant efficiency improvement has been reported.	• The benefits are lower weight and friction, and better packaging. • However, reducing low-end torque, lower life expectancy, and increased heat losses.
Downspeeding	• More popular than downsizing in HD engines. • Reducing friction and shifting engine operation to the optimum operating range. • Changing the final drive ratio, gearbox gear ratio, or shifting strategy.	• 5% to 8% FC reduction • Reduces friction losses • Operates engine within the optimum speed range.
Variable valve actuation (VVA)	• Variable cam phasing, valve lift, and cylinder deactivation • Valvetrain adjusts as a function of engine load and speed	• Limited benefits in diesel engines • Enables late inlet valve closure (Miller cycle) • Provides thermal management system of engine aftertreatment system (EATS) by changing exhaust gas temperature • FC reduction in part load of gasoline engines

¹ EGR: Exhaust gas recirculation system.

² FC: Fuel consumption. The reported reductions in fuel consumption are equivalent to the corresponding reductions in tailpipe CO₂ emissions.

including aerodynamic modifications, tire and trailer weight modifications, and drive assistant technologies. Vehicle-side improvements are essential in enhancing tractor-trailer efficiency, which has been reflected in the U.S. Phase-2 GHG regulation [20, 65]. Although the CO₂ emission of the trailer by itself is zero, its design greatly impacts the fuel consumption. The most substantial resisting force for Class 8 HD trucks during highway driving at zero grade is aerodynamic resistance. The aerodynamic resisting force is proportional to the second power of velocity, and it accounts for up to 25% of fuel consumption [66–68]. However, the impact of aerodynamic drag forces can be minimized by enhancing aero features on both the tractor and trailer sides [69, 70]. The critical areas in a tractor-trailer on which the aerodynamic drag affects are: 1) the front surface and roof of the tractor, 2) the tractor-trailer gap, 3) the rear end of the trailer, and 4) the side and under-body of the trailer.

The combined application of vehicle-side technologies, including aerodynamic improvements, low-resistance tires, and overall weight reduction, in Class 8 tractor-trailer trucks can achieve fuel consumption reductions of 11.9% [68, 71, 72] and 8.4% [65, 69, 70] over long- and regional-haul drive cycles, respectively. The schematic of the technologies to reduce the aerodynamic and tire rolling resistances in both tractor and trailer sides is shown in Figure 1.3. The contribution of using aerodynamics, tires, and weight reduction modifications in the research literature to mitigate both the drag coefficient and fuel consumption of Class 8 sleeper trucks over the highway has been shown in Table 1.4.

Considering a two year payback period, the U.S. and the EU have proposed a combination of aerodynamic advancements to achieve an almost 6% reduction in fuel consumption and tailpipe CO₂ emissions for Class 8 long-haul trucks. These advancements on the tractor side include roof deflectors, cabin fairings, and tractor-trailer gap reducers [20, 50]. On the trailer side, they include side skirts, underbody fairings, and rear vanes. Tire-related advances include lightweight tires, wheel cavity covers, and using wide-base single tires [1, 72, 76].

1.2.3 Role of Low- or Zero-Carbon-Fueled Internal Combustion Engines

Alternative fuels offer significant potential for reducing GHG emissions in HD transportation, with low-carbon options such as NG providing near-term benefits and zero-carbon fuels such as H₂ offering mid-term solutions [43, 48]. Main gaseous fuels with low carbon intensities include NG, syngas, biogas, biomethane (also referred to as renewable natural gas, RNG), and H₂. The carbon intensity of these fuels varies

Table 1.4: Contribution of various aerodynamic and tire rolling resistance technologies to reduce both the drag coefficient and fuel consumption of Class 8 tractor-trailer trucks in the literature.

Vehicle Side Technologies	Usage Domain in the Literature						
	[65, 70, 73]	[68, 70]	[69, 72]	[74]	[75]	[56]	[76]
Tractor Side Aerodynamic Improvements							
Roof deflector	✓	–	✓	–	–	–	–
Roof and cabin fairings	✓	–	✓	–	–	–	–
Active grille shutter	–	–	–	–	–	✓	–
Camera mirrors	–	–	–	–	–	–	✓
Tractor–Trailer Gap Aerodynamic Improvements							
Trailer fairings	✓	–	✓	–	–	–	–
Gap reducer	✓	–	✓	–	–	–	–
Cab-side extender	✓	–	✓	–	–	–	–
Trailer Side Aerodynamic Improvements							
Side skirt	✓	–	✓	–	✓	–	✓
Underbody fairings	✓	–	✓	–	✓	–	–
Boat tails and rear vanes	✓	–	✓	–	–	–	✓
Active flow control	–	–	✓	–	–	–	–
Tires and Wheels Improvements to Reduce Rolling Resistance and Weight							
Light-weight tires	–	✓	✓	✓	–	–	–
Automatic tire inflation system (ATIS)	–	✓	–	–	–	–	–
Wheel cavity cover (WCC)	–	✓	✓	–	✓	–	–
Wide base single tire (WBS)	–	✓	✓	✓	–	–	–
Relative Percentage Reduction in Drag Coefficient (%)							
	NA	NA	LH ¹ : 21.0	NA	NA	NA	LH: 12.0
Relative Percentage Reduction in Fuel Consumption (%)							
	LH: 6.3 RH ² : 3.6	LH: 4.4 RH: 2.6	LH: 11.9 RH: 8.4	LH: 4.0	LH: 5.0	LH: 1.6	NA

¹ LH: Long-haul trucks with daily driving range of 600-800 km.

² RH: Regional-haul trucks with daily driving range of 400-600 km.

✓: The technology is included in the study.

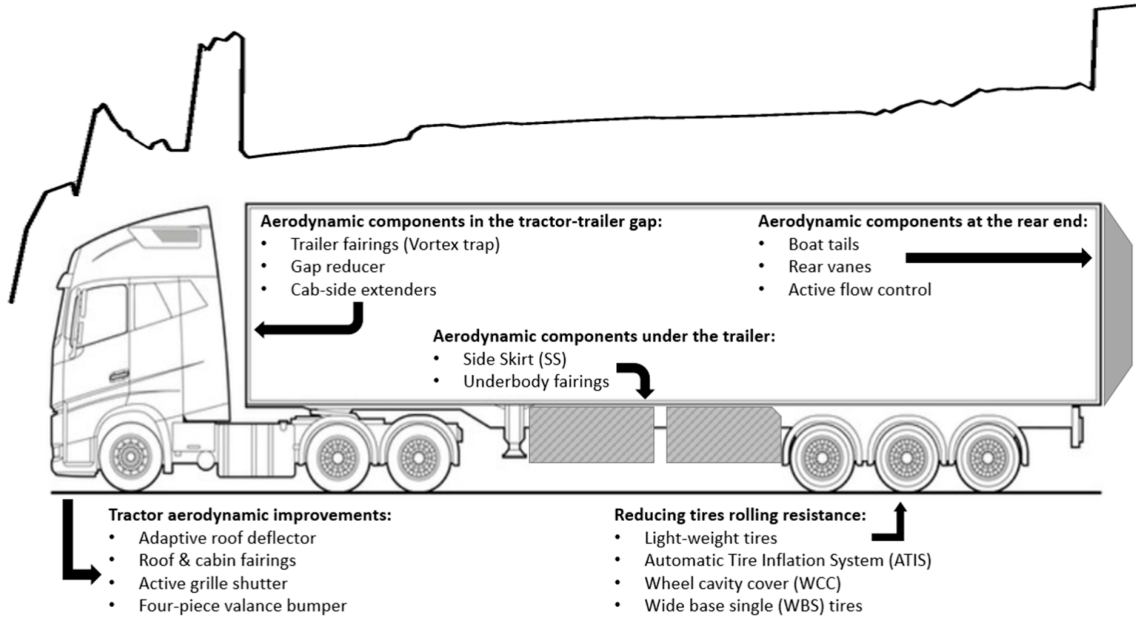


Figure 1.3: Schematic of a European cab-over tractor-trailer configuration for a Class 8 HD truck with indicative areas of aerodynamic drag and rolling resistance reductions. The trend line represents qualitatively the net contributions of each segment to the overall vehicle drag. Reprinted from my previously published work in [1], licensed under CC BY 4.0.

depending on the feedstock and production pathway. Based on the CARB low carbon fuel standard (LCFS) [77], the median certified carbon intensity of NG corresponds to a 5% to 20% reduction in well-to-wheel GHG emissions relative to conventional petroleum-derived diesel fuel. Bio-derived compressed natural gas (CNG) exhibits a wide range of carbon intensities, from -600 to +100 g CO₂e/MJ, which translates to reductions ranging from 0% to 700% compared to diesel [43]. Methane (CH₄), the primary component of NG, has the lowest carbon-to-hydrogen ratio among hydrocarbons, resulting in 20% to 25% lower tailpipe CO₂ emissions on an energy-equivalent basis compared to diesel combustion at similar efficiency [78]. However, due to the high global warming potential (GWP) of methane, 29.8 times that of CO₂ over a 100-year time horizon [79], incomplete combustion (‘methane slip’) can significantly diminish or even offset these GHG benefits, posing a major challenge for the decarbonization of NG-fueled HD engines.

Natural gas is commonly utilized in port fuel injection (PFI) SI engines, mainly due to its strong resistance to auto-ignition. This property enables the use of higher CR which can improve engine efficiency [80]. However, the relatively low volumetric energy density of NG necessitates either liquefaction for storage as liquefied natural

gas (LNG) or compression as compressed natural gas (CNG), which introduce additional cost and technical challenges [81]. The low density of NG can also reduce volumetric efficiency, although this can be mitigated by increasing intake manifold pressure [82]. Additional challenges include low flame speed, fuel-air premixing in crevice volumes, and resistance to ignition, which substantially increase CH_4 emissions and diminish net GHG benefits [83]. A more effective combustion technique to improve efficiency and reduce CH_4 emissions involves diesel-pilot ignition combined with late-cycle direct injection of NG, which primarily operates in a non-premixed combustion mode [43].

Hydrogen has attracted increasing attention as an alternative fuel for HD trucks, particularly in long-haul applications, despite its low volumetric energy density. It offers favorable combustion characteristics, including zero carbon content, wide flammability limits, high diffusivity and mixing capability, and fast flame propagation speeds [84, 85]. However, the application of H_2 in HD transportation has some challenges. The energy required to compress hydrogen to 350–700 bar or to liquefy it at approximately -250°C significantly increases the net energy cost and introduces technical complexity, particularly for on-board storage and compression systems in Class 8 trucks [81, 86]. From a GHG perspective, the environmental benefit of H_2 is dependent on its production pathway and source [53, 87]. Certain production methods may result in higher carbon intensities than diesel; for example, hydrogen produced by steam methane reforming (SMR) of NG exhibits carbon intensities up to 200% worse than diesel [77]. In contrast, hydrogen derived from SMR using manure RNG can achieve carbon intensities in the range of -300 to $-200\text{ g CO}_2\text{e/MJ}$, corresponding to reductions of 300% to 400% relative to diesel [53].

When integrated with ICE technology, H_2 offers a promising pathway to reduce GHG emissions in HD applications. H_2 -ICEs benefit from the current advanced combustion and ICE control technology and use the existing supply chain [88, 89]. Furthermore, H_2 -ICEs are relatively tolerant to fuel impurities and involve lower development costs compared to fuel-cell and battery electric powertrains [90]. Most early H_2 -ICE implementations have employed PFI systems, which present several challenges, including charge displacement, low power density, knocking and pre-ignition, as well as backfiring associated with abnormal combustion [89, 91]. The adoption of hydrogen direct injection (H_2 -DI) can resolve these issues, while improving engine efficiency and emissions characteristics [84, 85, 92].

The classification of NG- and H_2 -fueled ICEs for HD truck applications, based on ignition system and injection strategy, is presented in Figure 1.4. Beyond single-fuel

NG- or H₂-fueled SI engines, these fuels can also be utilized in diffusion-controlled combustion modes within dual-fuel CI engines, where a small quantity of diesel is used as a pilot to initiate ignition [43, 92, 93]. In such configurations, NG or H₂ can be supplied either through intake manifold injection at pressures of 3–10 bar or via direct in-cylinder injection at higher pressures in the range of 100–600 bar [94, 95]. High-pressure direct injection (HPDI) of NG or H₂ enables the highest overall engine efficiency among the configurations shown in Figure 1.4, while minimizing the risk of backfiring and reducing the likelihood of pre-ignition [96]. Direct injection of NG near top dead center (TDC) enables diesel-like efficiency and provides transient response characteristics comparable to conventional diesel engines [97]. An HPDI NG engine, which uses a small amount of diesel for ignition (typically less than 5% of total fuel energy), has similar efficiency to diesel engines while emitting 15%–25% lower GHGs [43].

A further advantage of HPDI is that it is already commercially established for natural gas applications, so its adaptation to hydrogen can leverage an existing fuel injection system with relatively limited changes [78, 93]. The diesel-pilot HPDI H₂ combustion process follows diesel-cycle thermodynamics and has been shown to increase BTE by approximately 2–8% compared with conventional diesel engines [94, 95]. Demonstrations on HD platforms, such as the Scania 13 L CBE1 engine, have reported peak BTE values exceeding 51% [98, 99], surpassing conventional diesel benchmarks under comparable operating conditions. These improvements are attributed to faster hydrogen combustion, reduced heat transfer losses associated with lower injection momentum, and molar expansion effects that increase work during the expansion stroke [96]. The pilot energy ratio (PER) is a critical parameter for determining whether HPDI H₂ engines can meet emerging ZEV regulations. The study in [39] showed that long-haul truck categories require an average PER of approximately 3–5% to satisfy the EU’s ZEV threshold, while higher PER values (> 6%) would exceed the regulatory limit.

1.2.4 Role of Powertrain Hybridization

Alongside advancing engine and vehicle-side technologies, powertrain electrification emerges as a potential solution to address the increasingly demanding fuel consumption and GHG emission standards for Class 8 HD truck applications [100, 101]. While the benefits of electrification have been well established for light-duty vehicles, further investigation is required to fully assess its effectiveness in HD applications [57, 102]. Techniques to reduce fuel consumption and GHG emissions for Class 8 HD trucks

Combustion System Options		Specifications	Advantages	Disadvantages
Otto premixed (Single fuel)	Port Fuel Injection (PFI)	- Spark ignition (SI). 3-10 bar injection pressure. - Injecting H₂ or NG during the intake stroke (Otto cycle).	- Simplest system with minimal engine modification. - Low-cost fuel system. - Low NO_x emissions.	- Loss of power density and efficiency. - High risk of backfire and pre-ignition. - Poor transient response. - Extreme turbocharging demand.
	Direct Injection (DI)	- SI. 10-50 bar injection pressure. - Injecting H₂ or NG during the early compression stroke. - Pent-roof cylinder head design.	- Higher power density, efficiency, and transient response than PFI. - Moderate engine modification. - No backfire risk, reduced pre-ignition.	- Higher NO_x emissions than PFI. - A high-durability injection system is required. - Development effort for optimization. - Aftertreatment system, i.e., selective catalytic reduction (SCR).
	Pre-chamber Technology	- SI (more ignition kernels). - SI configuration (Passive MJJ). 4-15 bar inj. pressure of the H₂ or NG, PFI. 50 bar DI, pre-chamber injector (Active MJJ).	- Faster combustion. - Knock limit is extended. - Reduces abnormal combustion. - Mitigate H₂ or CH₄ slip. - Better fuel economy and lower NO_x compared to DI.	- Hardware complexity, packaging, cost. - Durability/orifice sensitivity. - High boosting requirement at ultra-lean operation. - Calibration sensitivity and control burden.
Diesel diffusive (Dual fuel)	Dual Fuel	- Compression ignition. 3-10 bar H₂ or NG, and 1600-2000 bar diesel injection pressures. - Injecting H₂ or NG during intake stroke and diesel towards the end of compression stroke (Diesel cycle).	- Higher efficiency than Otto premixed, due to the diesel cycle. - Simple system with minimal engine modification. - Low-cost fuel system. - Low NO_x emissions.	- High risk of backfire and pre-ignition. - H₂ or NG substitution is limited at high load. - Poor transient response. - Development effort for optimization. - Extreme turbocharging demand.
	Diesel Pilot, High-pressure Direct Injection (HPDI)	- Compression ignition. 100-600 bar injection pressure. - Injecting H₂ or NG towards the end of compression stroke just after pilot ignition (Diesel cycle).	- Highest efficiency, power density, and transient response. - Reduced turbocharging demand. - Moderate engine modification. - No back-fire risk, reduced pre-ignition.	- High NO_x emissions. SCR is needed. - H₂ or CNG must be stored at high pressure (700 bar) in the on-board tanks. - Needs a high-durability injection system. - Development effort for optimization.

Figure 1.4: Different types of NG- and H₂-fueled ICEs for HD truck applications, classified by ignition system and injection strategy. The figure summarizes the technical specifications, as well as the advantages and disadvantages of each category, based on available literature [43, 80, 83, 95, 97].

cover a broad range, from idle elimination to full electric drive.

Hybrid electric powertrains offer a practical approach to enhance the net powertrain energy efficiency of HD vehicles, while requiring relatively smaller electrical driveline components compared to fully electric powertrains [47]. The key advantages of hybrid technology over conventional powertrains are twofold: 1) the recovery of a portion of braking energy at the driving wheels through regenerative braking, and 2) the improvement of average engine BTE by operating the engine more frequently within its high-efficiency region, thereby increasing the net powertrain energy efficiency [1, 103]. Higher engine BTE is achieved by utilizing the electrical driveline, including the high-voltage battery and e-motor(s), during transient or low power demand driving conditions where the ICE exhibits lower efficiency [104, 105]. The inclusion of electrical driveline components, alongside the engine as the fossil-based source of power, allows for various hybrid architectures with different placements of e-motor(s), transmission, and clutch(s).

Hybrid electric architectures can be categorized into three main groups based on the level of electrification (mainly battery size, system voltage, and e-motor(s) power) [106, 107]: 1) micro hybrids, 2) mild hybrids, and 3) full (or strong) hybrids. Full hybrid electric architectures can be further classified according to the powertrain configuration (type of coupling between ICE, e-motor(s), and driving wheels) [108], including series hybrids, parallel hybrids, series-parallel (power-split) hybrids, and plug-in hybrids. The selection of a hybrid powertrain configuration is influenced by several factors, including vehicle application, driving conditions, cost considerations, and performance requirements. Each configuration presents distinct advantages and limitations, making it more or less suitable for Class 8 HD truck applications.

Mild Hybrid Configuration for Class 8 HD Trucks

A medium voltage range (48 V) is utilized in a mild hybrid configuration, incorporating a small e-motor and battery. The motor/generator (MG) can be positioned as an integrated starter generator (ISG) at the front of the engine, replacing the conventional starter and alternator, or directly connected to the crankshaft between the engine and the clutch via a belt [109, 110]. The small e-motor and battery serve several functions in a mild hybrid configuration, including engine start/stop during idling, regenerative braking, drivetrain torque enhancement, and powering electrical accessories [111]. Micro hybrids, a subset of mild hybrid configurations, encompass all the mild hybrid functions, excluding powertrain torque enhancement due to lower operating voltage and e-motor power, making them less applicable for commercial

trucks [112].

The SuperTruck II program utilized 48 V mild hybrid using ISG mounted on the rear power take-off (PTO) as long-haul trucking, implying no need for aggressive hybridization due to the less regenerative braking potential in highway drive cycles [50]. Moreover, mild hybridization needs fewer changes in the main engine and transmission configuration and less battery cost and weight [113]. In the SuperTruck II program, the e-motor ranged from 15 kW to 30 kW for the mild hybrid powertrain, and Li-ion batteries were 7-14 kWh [114–118]. Furthermore, the mild hybrid configuration enables the usage of 48 V accessory systems in the engine, such as the e-turbo, e-WHR, and e-EGR pump [119, 120]. The limitations of using the mild hybrid configuration in long-haul applications include an almost zero all-electric range due to the small size of the e-motor and the high traction torque demand at wheels [100]. In addition, auxiliary loads are constrained during engine-off operation, and the battery capacity is limited for storing regenerative braking energy.

According to the literature, fuel consumption reductions achieved by mild hybrid Class 8 HD trucks, relative to their conventional counterparts with the same diesel engine, range from 1% to 2.9% over regional- and long-haul drive cycles under full cargo load conditions [19, 119, 121]. However, over urban drive cycles, these reductions become more pronounced, reaching 10.5% to 11.1%, due to increased regenerative braking opportunities and more transient driving conditions [110, 112]. These reported reductions are based on evaluations conducted over either real-world drive cycles that account for varying road grade profiles or standard drive cycles that neglect the effects of road grades.

Series Hybrid Configuration for Class 8 HD Trucks

In series hybrid configurations, the e-motor drives the wheels, and the ICE powers a generator to recharge the battery, as depicted in Figure 1.5(a). The ICE can be controlled to operate at high-efficiency steady-state conditions as it does not need to meet instantaneous load demand [108]. The mechanical energy generated by the ICE is converted to electrical energy by the generator, which can both charge the battery and power the e-motor for propulsion [100]. A transmission can be utilized to maintain crawling ability at lower speeds comparable to conventional trucks [122].

The series hybrid configuration has two modes: 1) all-electric mode with the ICE off and the battery powering the e-motor for propulsion, and 2) range-extender mode with the ICE providing power through the generator to the e-motor and driving wheels [123, 124]. For Class 8 HD trucks, operating in the first mode enables lower

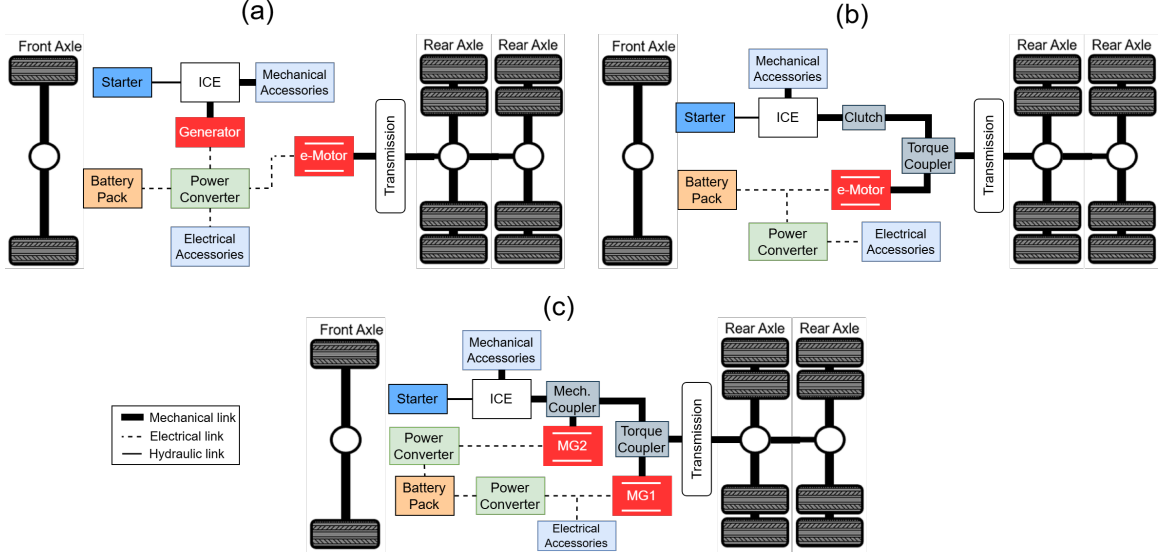


Figure 1.5: Powertrain configurations of (a) series hybrid, (b) pre-transmission parallel (P_2) hybrid, and (c) series-parallel hybrid for Class 8 HD trucks.

fuel consumption and emissions by relying solely on the battery power. However, in power-demanding situations like highway driving with limited regenerative brake opportunities, the powertrain cannot sustain all-electric mode for long distances, resulting in rapid battery charge depletion [100, 125]. In range-extender mode, the ICE must remain active to charge the battery or power the e-motor. The absence of a mechanical connection between the ICE and the driving wheels results in a multi-stage energy conversion process, in which fuel energy is first converted into mechanical (shaft) work, then into electrical energy, and finally back into mechanical power delivered to the wheels [108].

Studies in the literature [82, 126] report that series hybrid Class 8 HD truck configurations exhibit 5% to 16% higher fuel consumption compared to their conventional counterparts with the same diesel engine over regional- and long-haul drive cycles under full cargo load conditions. Limited regenerative braking opportunities over highway force the series hybrid configuration to operate predominantly in range-extender mode, in which additional energy conversion losses occur due to the multi-stage conversion of energy between mechanical and electrical domains [61, 66]. However, under urban drive cycles, including frequent braking and acceleration events, the fuel consumption of series hybrid configurations has been reported to improve by up to 18% relative to conventional counterparts [127]. Since the majority of fuel consumption and GHG emissions in Class 8 HD trucks occur during highway driving, the efficiency penalty associated with the series hybrid configuration under these conditions makes

it unsuitable for such applications.

Parallel Hybrid Configuration for Class 8 HD Trucks

In a parallel hybrid electric configuration, both the ICE and e-motor have mechanical connections to the transmission and driving wheels as depicted in Figure 1.5(b). The ICE serves as the primary power source, while the e-motor assists during acceleration and recovers energy during dynamic braking [128]. Compared to series hybrid designs, parallel hybrids typically have a smaller battery and e-motor as it does not need to meet the full load demand [105]. This offers operational flexibility, including ICE-only, all-electric, and hybrid modes.

For Class 8 HD trucks, parallel hybrid configurations perform better on highways than series hybrids because the ICE power can directly transmit to the wheels without multi-stage energy conversion or needing a heavy generator [105, 129]. Parallel hybrids require more complex control strategies to keep the ICE operating in high-efficiency zones. The battery state of charge (SOC) should be maintained at the same level at the beginning and end of the trip by recovering regenerative brake energy or charging from the ICE [128].

Parallel hybrid configurations include a range of different e-motor sizes and locations. They can be classified into four sub-groups based on the e-motor placement: P_0 , P_1 , P_2 , P_3 , and P_4 [130] as shown in Figure 1.6. In the P_0 setup, the alternator and conventional starter are replaced with a small e-motor acting as an ISG in the form of a micro hybrid [100, 131]. In the P_1 setup, which is mostly used in mild hybrid configurations, the e-motor is directly connected to the crankshaft [132]. In the P_2 setup, the e-motor can either be side-attached using a belt or integrated into the transmission input shaft via a clutch [128, 133]. It is also known as the pre-transmission parallel (P_2) hybrid configuration. In the P_3 setup, the e-motor is installed on the output shaft of the transmission and is also named as the post-transmission parallel hybrid configuration [100, 134]. In the P_4 setup, the e-motor is placed separately on the rear axle as an e-drive system, adding design complexity and cost [130].

These parallel hybrid configurations have advantages and disadvantages, summarized in Figure 1.6. Overall, P_2 and P_3 are the most commonly used parallel hybrid designs for HD trucks. P_3 has two disadvantages compared to P_2 : 1) the transmission is disengaged during the all-electric mode, and 2) the vibrations from wheels are directly transmitted to the e-motor, making the design less durable. The main advantage of P_3 over P_2 is that performance of the e-motor in P_2 is highly reliant on the gearbox ratio and shifting strategy [100, 132]. In terms of Class 8 HD trucks,

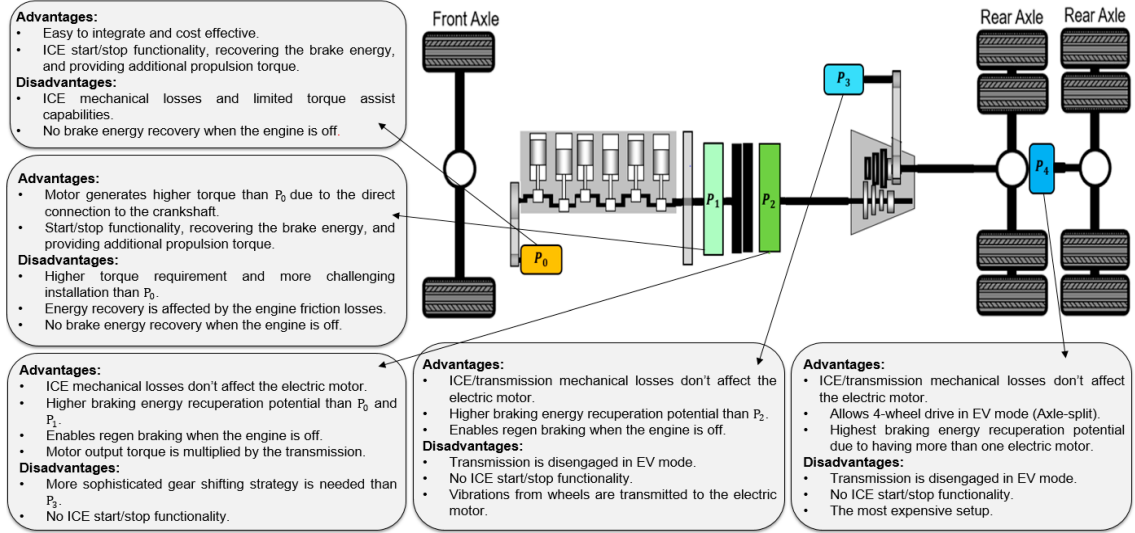


Figure 1.6: Schematic representation of various parallel hybrid configurations (P_0 , P_1 , P_2 , P_3 , and P_4) based on the e-motor placement for Class 8 HD trucks, along with the key advantages and disadvantages of each setup. Reprinted from my previously published work in [1], licensed under CC BY 4.0.

which are mostly driven over highways, the fuel consumptions of P_2 and P_3 are very close because the transmission is often engaged in lower gear ratios during cruising speed, and gear shifting is not conducted frequently compared to urban driving [128].

Studies in [19, 104, 127, 128] reported fuel consumption reductions of 2.1% to 8% for P_2 hybrid Class 8 truck configurations relative to their conventional counterparts with the same diesel engine over regional- and long-haul drive cycles under full cargo load conditions. However, these reductions become more pronounced under urban drive cycles, reaching 27.5% to 34.5%. The fuel savings potential is highly dependent on the powertrain configuration, the driving/duty cycle, the payload, and the employed control strategy. Much of this benefit is control optimized to operate the engine in high-efficiency regions, which can improve cycle-average engine BTE by 2% to 4% (absolute value) over regional- or long-haul drive cycles [19, 121]. The truck's trajectory path can be predicted through advanced optimization-based control strategies, and the energy consumed by the ICE to sustain battery charge would be minimized by maximizing the opportunities for regenerative braking during the trip [105, 125].

Series-Parallel Hybrid Configuration for Class 8 HD Trucks

Series-parallel hybrid electric configurations employ a conventional transmission system and multiple clutches (mechanical couplers) to leverage the advantages of both

series and parallel hybrid designs [135]. In such architectures, two electric machines (MG1 and MG2) are typically utilized, where MG2 primarily functions as a generator and MG1 serves as the main traction motor [136], as shown in Figure 1.5(c). MG2 operates mainly in generation mode to regulate the engine speed within its high-efficiency region. Transitions between series and parallel modes are governed by clutch states, which can introduce interruptions in continuous power delivery to the driving wheels [105, 137]. Therefore, frequent switching between these modes is generally limited.

The power-split hybrid is a variant of series-parallel hybrid configurations in which the conventional transmission system and torque coupler are replaced by a planetary gearset acting as a power-splitting device [138]. The planetary gearset kinematically couples the ICE, MG1, and MG2, enabling a continuous and simultaneous distribution of power between mechanical and electrical paths [139]. In this architecture, both MG1 and MG2 can operate in generating or motoring states, providing greater flexibility in hybrid operational modes [105]. Compared to clutch-based series-parallel configurations, an additional operating mode called the combined power-split mode is enabled, in which the ICE power is continuously divided into a mechanical path to the wheels and an electrical path through MG1 and MG2 [140]. The planetary gearset enforces a continuous relationship between the speeds and torques of the ICE and MGs, enabling smooth power transitions and electronic continuously variable transmission (e-CVT) functionality [141, 142]. However, power-split hybrid architectures require more complex control strategies and increased system complexity due to the inclusion of additional power converters and hybrid components compared to parallel and series counterparts [138]. In addition, designing a durable planetary gearset for HD truck applications capable of operating under high torque and power ratios is a challenge [135, 141, 142].

Only a limited number of studies in the literature evaluated the performance of series-parallel or power-split hybrid electric configurations for Class 8 HD trucks. Due to significant fluctuations in propelling torque, cargo load, and diverse driving cycles in class 8 HD trucks, power-split designs with a single planetary gearset may not consistently maintain the ICE and e-motors operations in their high-efficiency zones [142]. To address this limitation, complex power-split designs that incorporate either multiple planetary gearsets or a single planetary gearset paired with a multi-speed gearbox, can satisfy the torque requirement of HD trucks [143, 144]. The study in [142] showed that a power-split hybrid configuration equipped with a planetary gearset coupled with a two-speed automatic gearbox achieves over 10% fuel

consumption reduction relative to a conventional truck with the same diesel engine over long-haul drive cycles. In contrast, power-split designs featuring multiple interconnected planetary gearsets without multi-speed gearboxes have demonstrated fuel consumption reductions ranging from 5% [145] to 8.7% [82, 126]. The study in [146] reported that series-parallel hybrid configurations without a planetary gearset but equipped with a multi-speed gearbox achieve fuel consumption reductions of 5.6% to 7.4% relative to conventional counterparts over real-world long-haul drive cycles.

Plug-in Hybrid Configuration for Class 8 HD Trucks

This architecture represents a transitional step toward fully electric vehicles. A key feature of plug-in hybrid systems is the ability to charge the battery from an external electrical source, such as the grid [147]. The use of larger batteries and higher-power e-motors in plug-in hybrid designs enables the ICE to operate more frequently within its high-efficiency region, resulting in reduced fuel consumption and lower tailpipe GHG emissions [105, 148]. However, increasing the all-electric range through larger batteries leads to additional weight and cost penalties [149, 150]. Two potential plug-in hybrid architectures for Class 8 HD trucks are series plug-in and power-split plug-in configurations. These architectures are similar to their non-plug-in counterparts, with the key distinction that they incorporate larger batteries capable of off-board charging, thereby enabling extended operation in all-electric mode [19, 104]. The operational and control complexity of plug-in hybrid configurations is higher than that of non-plug-in systems, but they offer greater potential for fuel economy improvement [105, 151].

Plug-in hybrid electric configurations offer the potential to downsize the engine without sacrificing performance in many situations, offering engine cost and weight reductions [19, 104]. These can partially offset the weight and cost penalties of these configuration and can help improve engine efficiency through lower friction losses. Battery sizing for plug-in hybrid Class 8 HD trucks is a complex design problem, one approach in the literature is to size the battery to achieve almost half of the expected DDR in all-electric mode [19, 126]. Route specifications of Class 8 HD trucks between cargo depots can be analyzed to determine the share of electric driving and to appropriately size the engine and battery for a given application.

Few studies explored the fuel saving potential of plug-in hybrid electric configurations for Class 8 HD truck applications. The study in [104] designed a series plug-in hybrid configuration equipped with a downsized diesel engine and employed a blended charge-depleting strategy. The results showed a 40% reduction in fuel consumption

relative to a conventional diesel counterpart over long-haul drive cycles under full cargo load, accounting for both electrical and chemical (fuel) energy supplied to the vehicle. Under urban drive cycles, the reduction increased to nearly 50%. In plug-in hybrid configurations, a blended charge-depleting strategy [148], in which the ICE contributes to meet the vehicle power demand while the battery is being depleted, generally results in better fuel consumption performance compared to the charge-sustaining (CS) mode. In the CS mode, the battery is first depleted, after which the ICE is engaged to maintain the battery SOC at a minimum threshold [105]. Prada et al. [19] reported that a plug-in hybrid Class 8 HD truck operating exclusively in the CS mode exhibits 26% worse fuel economy over highway drive cycles compared to the conventional baseline under full cargo load conditions. In contrast, the use of a plug-in blended charge-depleting strategy leads to a 1.5% to 2% improvement in engine BTE over long-haul drive cycles compared to parallel hybrid configurations, and a 3% to 5% improvement relative to conventional trucks [104, 152].

Benchmarking of Hybridized Class 8 Truck Models Reported in the Literature

To compare the fuel saving potential of various hybrid electric Class 8 HD truck configurations, a benchmarking of the reviewed hybridized configurations reported in the literature is presented in Table 1.5. The fuel saving potential of these configurations is presented relative to conventional baseline models equipped with the same type of diesel engine. Moreover, the types of drive cycles, either experimental, including the road gradient, or standard without the road gradient, are indicated in the table. Finally, the implementation of engine downsizing for each configuration reported in the literature is indicated in the last column.

1.2.5 Propulsion System Performance Criteria for Powertrain Component Sizing of Class 8 HD Trucks

For a proper component sizing of a newly designed vehicle, it is critical to specify the required performance criteria to ensure that the vehicle’s propulsion system can effectively operate on different roads and adapt to varying driving conditions. The critical propulsion system requirements often include: 1) minimum acceleration time on a flat road with a fully loaded vehicle, 2) the maximum vehicle speed on a flat road, and 3) maintaining continuous gradeability on an inclined highway [153]. While the propulsion systems of Class 1 to 6 vehicles (GVWR < 11.8 tonne) are sized according to acceleration and top speed performance criteria, due to the higher GVWR,

Table 1.5: Benchmarking of available hybrid electric Class 8 truck configurations reported in the literature, along with their fuel-saving potential relative to conventional baseline models equipped with the same type of diesel engine. All the reported values are based on modeling results.

Ref	Architecture Type	GVWR (Tonne)	Sizes of Powertrain Component			Fuel Saving ¹ (L/100.km) Relative to Conventional Model in Full Cargo Load (%)	Engine Downsizing
			Diesel Eng. (kW)	e-Motor (kW)	Batt. (kWh)		
[66]	P ₂ hybrid	30.2	324	120	15	+4% (Long-haul)	No
	Series hybrid	35.5	391	420	64	−12% (Real-world long-haul)	No
[126]	P ₂ hybrid	35.4	391	220	35	+8% (Real-world long-haul)	No
	Power-split hybrid	35.4	391	220/75	35	+10.1% (Real-world long-haul)	No
[82]	Series hybrid	37.5	391	440	66	−16% (Real-world long-haul)	No
	P ₂ hybrid	37.4	391	240	36	+2.1% (Real-world long-haul)	No
[121]	Mild hybrid	36.3	317	50	5	+2.9% (Standard regional-haul)	No
	Series-parallel hybrid	36.7	317	200/50	25	+5.2% (Standard regional-haul)	No
[19]	Mild hybrid	36.3	400	30	Not disclosed (ND)	+1% (Standard long-haul)	No
	P ₂ hybrid	36.8	400	190	ND	+2.5% (Standard long-haul)	No
	Series plug-in hybrid	41.4	225	650	ND	+35% (Standard long-haul)	Yes
	Mild hybrid	ND	335	10	1	+1% (Standard long-haul)	No
[104]	P ₂ hybrid	ND	345	90	12	+3% (Standard regional-haul)	No
	Series plug-in hybrid	ND	275	660	880	+43% (Standard long-haul)	Yes
[146]	Series hybrid	ND	411	360	23	+4.1% (Real-world regional-haul)	No
	P ₂ hybrid	ND	325	375	21	+7.2% (Real-world regional-haul)	Yes
	Series-parallel hybrid	ND	411	196/196	21	+8.2% (Real-world regional-haul)	Yes

¹Fuel-saving percentages are calculated relative to the corresponding conventional models. Positive values indicate reduced fuel consumption compared to the conventional counterparts, while negative values indicate increased fuel consumption.

propulsion systems of Class 7 and 8 HD trucks are sized based on gradeability performance [154]. Depending on every specific application and cargo carrying capacity, Class 8 tractor-trailers come with a wide variety of engine rated powers. Figure 1.7 illustrates the engine power spectrum of conventional Class 8 tractor-trailers (regional-haul and long-haul), among the eleven major HD truck OEMs worldwide. This figure shows a broad power range for the Class 8 trucks diesel engines, ranging from 195 kW to 580 kW.

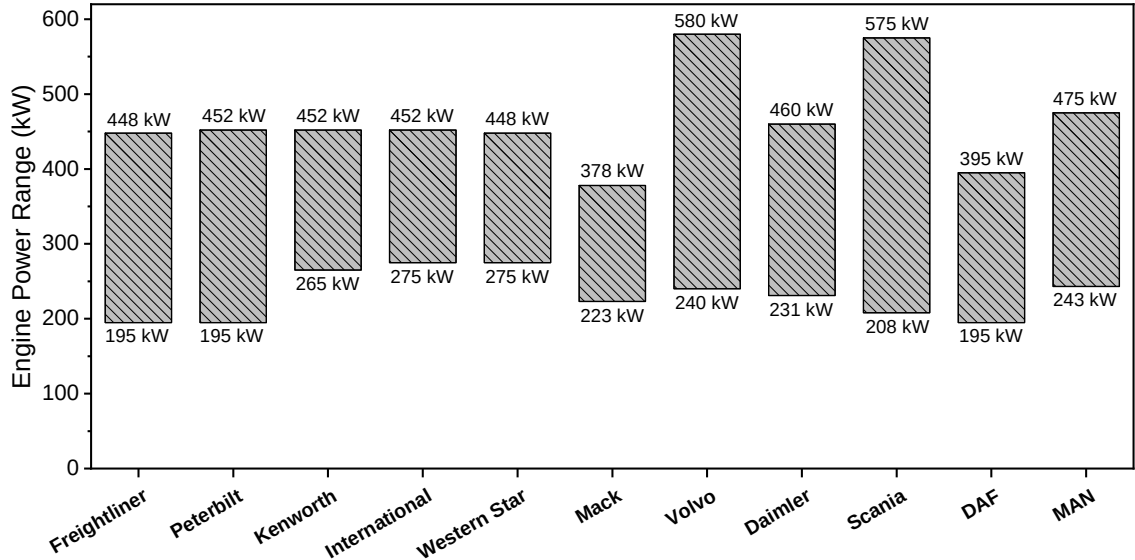


Figure 1.7: Engine power range of conventional Class 8 tractor-trailers (regional-haul and long-haul), from the eleven major HD truck manufacturers worldwide [155–164]. Reprinted from my previously published work in [3], licensed under CC BY 4.0.

The wide range of engine rated powers in Figure 1.7 is due to the diversity in commercial applications and a broad GVWR range of class 8 HD trucks. The existing standard to specify the propulsion system requirements of conventional Class 8 tractor-trailers includes the society of automotive engineers (SAE) J2807 standard [165]. According to this standard, propulsion systems must satisfy specific acceleration, launch, and gradeability criteria, encompassing: 1) achieving level road acceleration from 0 to 96 km/h (i.e., 60 mph) within 65 seconds, 2) attaining level road acceleration from 0 to 48 km/h (i.e., 30 mph) in 18 seconds, 3) accomplishing level road acceleration from 64 (i.e., 40 mph) to 96 km/h (i.e., 60 mph) in 21 seconds, and 4) demonstrating highway gradeability to sustain a minimum constant speed of 56 km/h (i.e., 35 mph) over the Arizona-SR-68 (Davis Dam) road. Davis Dam road features one of the steepest routes in the U.S., with a maximum road grade reaching 7% [19, 165]. The evaluation specifically focuses on the initial 19.7 km of the 44.4-km-

long Davis Dam road, representing the steepest section for testing the gradeability of conventional HD trucks.

When designing a hybrid electric Class 8 truck powertrain, it is crucial to establish performance criteria that are comparable to conventional trucks [166]. Gradeability is a key factor in powertrain component sizing of Class 8 HD trucks. For example, if the conventional baseline truck can navigate a specific steep route at a constant speed of 56 km/h with 35% cargo load, the newly designed hybridized configuration should replicate this performance under identical cargo load conditions.

Previous studies on gradeability testing of Class 8 regional- and long-haul trucks, whether hybrid or conventional, have primarily focused on fixed durations/distances at constant road gradients and vehicle speeds. Typical assessments involve 5-6% grades over approximately 20 minutes, with vehicle speeds maintained at around 50-60 km/h. Prada et al. [19] conducted a 20-minute steady climb on a constant 6% road gradient at speeds of 51.2 km/h and 49.6 km/h for hybrid and conventional Class 8 HD trucks. Garcia et al. [82] undertook a 17.6 km climb on a constant 5% gradient at a sustained speed of 60 km/h for hybrid Class 8 regional-haul trucks. Ganaoui et al. [167] used a 20-minute climb with a constant grade of 5%, maintaining a speed of 50 km/h, focusing on gradeability tests for conventional and hybrid Class 8 trucks. The fixed grade is a result of a lack of actual road grade versus distance data from global positioning systems (GPS) [104, 168]. A dynamically changing road grade profile like Davis Dam road, as specified by the SAE J2807 standard, offers a more realistic approach for evaluating the gradeability performance of HD trucks compared to a constant road grade [165].

In evaluating the gradeability of conventional trucks, the critical parameter is the maximum road gradient. Conventional vehicles rely exclusively on the ICE, which under normal conditions can deliver a steady power as long as fuel is available. Therefore, if the vehicle's propulsion system can meet power demands on the maximum road slopes, it inherently possesses the capability to provide ample power within lower slope variations [169]. However, in hybrid propulsion systems, which incorporate a secondary power source like the high-voltage battery besides the ICE, the secondary non-fossil-based power source must have the capacity for providing the remaining road power demands. This ensures that the performance criteria remain at least on par with the conventional baseline model [19, 104]. Therefore, for testing gradeability of hybrid electric HD trucks, besides considering the maximum road gradient, other crucial parameters come into play, including: 1) the length of the longest continuous distance with a slope surpassing the ICE rated power to meet road power

demands, and 2) portion of the regenerative brake opportunities throughout the road by predetermining the trip’s starting and ending points.

1.3 Integration of Powertrain Hybridization with Low- and Zero-Carbon Combustion Engines

Existing studies in the literature have focused primarily on integrating advanced diesel engine technologies with powertrain hybridization to reduce fuel consumption and GHG emissions in Class 8 HD trucks. In contrast, relatively limited attention has been given to the use of low- or zero-carbon-fueled ICEs, such as those powered by NG or H₂. The integration of NG- or H₂-ICEs, with overall efficiencies comparable to modern diesel engines, into hybrid powertrains can achieve fuel saving benefits while further reducing tailpipe GHG emissions. This section reviews existing studies that investigate the integration of NG and H₂ engines with powertrain hybridization for Class 8 HD tractor-trailer trucks.

1.3.1 Hybrid Powertrains Equipped with Natural Gas Engines

The literature includes a limited number of studies on the design of hybrid electric Class 8 HD truck configurations equipped with NG engines. Zhao et al. [66] reported a modest 0.3% reduction in diesel-equivalent fuel consumption for a P₂ hybrid Class 8 truck configuration equipped with an HPDI NG engine, relative to a conventional diesel baseline model with an identical rated engine power of 324 kW over long-haul drive cycles. The corresponding reduction in tailpipe CO₂ emissions was reported to be 23% under the full cargo load condition. In contrast, a P₂ hybrid configuration equipped with a 324 kW, PFI NG-SI engine exhibited 20.9% higher fuel consumption than the conventional diesel baseline, while achieving a CO₂ emission reduction of 8.2%. The higher fuel consumption of the hybrid model relative to the conventional diesel counterpart can be attributed to the lower overall BTE of NG engines compared to diesel. The overall engine BTE of the PFI NG-SI engine was 8–9% lower than that of the diesel engine, while the HPDI NG engine exhibited an 1–2% lower BTE.

In another study combining NG engines with powertrain hybridization for Class 8 HD trucks, Gao et al. [170] compared the equivalent fuel consumption of a P₂ hybrid configuration equipped with a 15 L PFI NG-SI engine (475 kW) to that of a conventional model equipped with a 15 L Cummins ISX diesel engine of the same rated power. Simulation results over a long-haul drive cycle under full cargo load conditions

showed a 2.8% increase in fuel consumption, measured in L/100,km, for the hybrid model relative to the conventional counterpart. However, over urban drive cycles, the fuel consumption of the hybrid model decreased by up to 14.6% compared to the conventional model, primarily due to greater availability of regenerative braking. The higher fuel consumption of the hybrid model under highway conditions was attributed to the 3–4% lower cycle-average BTE of the NG-SI engine compared to the diesel engine in the conventional model. As a limitation, the study did not provide any evaluation of CO₂ and CH₄ emissions for the proposed configurations.

In a recent study, Liu et al. [146] designed P₂ and series–parallel hybrid Class 8 truck configurations equipped with a PFI NG-SI engine. Fuel consumption and tailpipe CO₂ emissions were evaluated over a regional-haul drive cycle relative to a conventional diesel model with the same rated engine power as the hybrid configurations. The P₂ and series–parallel hybrid designs exhibited increases in fuel consumption of 5.5% and 4.1%, respectively, measured in L/100 km, compared to the diesel baseline under full cargo load conditions. However, tailpipe CO₂ emissions decreased by 18.4% and 20.6%, respectively, primarily due to the lower carbon content of NG compared to diesel. The higher fuel consumption of the hybridized models was mainly attributed to the 5–7% lower overall engine BTE of the 15 L, 411 kW NG-SI engine compared to the 13 L diesel engine with the same rated power. In other words, the marginal benefits of regenerative braking during highway cruising in the hybridized configurations can be offset, or even outweighed, by the lower average engine BTE over the drive cycle. No evaluation was performed regarding the potential CH₄ emissions from the NG-SI engine. Moreover, the effects of the hybrid powertrain on engine operation, as well as the interactions between the engine and the hybrid system, were not explained.

1.3.2 Hybrid Powertrains Equipped with Hydrogen Engines

Some industrial demonstration projects have resulted in the development of prototype H₂-SI engines (PFI or DI) for HD truck applications. Notable examples include the Weichai HD truck equipped with a 13 L DI H₂-SI engine, as well as the DAF XF truck [155] and the MAN hTGX truck [171]. Another example is the Southwest Research Institute, which recently launched the H₂-ICE Consortium, developing a Class 8 demonstration truck with an PFI H₂-SI engine aimed at near-zero CO₂ emissions and ultra-low NO_x levels [172]. However, there remains a lack of research and demonstration on the integration of these engines with other advanced technologies, such as engine performance enhancements or powertrain hybridization.

Few modeling works have examined the role of H₂-fueled ICEs in conventional HD trucks. The study in [90] presented an AVL concept and associated test results for a DI H₂-SI engine implemented in a 40-tonne Class 8 HD truck, and compared its performance with that of a fuel-cell truck over both highway and urban drive cycles. The results indicated that the H₂-ICE consumed 7% and 77% more hydrogen than the fuel-cell system under highway and urban conditions, respectively. To mitigate this higher H₂ consumption, AVL estimated that the application of powertrain hybridization and a WHR system could provide fuel consumption reductions of 6% and 2%, respectively, for long-haul applications. Additionally, the study in [173] designed a parallel hybrid powertrain and simulated a 25-tonne H₂-ICE truck over the China Heavy-duty Commercial Vehicle Test Cycle, showing that the hybrid configuration reduced fuel consumption by 20% compared to a conventional H₂-ICE truck. H₂-ICE hybrid electric powertrains are therefore considered a promising pathway for future commercial vehicle applications [174], although current research in this area remains limited.

In a study conducted by Liu et al. [146], a series-parallel hybrid Class 8 truck configuration equipped with a 16.8 L, PFI H₂-SI engine (411 kW) was modeled. Simulation results indicated that the H₂-ICE hybrid configuration achieved up to 9.2% reductions in diesel-equivalent fuel consumption, expressed in L/100 km, relative to a conventional model equipped with a 13 L diesel engine with the same rated power over a regional-haul drive cycle. In addition, the hybrid configuration achieved zero tailpipe CO₂ emissions. Although the overall BTE of the H₂-SI engine was 1–2% lower than that of the diesel engine, the hybrid design enabled the engine to operate predominantly within its high-efficiency region while recovering regenerative braking energy, resulting in improved fuel economy compared to the conventional diesel counterpart. As a limitation, the study did not provide a detailed comparison of the underlying technologies and design specifications of the H₂ and diesel ICEs to justify the differences in their overall efficiencies.

1.3.3 Identified Research Gaps and Proposed Solutions

This section highlights key research gaps in the so far reviewed literature concerning the integration of NG (low-carbon) and H₂ (zero-carbon) ICEs with full powertrain hybridization aimed at reducing fuel consumption and emissions in Class 8 HD trucks. Then, potential solutions to address these gaps are presented from powertrain and engine design perspectives.

Research Gaps

1- Hybrid Electric Hydrogen-Fueled CI Engine Powertrain: There is a lack of studies on the design of hybrid electric powertrain configurations equipped with near-zero-carbon H₂-CI engines capable of achieving overall thermal efficiencies comparable to or higher than those of modern diesel engines. Previous studies have primarily focused on H₂-SI engines (PFI or DI), which typically exhibit 1-5% lower overall BTE than equivalent diesel engines with the same rated power. The lower overall BTE of H₂-SI engines limits the effectiveness of powertrain hybridization in achieving diesel-equivalent fuel consumption reductions relative to conventional diesel configurations.

2- Hybrid Electric Natural Gas-Fueled CI Engine Powertrain: There is a lack of studies on the design of hybrid electric powertrain configurations equipped with low-carbon NG-CI engines capable of achieving overall thermal efficiencies comparable to or higher than those of modern diesel engines. Previous studies have primarily focused on NG-SI (PFI or DI), which typically exhibit 3-9% lower overall BTE than equivalent diesel engines with the same rated power.

There is a lack of comprehensive assessment of the effects of incorporating advanced technologies into NG- and H₂-CI engines to improve BTE, particularly at mid- to high-torque levels, where the engine operates for the majority of its time in hybridized powertrains. The reviewed literature does not provide sufficient details regarding the specific engine technology enhancements used to achieve the reported efficiencies of NG- and H₂-fueled ICEs.

3- Engine-Out Emissions Assessment: There is a lack of studies assessing potential engine-out emissions from NG- or H₂-ICEs in Class 8 conventional and hybridized truck configurations under real-world highway and urban drive cycles across different cargo load conditions. This includes CO₂, CH₄, and NO_x emissions from NG-ICEs, as well as tailpipe CO₂ emissions from H₂-CI engines associated with the use of diesel pilot fuel for ignition.

4- Engine-Specific Hybrid Powertrain Design and Optimization: In the reviewed literature, hybrid powertrain configurations have not been specifically designed or optimized to account for the distinct characteristics of NG- or H₂-fueled ICEs. Instead, those configurations were typically developed based on diesel engines, after which the diesel engine was replaced with an NG- or H₂-ICE without corresponding modifications to the hybrid powertrain architecture or its control strategy. This leads to suboptimal system performance, as the potential benefits of fuel-specific combustion characteristics and optimal operating regions are not fully exploited.

5- Real-World Gradeability Assessment for Powertrain Component Sizing: There is a lack of studies incorporating real-world steep road grade profiles into gradeability assessment for the proper sizing of propulsion system components in hybridized Class 8 truck configurations. The reviewed literature employ constant road grade assumptions (typically 5-7%) or utilize experimentally created regional- or long-haul drive cycles developed primarily for fuel consumption and emissions analysis, which include relatively mild road grades (typically below 5%). These assumptions can lead to suboptimal sizing of key powertrain components, particularly the propulsion system elements such as the engine, high-voltage battery, and e-motor(s) in hybrid configurations. As a result, comparable propulsion performance to conventional models may not be achieved across a wide range of real-world driving conditions.

Proposed Solutions

To address the identified research gaps, this thesis creates a vehicle-level design framework for modeling conventional and hybrid electric Class 8 truck configurations equipped with high-efficiency NG- or H₂-fueled CI engines. The proposed solutions to address the research gaps are summarized as follows:

- To reduce net powertrain energy consumption and achieve near-zero tailpipe GHG emissions, both conventional and hybrid electric truck configurations are designed and optimized using a commercially available HPDI H₂ CI engine model. New models are created to simulate the HPDI H₂ engine and the auxiliary load imposed on the engine by the on-board hydrogen compression system to maintain a 300 bar hydrogen injection pressure as on-board storage tanks are depleted.
- To design conventional and hybrid electric truck configurations equipped with high-efficiency low-carbon-fueled engines, different types of commercially available NG-CI engine models are incorporated into the vehicle-level design framework. These engines include a dual-fuel diesel-pilot premixed NG engine and different technology-enhanced variants of the HPDI NG engine. The incorporated technologies and engine modifications include increasing the compression ratio, advancing the start of injection, and integrating a turbo-compound system, which primarily improve engine efficiency at mid- to high-torque levels where engines in hybridized configurations predominantly operate.
- Incorporating experimentally validated emission maps of the considered NG- and H₂-ICEs into the vehicle models enables assessment of engine-out CO₂,

NO_x, and CH₄ (for NG-ICEs) emissions over both highway and urban drive cycles under different cargo load conditions. For this purpose, a real-world representative regional-haul drive cycles is created based on the collected data from on-road testing of a real Class 8 diesel truck.

- A multi-stage component sizing algorithm is created to optimize the sizing of the primary hybrid powertrain components and tune key control strategy parameters specifically for each considered NG- or H₂-ICE. This algorithm aims to minimize net powertrain energy consumption over a real-world regional-haul drive cycle while maintaining propulsion system performance criteria comparable to those of the conventional diesel baseline model. Three full hybrid powertrain architectures with high potential for reducing fuel consumption and GHG emissions are designed and specifically tailored to satisfy the high torque demands of Class 8 truck applications. These hybrid powertrains include a pre-transmission parallel (P₂) hybrid, a complex power-split hybrid, and a series plug-in hybrid configuration.
- To test gradeability, the most deterministic criterion for assessing propulsion system performance of the designed Class 8 truck configurations, the real-world varying road grade profiles of 15 identified steep highways worldwide are derived. The optimal powertrain component sizing of the hybrid electric truck models is then achieved using each newly identified steep highway as the gradeability criterion. This approach ensures propulsion performance comparable to conventional models across a wide range of real-world driving conditions while also enabling assessment of the feasibility of engine downsizing in hybridized Class 8 truck configurations.

1.4 Thesis Outline and Contributions

In response to the previously identified research gaps, the main contributions of this thesis are organized into six chapters, as shown in Figure 1.8, followed by a conclusion section. A summary of the key contributions of each chapter is as follows:

- Chapter 2: Conventional Vehicle Modeling and Validation
 - Development and validation of a conventional baseline model for a Class 8 regional-haul truck using real-world data collected from a tested Peterbilt 579 equipped with a 2019 Cummins X15 diesel engine.

- Creating a real-world regional-haul drive cycle including a varying road grade profile based on collected experimental data, for the assessment of fuel consumption and emissions of Class 8 tractor-trailer HD trucks.
- Development of a conventional natural gas (NG) truck model by replacing the diesel engine in the conventional baseline configuration with a diesel-pilot HPDI NG engine of the same rated power, along with corresponding modifications to the on-board fuel system.
- Chapter 3: Design of Class 8 Hybrid Electric Truck Configurations
 - Modeling of full hybrid electric powertrain configurations specifically for Class 8 HD truck applications to meet high torque demands at the driving wheels. These include: a pre-transmission parallel (P_2) hybrid, a complex power-split hybrid, and a series plug-in hybrid electric configuration.
 - Design of control strategies for the energy management system (EMS) for each proposed hybrid configuration to manage power distribution between the engine and electrical drivetrain. This includes constrained rule-based strategies for the non-plug-in hybrid configurations and a dynamic programming (DP)-based strategy for the plug-in hybrid configuration.
- Chapter 4: Optimal Powertrain Component Sizing and Performance Evaluation
 - Development of a multi-stage component sizing framework to optimize the primary sizes of hybrid powertrain components while tuning the critical EMS-related variables. This framework aims to minimize fuel consumption over a real-world highway drive cycle while ensuring performance comparable to that of the conventional baseline model.
 - Formulation of a preliminary component sizing procedure for each proposed hybrid configuration to determine the variation ranges of the optimization variables within the multi-stage component sizing framework.
- Chapter 5: Real-World Steep Drive Cycles for Gradeability Testing of Class 8 Hybrid Electric Trucks
 - Evaluation of the effect of real-world steep road grade profiles on gradeability testing and optimal powertrain component sizing of conventional and hybridized Class 8 truck configurations. The gradeability criterion is defined as the primary determinant of propulsion system performance in HD trucks.

- Identifying 15 new steep highways worldwide suitable for Class 8 HD trucks and obtaining their grade versus distance profiles allows for assessing the gradeability of the designed HD truck configurations. A dataset for these 15 highways is created and shared publicly through this thesis for use by scholars and vehicle designers.
- Assessment of the feasibility of engine downsizing for non-plug-in hybrid electric Class 8 truck configurations while maintaining propulsion system performance comparable to that of the conventional baseline model.
- Chapter 6: Performance Assessment of Class 8 Hybrid Electric Trucks Equipped with Low-Carbon Natural Gas Engines
 - Integration of powertrain hybridization with low-carbon NG-fueled CI engines to assess the potential for reducing energy consumption and engine-out emissions (CO_2 , CH_4 , and NO_x) in Class 8 regional-haul truck applications.
 - Representation of fuel consumption and emissions maps for different variants of dual-fuel NG–diesel engines, incorporating various technology packages that achieve overall efficiencies comparable to or exceeding that of the baseline diesel engine used in this study. The dual-fuel CI engines considered are categorized into a premixed NG–diesel engine and multiple variants of a diesel-pilot HPDI NG engine.
 - Comparative evaluation of the designed hybrid electric configurations, equipped with different NG engines, focusing on engine operating points, component-level energy losses, and powertrain efficiencies over highway and urban drive cycles under varying cargo load conditions.
- Chapter 7: Energy Saving and Near-Zero Greenhouse Gas Emissions in Hybrid Electric Regional-Haul Trucks Equipped with a High-Pressure Direct Injection Hydrogen Engine
 - Assessment of fuel consumption and tailpipe CO_2 emissions for conventional and hybridized Class 8 HD trucks equipped with an HPDI H_2 engine over real-world regional-haul and standard urban drive cycles under different cargo load conditions.
 - Quantifying the parasitic auxiliary load of a mechanically driven on-board hydrogen compression system required to maintain a 300-bar injection

pressure in HPDI H₂ engines for both conventional and P₂ hybrid Class 8 regional-haul trucks.

- Comparative evaluation of the operational efficiencies of the primary powertrain components among the designed conventional and hybridized truck configurations across different drive cycles and cargo load conditions.

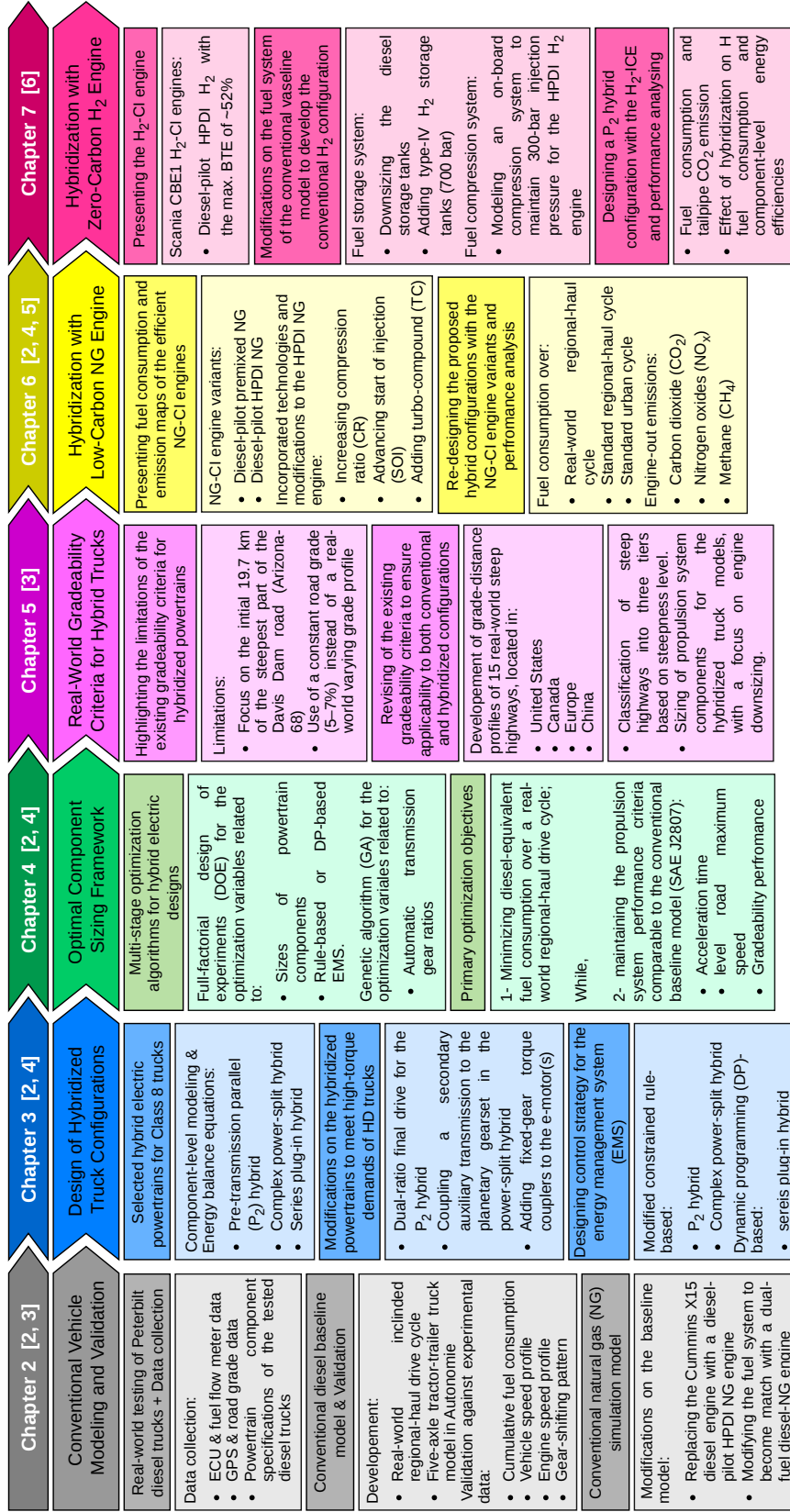


Figure 1.8: Overview of the thesis structure, illustrating the main contributions and the progression of each chapter. Resulting journal publications/submissions from each chapter are listed in [2–6].

Chapter 2

Conventional Vehicle Modeling and Validation¹

The integration of high-efficiency low- or zero-carbon-fueled ICEs with powertrain hybridization was introduced in Chapter 1 as a promising near-term pathway to reduce fuel consumption and GHG emissions in Class 8 HD trucks. As a first step toward this evaluation, a validated baseline vehicle model is required to accurately represent the energy consumption and GHG emissions of real-world Class 8 conventional diesel trucks. Building upon this baseline, conventional truck models equipped with low- or zero-carbon ICEs can be developed by replacing the engine and modifying the on-board fuel system, while maintaining consistency in the remaining vehicle and driveline components. This chapter presents a system-level vehicle modeling and validation framework for Class 8 conventional truck configurations equipped with either a diesel or a NG-CI engine. In addition, a real-world regional-haul drive cycle is developed based on data collected from on-road testing of a Class 8 Peterbilt diesel truck over a highway route.

2.1 Introduction and Background

In addition to the operational efficiency and type of powertrain components in Class 8 conventional trucks, fuel consumption and emissions are highly sensitive to the characteristics of the driving route. Real-world drive cycles introduce greater transients in engine speed and torque due to varying road grade profiles, as well as increased fluctuations in vehicle speed resulting from interactions with surrounding traffic (e.g., cut-in and cut-out events) [15]. These factors enable a more realistic assessment of vehicle performance compared to standard drive cycles that do not account for road

¹This chapter is partially based on the published journal papers in [2, 3].

grade variations and traffic interaction effects. The key characteristics of a newly developed regional- or long-haul drive cycle for evaluating the performance of Class 8 HD trucks should fall within the ranges specified by the National Renewable Energy Laboratory (NREL) [175], as summarized in Table 2.1. These ranges are derived from an extensive dataset comprising 65,371 trips collected from 487 Class 8 trucks in 2022.

Table 2.1: Summary of key characteristics of real-world regional- and long-haul drive cycles for Class 8 conventional trucks based on NREL large-scale dataset, reported from [175].

Avg. Speed	Max. Speed	Idle Time	Cruise Time	Ac-cel. Time	De-cel. Time	Avg. Accel.	Avg. De-cel.	Road Grade (%)		
(km/h)	(km/h)	(%)	(%)	(%)	(%)	(m/s²)	(m/s²)	Mean	Max.	Min.
48 to 96	88 to 115	0.7 to 3	75 to 88	7 to 10	7 to 10	0.2 to 0.7	-0.5 to -0.2	-0.3 to 0.3	5 to 7	-7 to -5
Idling: $v \leq 1 \text{ m/s} \cap -0.15 \leq a \leq 0.15 \text{ m/s}^2$; Acceleration: $a > 0.15 \text{ m/s}^2$; Deceleration: $a < -0.15 \text{ m/s}^2$; Cruise: $v \geq 1 \text{ m/s} \cap -0.15 \leq a \leq 0.15 \text{ m/s}^2$.										

Conventional Class 8 tractor-trailer trucks typically carry 700-1,100 L of diesel fuel on-board, enabling a driving range of almost 1,000-1,800 km per refueling, depending on vehicle efficiency and operating conditions [67]. Achieving a similar range with CNG-fueled ICEs is challenging due to the significantly lower volumetric energy density of CNG (9-13 MJ/L at 350-700 bar) [78] compared to diesel fuel. Consequently, much larger storage volumes are required to provide equivalent on-board energy, which limits practical fuel capacity and driving range. In contrast, LNG offers a higher volumetric energy density (21-24 MJ/L at 3-10 bar) [16], making it more suitable for HD applications and enabling driving ranges closer to those of diesel trucks. However, LNG storage and fuel supply systems introduce an additional weight penalty due to the use of cryogenic tanks, insulation, and associated components.

Employing advanced low-carbon-fueled ICEs, such as commercially available diesel-pilot HPDI NG-CI engines, offers diesel-like brake thermal efficiency (BTE) and helps to mitigate the fuel consumption penalty typically associated with NG engines. By combining high efficiency with the use of lower-carbon fuels, these engines can achieve reductions in CO₂ emissions while maintaining comparable performance and driving range to conventional diesel trucks.

2.2 Baseline Diesel Truck Model Development and Validation

The conventional baseline vehicle model in this study is developed and validated using experimental data collected from tested Class 8 Peterbilt 579 diesel trucks over real-

world highway drive cycles. The tested trucks were equipped with a 2019 Cummins X15 (Efficiency Series) diesel engine, compliant with the U.S. EPA Phase-2 GHG emissions regulations [176]. The experimental testing campaign was conducted in 2021 in collaboration between the University of Alberta and industrial partners [177]. The baseline diesel model is built in the Autonomie[®] vehicle simulation software using the same design specifications as the tested vehicles and the validation results are presented over the Highway-2 road between the cities of Calgary and Edmonton, Alberta, Canada. The specifications of the tested Class 8 Peterbilt trucks, including both tractor and trailer, are summarized in Table 2.2.

Table 2.2: Specifications of the Peterbilt 579 Class 8 conventional diesel truck.

Component	Specification	Component	Specification
Engine and Aftertreatment System		Transmission System	
Model (Volume)	Cummins X15 (14.8 L)	Model	Eaton 10-speed Manual
Rated power	336 kW	Gear ratios	0.8, 1.08, 1.44, 1.92, 2.56, 3.42, 4.57, 6.1, 8.23, 12.6
Number of cylinders	inline 6-cylinder	Chassis	
Max. torque	2,332.2 N.m	Axle type (Configuration)	5 tandem axles (4 × 2)
Max. speed	1,912 rpm	Tractor mass	9,340 kg
Max. brake mean effective pressure (BMEP)	20 bar	Empty trailer mass	6,500 kg
Compression ratio (CR)	18.5:1	Max. cargo capacity	23,660 kg
Technologies	VGT/C ¹ , XPI ² , and EGR ³	Fully loaded vehicle mass	39,500 kg
Aftertreatment	Single-module: DOC ⁴ + DPF ⁵ + SCR ⁶	Additional Parameters	
Dry weight	1,430 kg	Wheel radius	0.5 m
Fuel System		Frontal area	10 m ²
Fuel type	Diesel	Final drive ratio	2.85
Fuel tank capacity	800 L		
Weight of fuel + tank	768 kg		

¹ Variable geometry turbocharger ² Extreme pressure injection fuel system ³ Exhaust gas recirculation system
⁴ Diesel oxidation catalyst ⁵ Diesel particulate filter ⁶ Selective catalyst reduction

2.2.1 On-Road Testing and Data Collection

Over 26,000 km of on-road truck testing was performed in January 2021 across multiple highways in Alberta, including: Highway-2 (Queen Elizabeth II) between Calgary and Edmonton; Highway-1 (Trans-Canada Highway) between Calgary and Canmore; and flat sections of Highway-16 located in the northeastern region of Edmonton. During these tests, data from the engine control unit (ECU) and a diesel fuel flow meter were collected. Further details regarding the performed on-road testing are provided in [177].

The tested segment of Highway-2 route, with a total length of 265 km, serves as a representative regional-haul drive cycle that meets the key road characteristic factors specified in the NREL large-scale dataset (Table 2.1) for HD trucks. The distance traveled by the tested vehicles was determined using vehicle speed data over time

obtained from the ECU. The Highway-2 grade–distance profile was derived from geometric roadway data provided by Alberta transportation [178], a department within the Government of Alberta, Canada. By correlating vehicle speed with road grade over distance, the Highway-2 drive cycle profile was generated, as shown in Figure 2.1(b). This drive cycle features a maximum vehicle speed of 110.8 km/h, an average speed of 92.2 km/h, and road grades ranging from -5.7% to 5.3%. Based on standard thresholds used for classifying driving modes in HD drive cycles [179, 180], Highway-2 includes 83% cruising, 1% idling, 8% acceleration, and 8% deceleration. Two repetitions of the Highway-2 cycle result in a total driving distance of 530 km, which falls within the standard DDR of 400–600 km for Class 8 regional-haul trucks [104].

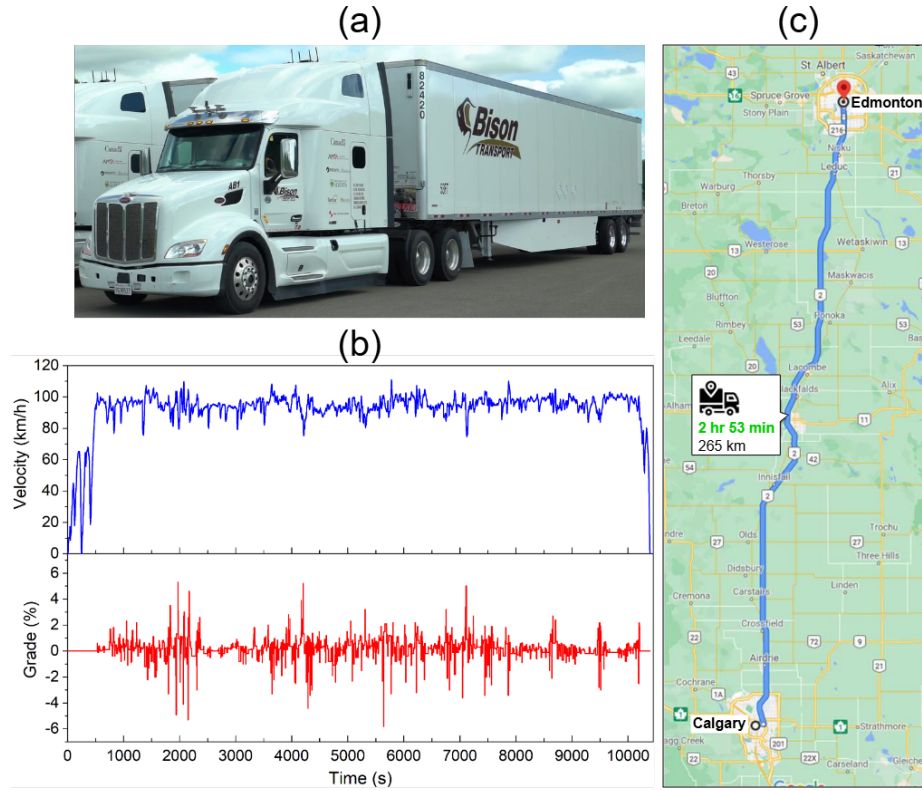


Figure 2.1: Baseline diesel truck testing over the Highway-2 drive cycle from Calgary to Edmonton in Alberta, Canada: (a) actual image of the tested Peterbilt 579 truck, (b) vehicle velocity and road grade versus time, and (c) geographical map.

The brake specific fuel consumption (BSFC) map of the Cummins diesel engine on the tested trucks is shown in Figure 2.2. This map was generated using engine speed and torque data collected from the ECU, along with injector fuel mass flow rate measurements from the fuel flow meter at the corresponding operating points. The maximum BTE of 44.6% corresponds to a minimum BSFC of 189.3 g/kWh,

occurring at an engine torque of 1780 N.m and speed of 1230 rpm. To mitigate noise and uncertainty in the fuel rate measurements across varying engine speeds and torques, data collected over flat segments of Highway-16 under steady vehicle speed conditions were used. Hence, transient engine effects were minimized by eliminating the influence of fluctuating vehicle speed and road grade. In addition, the AIC-7004 diesel flow meter measured both fuel supply and return flows [181], enabling highly accurate and time-resolved fuel consumption measurements.

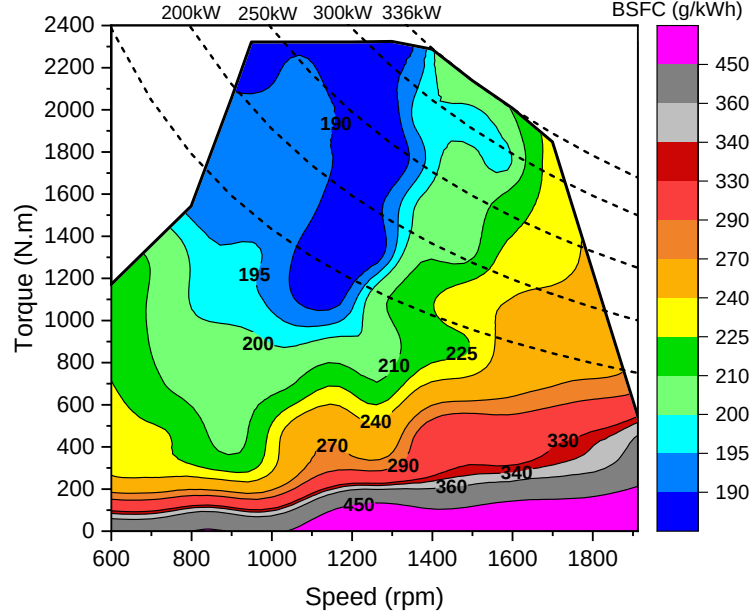


Figure 2.2: Brake specific fuel consumption (BSFC) map of the 336 kW Cummins X15 diesel engine in the tested Peterbilt 579 Class 8 truck in this study.

From an aerodynamic perspective, the vehicle incorporates a package of enhancements, including side skirts, a roof deflector, a tractor-trailer gap reducer, trailer fairings, and rear vanes. These features ensure compliance with recognized vehicle-side technologies designed for Class 8 HD trucks from model years 2021-22, optimizing aerodynamic performance and fuel efficiency [1].

2.2.2 Vehicle Simulation Modeling and Performance Validation

A generic Class 8 conventional tractor-trailer HD truck model with a five-tandem-axle configuration is selected from the Autonomie vehicle simulation software. This baseline model incorporates high-fidelity component submodels representing real-world driveline systems, which have been previously validated against commercially available Class 8 truck components. Building upon this framework, the design parameters

of individual components are tuned to match the specifications of the tested Class 8 trucks, as previously summarized in Table 2.2. The powertrain configuration of the conventional truck model developed in the Autonomie is shown in Figure 2.3.

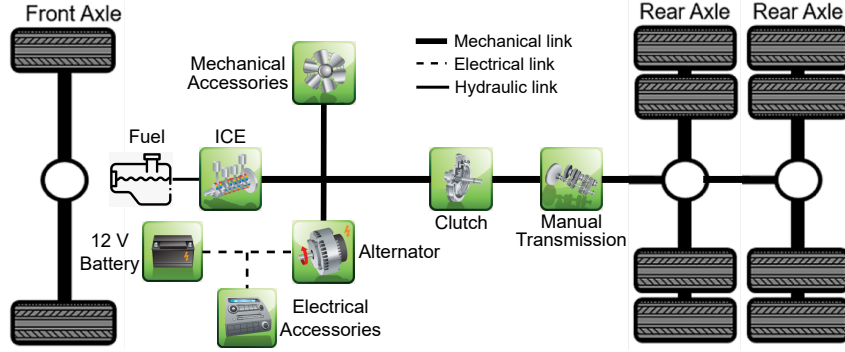


Figure 2.3: Powertrain configuration of the designed conventional Class 8 regional-haul truck.

The generated BSFC map of the Cummins X15 diesel engine is incorporated into the CI engine model. The instantaneous fuel mass flow rate (diesel-equivalent), expressed as a function of engine speed and torque, is calculated as follows:

$$\dot{m}_{f,de} = \frac{T_e \omega_e \text{BSFC}_{e,de}}{3.6 \times 10^6} \quad (2.1)$$

where T_e and ω_e denote the engine torque and angular speed, respectively. $\text{BSFC}_{e,de}$ represents the diesel-equivalent BSFC of the engine at the corresponding operating condition.

The clutch, located between the engine and the manual transmission, is modeled using a binary-state variable representing engaged and disengaged conditions. A generic clutch model for HD truck applications available in the Autonomie library is employed. When the clutch is unlocked ($u_c = 0$), the engine is decoupled from the drivetrain, no torque is transmitted, and the driveline speed is independent of the engine speed. This behavior is described as follows:

$$T_c = u_c \eta_c \left(T_e - \frac{P_{aux}}{\omega_e} \right), \quad \omega_c = u_c \omega_e + (1 - u_c) \omega_{g,in} \quad (2.2)$$

where T_c and η_c denote the clutch output torque and mechanical efficiency, respectively. P_{aux} is the auxiliary power demand of the mechanical and electrical accessories, which is extracted from the engine torque prior to reaching the clutch. ω_c represents the clutch output angular speed, and $\omega_{g,in}$ denotes the transmission input shaft angular speed. The binary variable $u_c \in \{0, 1\}$ indicates the clutch operating state.

An available model of the manual 10-speed Eaton Fuller RTLO-16910B transmission [182] from the Autonomie library is employed, consistent with the transmission installed on the tested Peterbilt truck. The transmission scales the driveline torque and speed according to the selected gear ratio, as previously detailed in Table 2.2, where gear shifting is governed by a predefined shift schedule. This behavior is described as follows:

$$T_{g,\text{out}} = \eta_g n_g T_c, \quad \omega_{g,\text{out}} = \frac{\omega_c}{n_g} \quad (2.3)$$

where $T_{g,\text{out}}$ and $\omega_{g,\text{out}}$ denote the transmission output torque and angular speed, respectively. η_g and n_g represent the gearbox mechanical efficiency and the selected gear ratio, respectively.

A generic final drive model for HD truck applications is selected and implemented using two identical units corresponding to the tandem rear axles. Each final drive is represented using a simplified formulation by neglecting its rotational inertia. The transmission output torque is assumed to be equally distributed between the two axles, with identical gear ratios and efficiency characteristics. The final drives scale the transmitted torque and speed as follows:

$$T_{\text{fd}} = T_{g,\text{out}} n_{\text{fd}} \eta_{\text{fd}}, \quad \omega_{\text{fd}} = \frac{\omega_{g,\text{out}}}{n_{\text{fd}}} \quad (2.4)$$

where T_{fd} and ω_{fd} represent the final drive output torque and angular speed, respectively. η_{fd} and n_{fd} correspond to the final drive mechanical efficiency and the final drive gear ratio, respectively.

The energy balance equation of the Class 8 conventional truck model can be expressed as below:

$$\dot{m}_{\text{f,de}}(\omega_e, T_e) \text{LHV}_{\text{diesel}} \text{BTE}_e \eta_c \eta_g \eta_{\text{fd}} \eta_w = [E_{\text{tr}} + E_{\text{aux}}] - E_{\text{w-e}} \eta_w \eta_{\text{fd}} \eta_g \eta_c \quad (2.5)$$

where the term on the left side represents the energy needed for propulsion from the engine to the driving wheels. $\text{LHV}_{\text{diesel}}$ and BTE_e denote the lower heating value of the fuel (42.6 MJ/kg for diesel) and the brake thermal efficiency of the engine, respectively. η_w corresponds to the efficiency of driving wheels during propulsion. On the right side, E_{tr} refers to the required traction energy of the drive cycle, accounting for the resistance forces and acceleration. E_{aux} denotes the energy demand of the auxiliary systems. The last term represents the motoring operation of the ICE, where the external torque from deceleration contributes to energy recovery, thereby reducing fuel consumption.

The traction energy demand at the driving wheels over the drive cycle is calculated as follows:

$$E_{\text{tr}} = \sum_{i=1}^N \left[\frac{1}{2} \rho_a C_d A_f v_i^3 + (f_{r1} + f_{r2} v_i) M_v g v_i \cos \theta_i + M_v g v_i \sin \theta_i + M_v v_i \frac{v_i - v_{i-1}}{\Delta t} \right] \Delta t \quad (2.6)$$

where ρ_a and C_d denote air density and aerodynamic drag coefficient, respectively. A_f and v_i correspond to the vehicle's frontal area and speed. M_v , θ , and f_r represent the vehicle's total mass, road grade percentage, and rolling resistance coefficient, respectively. Δt indicates the time interval between consecutive points in the drive cycle, which is set to 0.1 s.

The total auxiliary load in the baseline diesel truck model accounts for energy consumed by both mechanical and electrical accessories, as well as alternator losses over the trip. The mechanical and electrical accessory powers are assumed constant at 0.3 kW and 0.4 kW, respectively, based on average auxiliary power demands recorded from the tested Peterbilt truck across the Highway-2 route at normal ambient temperatures (5-25 °C) [177]. The total energy consumption of the auxiliary systems in the conventional model is calculated as follows:

$$E_{\text{aux}} = \sum_{i=0}^N \left[\frac{P_{\text{elec},i}}{\eta_{\text{alt}}} + P_{\text{mech},i} \right] \quad (2.7)$$

where N denotes the total drive cycle time with a fixed time interval of 0.1 s. P_{elec} and P_{mech} refer to the constant power demands of electrical and mechanical accessories, as listed in Table 2.3. η_{alt} represents the alternator efficiency used to supply power to the electrical accessories. A 12-volt alternator model is implemented, featuring a maximum output of 6 kW, constant efficiency of 87%, and a peak current capacity of 160 amps. These parameters closely match those of the actual alternator installed in the tested truck.

To evaluate the performance of the conventional diesel baseline model, comparisons are made for vehicle speed, engine speed, cumulative fuel consumption, and gear shift pattern between the simulation model and the data collected from the tested Peterbilt truck over the Highway-2 drive cycle. Validation is conducted at 77% of the full cargo load, corresponding to a total vehicle weight of 34,058 kg. Figure 2.4 shows the comparison results, including the root mean square error (RMSE) and the mean absolute percentage error (MAPE). The results demonstrate good agreement between the model and the experimental data, indicating acceptable model accuracy.

Table 2.3: Supplementary design parameters for developing the Class 8 conventional truck model in Autonomie.

Parameters	Symbol	Unit	Value
Rolling resistance coefficient 1	f_{r1}	-	0.006
Rolling resistance coefficient 2	f_{r2}	-	0.00012
Air density @ 20°C	ρ_a	kg/m ³	1.21
Aerodynamic drag coefficient	C_d	-	0.6
Mechanical accessories power	P_{mech}	kW	0.3
Electrical accessories power	P_{elec}	kW	0.4

In particular, the model accurately replicates the cumulative fuel consumption of the real vehicle over the 265 km Highway-2 drive cycle, with an RMSE of 1.2 L.

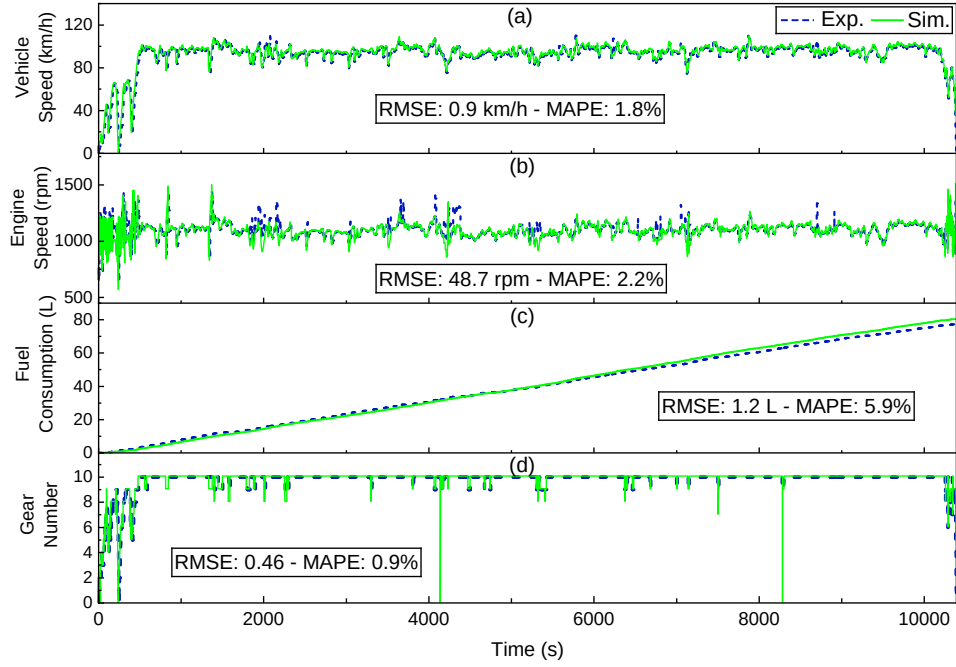


Figure 2.4: Vehicle performance comparison between simulation and experiment for the conventional diesel baseline in this study. (a) Vehicle speed vs. time, (b) Engine speed vs. time, (c) Cumulative fuel consumption vs. time, and (d) Gear number vs. time.

2.3 Conventional Natural Gas Truck Model

To develop a Class 8 conventional truck model powered by low-carbon-fueled CI engines, a 13 L HPDI NG engine model is incorporated into the conventional baseline model, replacing the Cummins diesel engine. Consequently, the on-board storage

tanks for LNG and diesel fuels are sized to match the driving range of the diesel baseline model.

2.3.1 Diesel-Pilot HPDI NG Engine Specifications

The diesel-pilot HPDI NG engine model used in this study was previously developed and validated by the collaborative research team at Simon Fraser University as part of a joint research project. The development of this engine model, including a predictive phenomenological combustion model (GT-SUITE™ DI-Pulse), is detailed in [183]. The engine operates with direct injection of NG as the main fuel, while a small amount of diesel fuel (< 5% of the total fuel energy) is used as a pilot to initiate ignition. The model was calibrated using five representative operating modes and validated across five additional conditions, covering a wide range of the engine map. Validation against experimental data under both steady-state and transient conditions showed accurate predictions of key parameters such as BSFC, torque, airflow, and exhaust gas temperature, with relative errors lower than 5% [43]. The engine incorporates an HPDI dual concentric needle injector fuel system developed by Cespira, a joint venture between Westport Fuel Systems Inc. and Volvo group, established to commercialize the HPDI technology initially created by Westport. Key engine specifications are detailed in Table 2.4.

Table 2.4: Key specifications of the 13 L diesel-pilot HPDI NG engine in this study.

Parameter	Description
Engine configuration	Inline six-cylinder
Power per cylinder (Max. power), kW	56 (336)
Max. brake mean effective pressure (BMEP), bar	23
Max. torque, N.m	2,329.6 @1378 rpm
Max. speed, rpm	1,912
Fuel injection system	Direct injection, dual concentric needle HPDI (300 bar NG injection pressure)
Displacement per cylinder, L	2.1
Compression ratio (CR)	17:1
Bore $\times$ stroke, mm $\times$ mm	131 $\times$ 158
Dry weight, kg	1,259
Aspiration	FGWT/C ¹
Aftertreatment system	Single-module: DOC + DPF + SCR

¹ Fixed-geometry wastegate turbocharger

For vehicle modeling purposes, the diesel-equivalent BSFC of the HPDI NG engine, based on the individual diesel and NG fuel injections across different engine torque and speed operating conditions, is calculated as follows:

$$\text{BSFC}_{\text{e,de}} = \frac{3600}{P_{\text{brake}} (\text{kW})} \times \left[\dot{m}_{\text{f,diesel}} (\text{g/s}) + \left(\dot{m}_{\text{f,NG}} \times \frac{\text{LHV}_{\text{NG}}}{\text{LHV}_{\text{diesel}}} (\text{g/s}) \right) \right] \quad (2.8)$$

where, $\dot{m}_{\text{f,diesel}}$ and $\dot{m}_{\text{f,NG}}$ represent the mass flow rates of diesel and NG fuel, respectively. $\text{LHV}_{\text{diesel}}$ and LHV_{NG} denote the lower heating values of diesel and NG fuels, taken as 42.6 MJ/kg and 48.9 MJ/kg, respectively. P_{brake} also represents the engine brake power.

The diesel-equivalent BSFC map of the 13 L HPDI NG engine is shown in Figure 2.5, and its accuracy was corroborated by the engine experimental data obtained at discrete operating points [183]. At lower loads ($T < 0.5 T_{\text{max}}$), the energy ratio of pilot diesel fuel to main NG fuel reaches up to 5%, but at high loads, this ratio is between 0.5% and 1.5%.

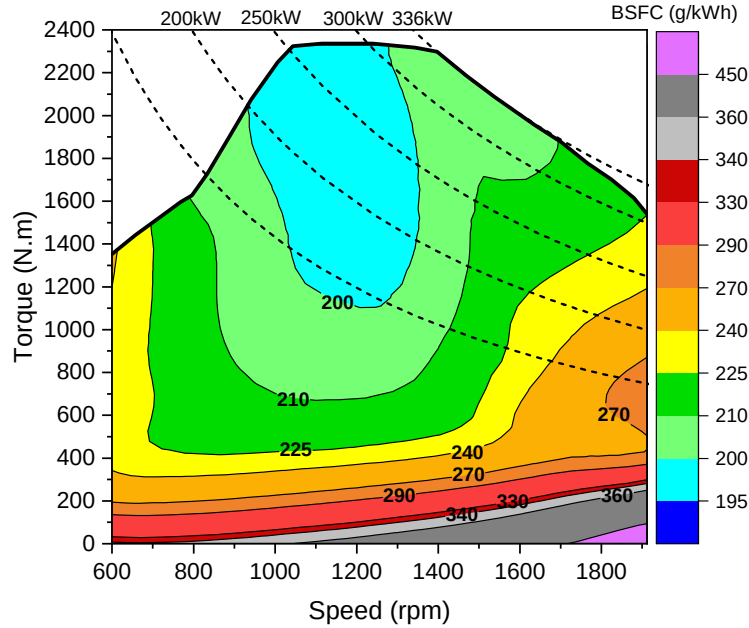


Figure 2.5: Model-generated diesel-equivalent BSFC map of the 336 kW, 13 L HPDI NG engine.

The engine-out CO_2 emissions map of the 13 L HPDI NG engine is derived directly from its diesel equivalent BSFC map. For this conversion, the carbon intensity of NG is assumed to be 50.2 g CO_2/MJ (equivalent to 52.9 kg CO_2 per million BTU) [184], meaning that 1 kg of NG yields approximately 2.5 kg of CO_2 . The resulting brake specific CO_2 emission map of the 13 L HPDI NG engine is presented in Figure 2.6(a).

In NG engines, CH_4 emissions must be considered when assessing GHG emissions. The brake specific CH_4 emissions map of the 13 L HPDI NG engine is shown in Figure 2.6(b). This map was obtained using a validated random forest machine learning

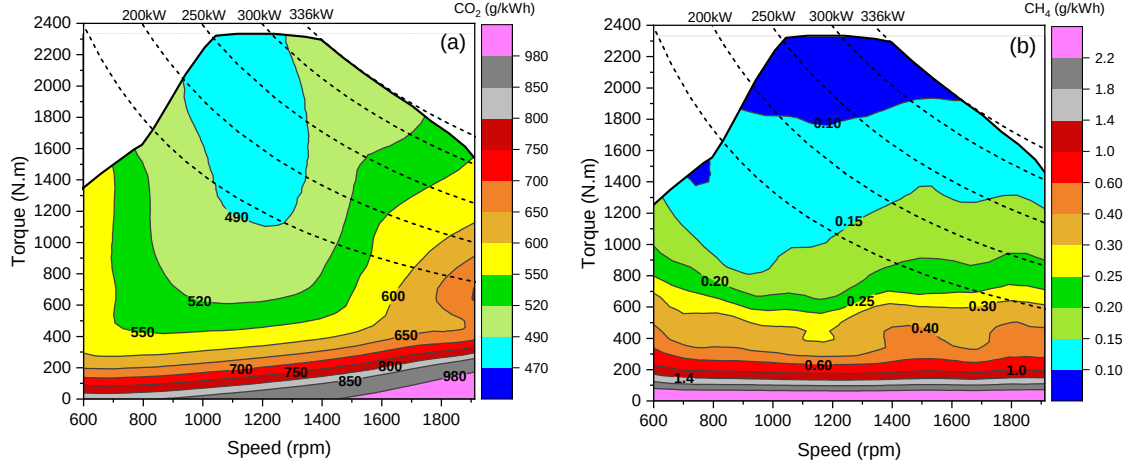


Figure 2.6: Engine-out brake specific: (a) CO_2 , and (b) CH_4 emissions maps of the 13 L HPDI NG engine.

model developed by the collaborative research team at Simon Fraser University. Further details on the modeling and validation process can be found in [183, 185].

2.3.2 Vehicle Simulation Modeling

Except for the engine and the on-board NG and diesel fuel storage tanks, which are modified for the conventional NG configuration, all other powertrain component subsystems remain identical to those of the diesel baseline model. The maximum torque curve of the 13 L HPDI NG engine is aligned with that of the Cummins diesel engine, and both engines have the same rated power of 336 kW. However, due to differences in the fuel injection system, technology package, and engine volume, the curb weight of the HPDI NG engine, including its aftertreatment system, is 171 kg lighter than that of the Cummins X15.

In the conventional diesel baseline model, as previously detailed in Table 2.2, the total fuel capacity is 800 L, resulting in a combined weight of 768 kg for the fuel and two 400 L tanks. To design the conventional NG truck model with the HPDI NG engine, the capacities and weights of diesel and NG tanks are estimated to match the driving range of the diesel counterpart. Assuming energy densities of 35.6 MJ/L for diesel and 21.9 MJ/L for LNG, the LNG volume required in the conventional NG truck model to match the diesel baseline's driving range is calculated as 1300.5 L. Since diesel contributes no more than 5% of the total fuel energy in the HPDI NG engine across different loads, the diesel fuel capacity is set to 5% of that in the conventional diesel baseline model, resulting in 40 L. The powertrain penalty weight of the conventional NG model relative to the diesel baseline, due to differences in engine,

fuel, and fuel tank configurations, is summarized in Table 2.5. A weight density of 0.8 kg/L is assumed for double-walled vacuum-insulated LNG tanks, including fuel, piping, and pumps, based on available commercial data [186, 187].

Table 2.5: Weight specifications of the engine, fuel, and fuel tank systems for the Cummins X15 diesel engine and the 13 L HPDI NG engine, resulting difference in fully loaded vehicle mass between the conventional diesel and NG truck models in this study.

Class 8 Conventional Truck Model	Engine Dry Weight	Fuel Tank Capacity	Total Weight of Fuel and Tanks	Fully Loaded Vehicle Mass
Conv. Diesel Baseline	1,430 kg	Diesel: 800 L	768 kg	39,500 kg
Conv. HPDI NG	1,259 kg	Diesel: 40 L LNG: 1,300.5 L	Diesel: 53.4 kg LNG: 1,040.4 kg	39,655 kg

Under the constraint of maintaining the same maximum cargo capacity, the designed conventional NG powertrain is 155 kg heavier than the conventional diesel counterpart. Additionally, to minimize fuel consumption over the Highway 2 reference drive cycle, the final drive ratio is reduced from 2.85 to 2.80 for the conventional NG model.

2.4 Summary of Chapter

This chapter developed Class 8 five-axle conventional tractor-trailer truck models equipped with either a modern 2019 Cummins X15 diesel engine or a diesel-pilot HPDI NG engine, both with a rated power of 336 kW. The powertrain energy balance equations, along with the component-level governing equations, were derived for the conventional truck models. A real-world regional-haul drive cycle was created using vehicle speed and road grade data collected from on-road testing of a Peterbilt diesel truck along the Highway-2 road between Calgary and Edmonton, Alberta, Canada. The conventional diesel baseline model was then validated over the Highway-2 drive cycle against the corresponding experimental data. The conventional NG truck model was developed from the validated diesel baseline by replacing the engine and modifying the on-board fuel and injection systems. The main findings are summarized as follows:

- The real-world Highway-2 drive cycle is identified as a representative and widely applicable regional-haul cycle for performance evaluation of Class 8 HD trucks, as its key characteristics align with the ranges defined by the NREL [175] based

on a large-scale dataset of truck operations. These characteristics include the distribution of operating modes (idling, cruising, acceleration, and deceleration), as well as statistical ranges for vehicle speed (average and maximum) and road grade (average, maximum, and minimum) over the drive cycle.

- The validation results demonstrate acceptable agreement between the performance of the diesel baseline model and the experimental test data over the Highway-2 drive cycle. This is reflected in RMSE values of 0.9 km/h for vehicle speed, 48.7 rpm for engine speed, and 1.2 L for cumulative fuel consumption.
- To achieve the same driving range per refueling under the same cargo load condition, the conventional NG truck model is 155 kg heavier than the diesel baseline counterpart. This weight penalty results from the combined effect of a 326 kg increase in the mass of LNG, diesel fuel, and associated storage tanks, partially offset by a 171 kg reduction in engine dry weight for the HPDI NG engine compared to the Cummins X15 diesel engine.

The developed conventional truck models can be simulated over a range of drive cycles and under varying cargo load conditions to assess fuel consumption and engine-out emissions of Class 8 non-hybrid trucks. In the next chapter, hybrid electric Class 8 truck powertrains are developed by incorporating electrical driveline components and implementing the corresponding modifications to the mechanical driveline of conventional models.

Chapter 3

Design of Class 8 Hybrid Electric Truck Configurations¹

The conventional Class 8 truck configurations, equipped with either a diesel or an HPDI NG engine, were designed and validated in the previous chapter. This chapter aims to derive the fundamental component-level equations for modeling full hybrid electric powertrains, which demonstrate a high potential to reduce fuel consumption and emissions in Class 8 HD truck applications. The energy balance equations are then formulated to represent the power distribution between mechanical and electrical power sources across different hybrid operating modes. The considered hybrid configurations include: 1) a pre-transmission parallel (P_2) hybrid with a single e-motor, 2) a complex power-split hybrid incorporating a planetary gearset coupled to a two-speed automatic gearbox, and 3) a series plug-in hybrid configuration. Finally, control strategies for the energy management system (EMS) are designed to manage hybrid operating modes based on the architecture of each proposed hybrid configuration. These include constrained rule-based strategies for the P_2 and power-split configurations, and a dynamic programming (DP)-based strategy for the plug-in hybrid configuration.

3.1 Introduction and Background

Electric machines are a key component of hybrid electric powertrains. Several e-motor types have been widely employed in hybrid electric vehicles, including induction motors (IM) [188], switched reluctance motors (SRM) [189], and permanent magnet synchronous motors (PMSM) [190]. IMs are cost-effective but exhibit moderate efficiency ($\sim 90\text{-}94\%$) and lower power density ($\sim 1.5\text{-}3$ kW/kg) [188]. SRMs offer high

¹This chapter is partially based on the published journal papers in [2, 4].

reliability and eliminate the need for rare-earth materials, but their efficiency ($\sim 88-93\%$) and torque density ($\sim 10-20$ Nm/kg) [189], along with high torque fluctuations, limit their suitability for automotive applications. In contrast, PMSMs provide superior efficiency ($\sim 94-97\%$), higher power density ($\sim 3-6$ kW/kg), and greater torque density ($\sim 20-40$ Nm/kg) [190], making them particularly well suited for Class 8 HD trucks that require high torque capability and high energy efficiency.

The high-voltage battery is another key component of hybrid electric powertrains. Several types of Li-ion chemistries have been widely used in HD truck applications, including lithium iron phosphate (LFP) [191], lithium titanate oxide (LTO) [52], nickel manganese cobalt (NMC) [192], and solid-state batteries [193]. LFP batteries offer a long life-cycle ($\sim 3,000-6,000$ cycles) and high safety but a lower energy density ($\sim 90-160$ Wh/kg) [194]. LTO batteries provide an ultra-long life-cycle ($\sim 10,000-20,000$ cycles), but their very low energy density ($\sim 60-100$ Wh/kg) leads to a higher system weight [52]. Solid-state batteries show promising energy density ($\sim 300-500$ Wh/kg) and improved safety; however, their high cost and limited commercial maturity currently restrict large-scale deployment [193]. In contrast, NMC batteries provide high energy density ($\sim 150-250$ Wh/kg), along with good power density ($\sim 1-3$ kW/kg) and moderate life-cycle ($\sim 1,000-2,500$ cycles) [192], making them well suited for Class 8 hybrid applications that require a balance between energy capacity, weight, and performance.

Control strategies for EMS in hybrid electric vehicles are classified into rule-based and optimization-based approaches [195]. Rule-based strategies rely on predefined heuristics and are suitable for real-time implementation with low computational complexity, without requiring full drive cycle knowledge, consistent with the assumption of non-connected vehicles [105]. In contrast, optimization-based strategies aim to achieve an optimal energy distribution that requires knowledge of the projected drive cycle [128, 196]. These methods can be further divided into global optimization approaches, such as DP, and real-time optimization approaches, such as the equivalent consumption minimization strategy (ECMS) [197, 198]. In addition, the optimal control trajectories obtained from DP-based optimization can be used as offline benchmarks to identify dominant operating trends and support the design or calibration of practical rule-based control strategies [199].

In this study, rule-based strategies are employed for the P_2 and power-split hybrid configurations due to their lower computational requirements and their effectiveness in charge-sustaining operation. In contrast, a DP-based strategy is adopted for the plug-in hybrid configuration, as the availability of a predefined drive cycle enables global

optimization of energy usage, particularly for managing the battery SOC trajectory under a blended charge-depletion strategy [148].

3.2 Pre-Transmission Parallel Hybrid Electric Truck

In this section, the component-level governing equations for the P_2 hybrid Class 8 truck powertrain model are derived. In addition, the energy balance equation describing power distribution between mechanical and electrical power sources under different hybrid operating modes is presented. Then, a constrained rule-based control strategy for the EMS is formulated to manage operation across these modes. The powertrain configuration of the P_2 hybrid electric design, featuring a single e-motor coupled to an automatic transmission via a torque coupler, is shown in Figure 3.1.

The P_2 hybrid electric configuration offers the most favorable balance among full hybrid architectures in terms of efficiency, system complexity, and cost for Class 8 HD truck applications [129]. The e-motor output shaft is connected to the driveline between the ICE and the transmission system, enabling the ICE and e-motor to contribute independently or together to the torque delivered to the wheels [136]. This allows the transmission to amplify the e-motor's torque, even in all-electric mode, making it an effective approach to meet the high torque demands of HD trucks during acceleration and operation on steep grades. The engine torque is partially decoupled from the wheels, allowing the engine to operate more frequently within its high-efficiency region [131]. In addition, recovering regenerative brake energy into the battery during deceleration increases the net powertrain energy efficiency.

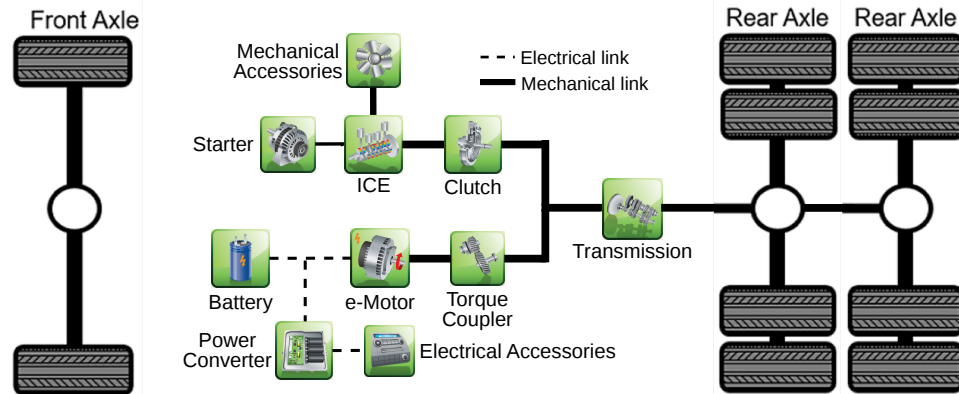


Figure 3.1: Powertrain configuration of the pre-transmission parallel (P_2) hybrid Class 8 regional-haul truck in this study.

Introducing a fixed-ratio torque coupler between the e-motor and the transmission

improves e-motor efficiency by decoupling its operating speed from the main shaft speed imposed by the driving wheels and the engine (Figure 3.1). This coupling adjusts the operating torque-speed range of the e-motor, enabling it to operate closer to its high-efficiency region.

3.2.1 Powertrain Modeling

A generic CI engine model is selected from the Autonomie library for the P₂ hybrid electric configuration to calculate the injected fuel mass flow rate at each simulation time step over the drive cycle. Depending on the type of engine (in terms of fuel and displacement) selected for the hybrid configuration, the corresponding diesel-equivalent BSFC map and engine specifications are incorporated into the model. The engine fuel mass flow rate, expressed as a function of engine torque and speed, is calculated based on Eq. (2.1), as previously derived in Chapter 2 for conventional truck configurations.

A PMSM with a peak motoring and regenerative efficiency of 96% is selected in the Autonomie. The e-motor is modeled based on the torque–speed efficiency map, as shown in Figure 3.2, which is representative of manufacturer-provided data for HD truck applications [200, 201]. Its capability to deliver peak torque within the 0–1200 rpm range and more than 98% of its peak power within the 1100–2000 rpm range aligns well with the typical operating speed range of the CI engines considered in this study. The e-motor power during motoring and regenerative operation is expressed as follows:

$$P_m = \begin{cases} \frac{T_m \omega_m}{\eta_m}, & T_m \geq 0 \\ T_m \omega_m \eta_m, & T_m < 0 \end{cases} \quad (3.1)$$

where T_m and ω_m denote torque and angular speed of the e-motor, respectively. η_m represents the e-motor efficiency at the corresponding speed and torque. When the e-motor operates in drive mode, the torque is positive; conversely, when it operates in generator mode, the torque is negative.

The Li-ion battery is modeled using a quasi-static equivalent-circuit formulation, in which the battery is represented by an open-circuit voltage source in series with an internal resistance. This formulation enables the prediction of battery SOC and current under a given power demand. The battery pack consists of multiple parallel strings, each containing 98 cells in series, with a single-cell energy capacity of 15.8 Wh. This energy capacity is selected based on the specifications of commercially

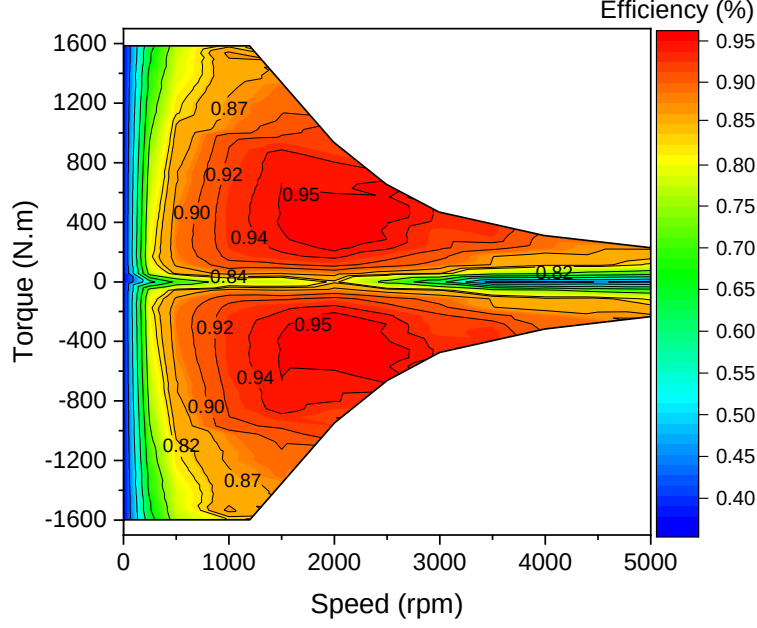


Figure 3.2: Torque–speed efficiency map of the permanent magnet synchronous motor (PMSM) used in the P₂ hybrid electric configuration, showing motoring and regenerative operating regions along with the corresponding efficiency contours [200, 201]. The baseline rated power of the e-motor is 200 kW.

available NMC batteries used in HD truck applications [202, 203]. The battery model is formulated as follows:

$$\begin{cases} \text{SOC} = \text{SOC}_{\text{in}} - \frac{1}{Q_{\text{batt, cell}}} \int \frac{I_{\text{batt}}}{N_{\text{batt, p}}} dt \\ I_{\text{batt}} = \frac{N_{\text{batt, p}}}{2R_{\text{batt, cell}}} \left(V_{\text{oc, cell}} - \sqrt{V_{\text{oc, cell}}^2 - \frac{4R_{\text{batt, cell}}}{N_{\text{batt, s}} N_{\text{batt, p}}} P_{\text{batt}}} \right) \end{cases} \quad (3.2)$$

where $N_{\text{batt, s}}$ and $N_{\text{batt, p}}$ denote the number of cells connected in series and the number of parallel strings, respectively. SOC_{in} represents the initial battery state of charge. $Q_{\text{batt, cell}}$ and I_{batt} correspond to the single-cell energy capacity and the battery current, respectively. $V_{\text{oc, cell}}$, $R_{\text{batt, cell}}$, and P_{batt} denote the single-cell open-circuit voltage, single-cell equivalent internal resistance, and battery power, respectively.

In the P₂ hybrid design, a 6-speed automatic transmission replaces the 10-speed manual one used in the conventional configurations. This reduction in the number of gear ratios is enabled by the e-motor, which can deliver high torque at low vehicle speeds, typically encountered during urban driving, and provide seamless power assistance during acceleration. A 6-speed automatic gearbox model (Eaton Fuller FSO-8406 [204]), specifically designed for HD truck applications, is selected from the

Autonomie library. Considering that the e-motor is coupled to the gearbox input shaft through a fixed-ratio torque coupler (Figure 3.1), the resulting gearbox output torque ($T_{g,out}$) and angular speed ($\omega_{g,out}$) are expressed as follows:

$$T_{g,out} = (T_c + T_m n_{tc}) \eta_g n_g, \quad \omega_{g,out} = \frac{\omega_c}{n_g} \quad (3.3)$$

where T_c and ω_c denote the clutch output torque and angular speed, respectively, consistent with the formulation used for the conventional model in Eq. (2.2). n_{tc} represents the torque coupler fixed gear ratio, with torque transmission losses neglected. n_g and η_g correspond to the selected transmission gear ratio and its mechanical efficiency, respectively.

The energy balance equation of the P₂ hybrid electric configuration, illustrating the component-level energy flows under different hybrid operating modes, is formulated as follows [205]:

$$\begin{aligned} & [\dot{m}_{f,de}(\omega_e, T_e) \text{LHV}_{\text{diesel}} \text{BTE}_e \eta_c \eta_{g,pr} \eta_{fd,pr} \eta_{w,pr}]_{ls1} + [E_b \eta_b \eta_{pc} \eta_{m,pr} \eta_{tc,pr} \eta_{g,pr} \eta_{fd,pr} \eta_{w,pr}]_{ls2} = \\ & [E_{tr} + E_{aux}]_{rs1} + [E_{elec} \eta_b \eta_{pc} \eta_{m,pr} \eta_{tc,pr} \eta_{g,pr} \eta_{fd,pr} \eta_{w,pr}]_{rs2} + [\Delta E_b \eta_b \eta_{pc} \eta_{m,pr} \eta_{tc,pr} \eta_{g,pr} \eta_{fd,pr} \\ & \eta_{w,pr}]_{rs3} - [(E_{brk} - \frac{E_{w,brk}}{\eta_{w,brk}} - \frac{E_{w-e}}{\eta_{w,brk} \eta_{fd,brk} \eta_{g,brk}}) \eta_{w,brk} \eta_{f,brk} \eta_{g,brk} \eta_{tc,brk} \eta_{m,brk} \eta_{pc} \eta_b^2 \eta_{pc} \eta_{m,pr} \\ & \eta_{tc,pr} \eta_{g,pr} \eta_{fd,pr} \eta_{w,pr}]_{rs4} - [E_{w-e} \eta_{w,brk} \eta_{fd,brk} \eta_{g,brk} \eta_c^2 \eta_{g,pr} \eta_{fd,pr} \eta_{w,pr}]_{rs5} + [E_{e-b} \eta_c \eta_{g,pr} \eta_{fd,pr} \\ & \eta_{w,pr} (1 - \eta_{tc,brk} \eta_{m,brk} \eta_{pc} \eta_b^2 \eta_{pc} \eta_{m,pr} \eta_{tc,pr})]_{rs6} \end{aligned} \quad (3.4)$$

where the first term on the left-hand side (ls1) represents the energy required for propulsion from the engine, while ls2 corresponds to the energy needed for propulsion from the battery. On the right-hand side (rs), rs1 and rs2 denote the energies required for traction at the driving wheels and electrical accessories, respectively. rs3 accounts for the battery energy penalty to maintain the same SOC at the beginning and end of the drive cycle. rs4 represents the recaptured energy from regenerative braking, acknowledging that not all braking energy during the drive cycle can be recuperated due to losses in the wheels and the portion consumed by the ICE during motoring mode. rs5 captures the external energy from vehicle deceleration that contributes to energy recovery by the ICE, while rs6 represents the energy consumed by the ICE to charge the battery throughout the drive cycle. $\dot{m}_{f,de}$ denotes the diesel-equivalent fuel mass flow rate of the engine as a function of engine speed (ω_e) and torque (T_e). $\text{LHV}_{\text{diesel}}$ and BTE_e represent the lower heating value of diesel fuel and the brake thermal efficiency of the engine, respectively. E_b and E_{elec} represent the total energy

consumed by the battery and electrical accessories. E_{brk} and $E_{\text{w,brk}}$ denote the total braking energy during the drive cycle and the braking energy on the wheels, respectively. $E_{\text{e-b}}$ indicates the energy consumed by the ICE to charge the battery, and $E_{\text{w-e}}$ denotes the motoring energy from the wheels to the ICE. The terms η_{b} , η_{pc} , η_{m} , η_{tc} , η_{g} , η_{c} , η_{fd} , and η_{w} represent the efficiency of the battery, power converter (PC), e-motor, torque coupler, gearbox, clutch, final drive, and driving wheels, respectively. The subscripts ‘brk’ and ‘pr’ refer to the powertrain braking and propulsion modes, respectively.

The traction energy at the wheels required to overcome the driving resistance forces is calculated using Eq. (2.6), as previously defined in Chapter 2 for the conventional configurations. The total auxiliary energy demand, including both mechanical and electrical accessories, in the P_2 hybrid electric configuration is calculated as follows:

$$E_{\text{aux}} = \sum_{i=0}^N \left[\frac{P_{\text{elec},i}}{\eta_{\text{PC}}} + P_{\text{mech},i} \right] \quad (3.5)$$

where N denotes the total drive cycle time with a fixed time interval of 0.1 s. P_{elec} and P_{mech} refer to the constant power demands of electrical and mechanical accessories. η_{PC} denotes the operational efficiency of the direct current (DC) PC, which operates at a constant output voltage of 12 V to supply electrical accessories from the Li-ion battery. A generic 12 V DC-PC model is selected from the Autonomie library, with a maximum electrical conversion efficiency of 98%, consistent with commercially available buck converters used in hybrid electric vehicles [206, 207].

In the P_2 hybrid electric configuration, all mechanical accessory loads described in Chapter 2 for the conventional configurations are replaced with electrical auxiliary loads. The hybrid electrical accessory load model accounts for electrical energy consumption of the high-voltage battery pack, the e-motor, HVAC systems, and thermal management of electrical components. The total electrical accessory power demand is set at a constant 3 kW [19, 127, 128], consistent with values reported in the literature for Class 8 hybrid regional-haul trucks operating under normal ambient temperatures (5-25 °C).

3.2.2 Control Strategy for Energy Management System

In the P_2 hybrid design, a constrained on/off rule-based EMS [105, 196] is employed to promote engine operation within its high-efficiency region, corresponding to lower BSFC values. When the engine operating points deviate from the efficient region, the e-motor is engaged to supplement or replace the engine torque. The proposed EMS

partitions the engine operating domain into four distinct torque zones by defining low, optimal, and high torque lines (T_{low} , T_{opt} , and T_{high}), each associated with different levels of torque demand. As an example, the locations of these torque lines on the diesel-equivalent BSFC map of the 13 L HPDI H_2 engine are illustrated in Figure 3.3. The optimal torque line at each engine speed is obtained by identifying the torque corresponding to the maximum BTE, or equivalently minimum BSFC, as expressed below:

$$\begin{cases} T_{\text{opt}}(\omega) = \arg \max_{T \in [0, T_{\text{max}}(\omega)]} \text{BTE}_e(\omega, T), \\ \text{BTE}_{\text{max}}(\omega) = \text{BTE}(\omega, T_{\text{opt}}(\omega)) \end{cases} \quad (3.6)$$

where T_{opt} and T_{max} denote the optimal and maximum torques of the engine, respectively. BTE_{max} and ω correspond to the highest brake thermal efficiency and angular speed of the engine, respectively.

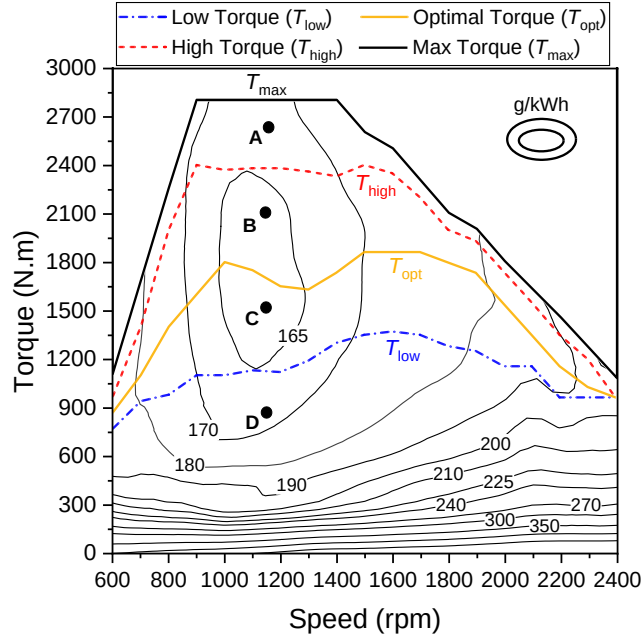


Figure 3.3: Derived torque-speed curves of the constrained on/off rule-based energy management strategy for the P_2 hybrid electric configuration. Points A, B, C, and D illustrate operation in the high, upper-optimal, lower-optimal, and low torque zones, respectively, overlaid on the diesel-equivalent BSFC map of the 13 L HPDI H_2 engine in this study.

To define a practical operating window around the optimal torque line, an efficiency threshold parameter is introduced, such that operating points whose efficiency remains within a specified fraction of the maximum efficiency at each speed are considered near-optimal. The low and high torque lines that satisfy the prescribed efficiency threshold within the engine's feasible torque limits are obtained as follows:

$$\begin{cases} T_{\text{low}}(\omega) = \min \left\{ T \mid \text{BTE}_e(\omega, T) \geq (1 - \delta) \text{BTE}_{\text{max}}(\omega) \right\}, \\ T_{\text{high}}(\omega) = \max \left\{ T \mid \text{BTE}_e(\omega, T) \geq (1 - \delta) \text{BTE}_{\text{max}}(\omega) \right\}, \\ \text{s.t.} \quad T_{\text{low}}(\omega) \leq T_{\text{opt}}(\omega) \leq T_{\text{high}}(\omega) \leq T_{\text{max}}(\omega) \end{cases} \quad (3.7)$$

where T_{low} and T_{high} denote the satisfied low and high torque boundaries, respectively, that meet the prescribed efficiency criterion at each engine speed. T_{max} represents the maximum engine torque as a function of speed, and δ denotes the relative engine efficiency threshold to identify the low and high torque lines.

The EMS in the P₂ hybrid design divides the battery SOC into three levels using low and high SOC thresholds (SOC_{L} and SOC_{H}). These three levels are classified as high ($\text{SOC} \geq \text{SOC}_{\text{H}}$), medium ($\text{SOC}_{\text{L}} < \text{SOC} < \text{SOC}_{\text{H}}$), and low ($\text{SOC} \leq \text{SOC}_{\text{L}}$). The powertrain requested torque may fall within any of the four predefined engine torque zones illustrated in Figure 3.3, including: the low-torque zone ($T_{\text{req}} < T_{\text{low}}$), the lower-optimal torque zone ($T_{\text{low}} \leq T_{\text{req}} < T_{\text{opt}}$), the upper-optimal torque zone ($T_{\text{opt}} \leq T_{\text{req}} < T_{\text{high}}$), and the high-torque zone ($T_{\text{high}} \leq T_{\text{req}} \leq T_{\text{max}}$). Based on the specified battery SOC levels and engine torque zones, the corresponding control rules are defined and summarized in Table 3.1. T_e and T_{chg} correspond to the engine's torque demand and the battery charging torque, respectively. The low and high SOC thresholds, SOC_{L} and SOC_{H} , are tuned to regulate the battery charge in 60% at both the start and end of the trip, while avoiding excessive charging and deep discharging by constraining SOC within the 20-90% range. Additional details about the constrained on/off EMS are found in [105, 196].

Table 3.1: Constrained on/off energy management strategy rules for the pre-transmission parallel (P₂) hybrid electric configuration in this study.

Requested Torque	SOC Level		
	Low $\text{SOC} \leq \text{SOC}_{\text{L}}$	Medium $\text{SOC}_{\text{L}} < \text{SOC} < \text{SOC}_{\text{H}}$	High $\text{SOC} \geq \text{SOC}_{\text{H}}$
Low torque zone (e.g., point D) ($T_{\text{req}} < T_{\text{low}}$)	$T_e = T_{\text{opt}}$ $T_{\text{chg}} = T_{\text{opt}} - T_{\text{D}}$	$T_e = T_{\text{low}}$ $T_{\text{chg}} = T_{\text{low}} - T_{\text{D}}$	$T_e = 0$ $T_{\text{chg}} = 0$
Lower-optimal torque zone (e.g., point C) ($T_{\text{low}} \leq T_{\text{req}} < T_{\text{opt}}$)	$T_e = T_{\text{high}}$ $T_{\text{chg}} = T_{\text{high}} - T_{\text{C}}$	$T_e = T_{\text{opt}}$ $T_{\text{chg}} = T_{\text{opt}} - T_{\text{C}}$	$T_e = T_{\text{C}}$ $T_{\text{chg}} = 0$
Upper-optimal torque zone (e.g., point B) ($T_{\text{opt}} \leq T_{\text{req}} < T_{\text{high}}$)	$T_e = T_{\text{high}}$ $T_{\text{chg}} = T_{\text{high}} - T_{\text{B}}$	$T_e = T_{\text{B}}$ $T_{\text{chg}} = 0$	$T_e = T_{\text{opt}}$ $T_{\text{chg}} = 0$
High torque zone (e.g., point A) ($T_{\text{high}} \leq T_{\text{req}} \leq T_{\text{max}}$)	$T_e = T_{\text{max}}$ $T_{\text{chg}} = T_{\text{max}} - T_{\text{A}}$	$T_e = T_{\text{A}}$ $T_{\text{chg}} = 0$	$T_e = T_{\text{high}}$ $T_{\text{chg}} = 0$

* Points A, B, C, and D represent engine operations within the high, upper-optimal, lower-optimal, and low torque zones, respectively, as shown in Figure 3.3.

3.3 Complex Power-Split Hybrid Electric Truck

In this section, the component-level governing equations and the powertrain energy balance equation for the complex power-split hybrid Class 8 truck model are derived. Then, a constrained rule-based control strategy for the EMS is formulated to manage powertrain operation across different hybrid modes. The powertrain configuration of the complex power-split hybrid electric design, featuring a planetary gearset coupled to an automatic transmission, is shown in Figure 3.4.

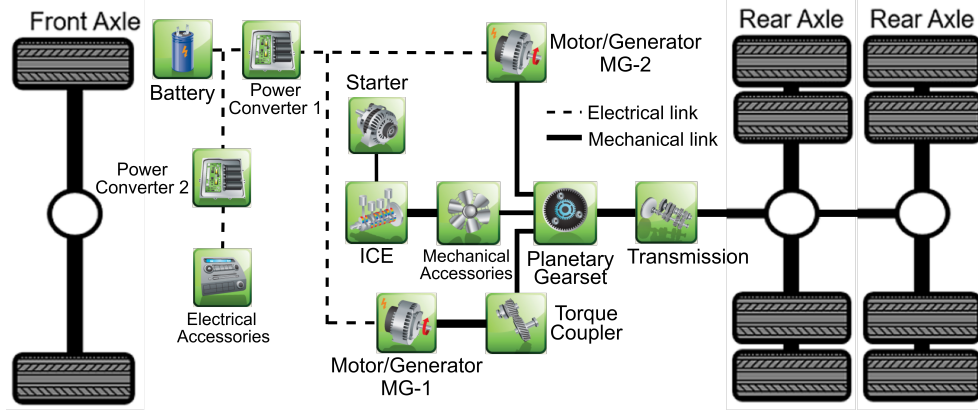


Figure 3.4: Powertrain configuration of the complex power-split hybrid Class 8 regional-haul truck in this study.

By integrating two motor/generators (MG1 and MG2) and a planetary gearset (Figure 3.4), the power-split hybrid configurations offers greater operational flexibility than the P_2 hybrid configuration, combining series, parallel, and series-parallel characteristics [105, 139]. Coupling the planetary gearset with an automatic transmission further enhances this flexibility by providing multiple discrete gear ratios, which is particularly beneficial for Class 8 HD truck applications with wide torque and speed requirements [138, 142]. The use of an automatic transmission allows the engine and MGs to operate within their high-efficiency regions across varying driving conditions, while also improving drivability and load-handling capability.

3.3.1 Powertrain Modeling

In the power-split hybrid electric configuration, the diesel-equivalent fuel mass flow rate of the CI engine is calculated using the same approach as in the P_2 hybrid configuration. The MG1 is modeled similarly to the single e-motor used in the P_2 hybrid configuration, with the same efficiency map as previously shown in Figure 3.2. For the MG2, a PMSM with peak motoring and regenerative efficiencies of 96% [206,

208] and an extended speed operating range is selected from the Autonomie library, as shown in Figure 3.5. The extended speed range and the capability to maintain 90% of peak power over 1200–6000 rpm make the MG2 well suited for coupling with the engine to support operation within its high-efficiency speed range across different driving scenarios.

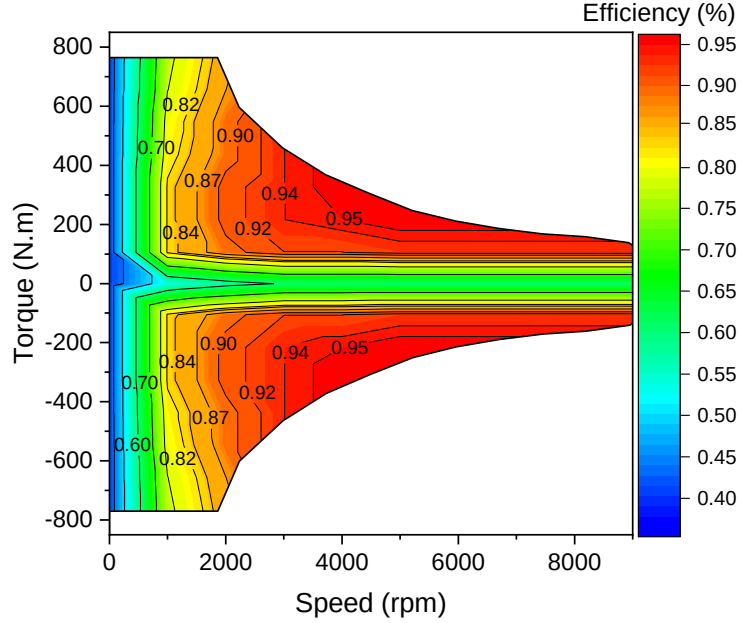


Figure 3.5: Torque–speed efficiency map of the permanent magnet synchronous motor (PMSM) used as MG2 in the power-split hybrid electric configuration, showing motoring and regenerative operating regions along with the corresponding efficiency contours [206, 208]. The baseline rated power of the e-motor is 150 kW.

The Li-ion battery model used in the power-split hybrid configuration is identical to that employed in the P_2 hybrid configuration, with the same design specifications. A generic bidirectional PC model is selected from the Autonomie library to regulate the power flow between the high-voltage battery and the two MGs in the power-split hybrid configuration. The converter operates over a voltage range of up to 500 V [207] and is rated to match the maximum power among MG1 and MG2. In addition, the model used to estimate the total auxiliary power demand, including both mechanical and electrical accessories, is identical to that defined in Eq. (3.5) for the P_2 hybrid configuration.

In the power-split hybrid configuration, the planetary gearset functions as a speed coupling device that regulates the engine speed within its efficient operating range through coordinated control of the MG2. A generic planetary gearset model is selected from the Autonomie library, consisting of four planet gears, a ring gear, a centrally

located sun gear, and a yoke (carrier) that connects the planet gears and supports their rotation. The engine and MG2 are mechanically connected to the yoke and the sun gear, respectively, while the ring gear serves as the output port, transmitting power to the drive wheels through an automatic transmission [105, 140]. The MG1 is connected downstream of the ring gear via a torque coupler, located upstream of the transmission, to enable torque coupling. The rotational speed and torque of the ring gear (ω_r and T_r) prior to coupling with MG1 are expressed as follows:

$$\omega_r = k_{yr} \omega_y - \frac{k_{yr}}{k_{ys}} \omega_s \quad (3.8)$$

$$T_r = -\frac{1}{k_{yr}} T_y = \frac{k_{ys}}{k_{yr}} T_s \quad (3.9)$$

where ω_y and ω_s represent the rotational speeds of the yoke and the sun gear, respectively. T_y and T_s denote the operating torques of the yoke and the sun gear, respectively. k_{yr} and k_{ys} are expressed in terms of the ring-to-sun gear teeth ratio ($k = \frac{Z_r}{Z_s}$) as $k_{yr} = \frac{1+k}{k}$ and $k_{ys} = 1 + k$.

The employed planetary gearset model incorporates two locking mechanisms. Lock 1 is used to lock or release the sun gear and the MG2 shaft to or from the stationary frame of the vehicle, while Lock 2 is used to lock or release the yoke to or from the stationary frame. Based on the states of these locks, different operating modes are achieved during the speed coupling process prior to coupling with MG1, as summarized in Table 3.2. In this table, ω_e and T_e denote the engine rotational speed and torque, respectively, while ω_{MG2} and T_{MG2} represent the rotational speed and torque of the MG2. P_{mech} denotes the mechanical power extracted by the engine-driven accessories prior to coupling with the yoke. η_{yr} and η_{sr} represent the mechanical efficiencies of the PG system between the carrier and the ring gear, and between the sun and the ring gear, respectively.

The torque coupling process in the power-split hybrid configuration is achieved by the MG1, which is connected via a torque coupler to the shaft between the ring gear output and the transmission [142], as previously shown in Figure 3.4. The MG1 provides additional torque to the powertrain during acceleration and peak power demands, and can also replace the engine under low torque conditions, such as during urban driving or at low cargo load conditions. Therefore, the combination of speed coupling and torque coupling provides high flexibility for the engine to operate predominantly within its high-efficiency speed-torque region, corresponding to lower BSFC values [138, 140].

Table 3.2: Summary of the operating modes of the planetary gearset during the speed coupling process in the power-split hybrid electric configuration. The ring gear rotational speed and torque are derived prior to coupling with the MG1.

ID	Operating Modes	Ring Gear Speed	Ring Gear Torque
1	Engine alone traction	$\omega_r = k_{yr} \omega_e$	$T_r = \frac{\eta_{yr}}{k_{yr}} (T_e - P_{\text{mech}}/\omega_e)$
2	MG2 alone traction	$\omega_r = -\frac{k_{yr}}{k_{ys}} \omega_{\text{MG2}}$	$T_r = \frac{k_{ys} \eta_{sr}}{k_{yr}} T_{\text{MG2}}$
3	Engine + MG2 traction	$\omega_r = k_{yr} (\omega_e - \omega_{\text{MG2}}/k_{ys})$	$T_r = \frac{\eta_{yr}}{k_{yr}} (T_e - P_{\text{mech}}/\omega_e)$

In the different operating modes of the planetary gearset during the speed coupling process (Table 3.2), the output torque of the ring gear is consistently lower than the engine torque [105]. Therefore, an auxiliary 2-speed automatic transmission model, selected from the Autonomie library, is implemented downstream of the planetary gearset to amplify the propulsion torque delivered to the driving wheels. The selected transmission model is based on the commercially available Eaton Fuller AT-1202 gearbox [209] for HD truck applications. The gearbox output torque ($T_{g,\text{out}}$) and rotational speed ($\omega_{g,\text{out}}$), accounting for the torque coupling effect of MG1, can be expressed as follows:

$$T_{g,\text{out}} = (T_r + \eta_{tc} n_{tc} T_{\text{MG1}}) \eta_g n_g, \quad \omega_{g,\text{out}} = \frac{\omega_r}{n_g} \quad (3.10)$$

where n_g and n_{tc} represent the selected gear ratio of the transmission and the fixed gear ratio of the torque coupler, respectively. T_{MG1} corresponds to the output torque of the MG1. η_g and η_{tc} denote the mechanical efficiencies of the gearbox and the torque coupler, respectively.

All possible hybrid operating modes achieved by the designed power-split hybrid configuration through the combination of speed coupling and torque coupling are summarized in Table 3.3. The resulting traction energy at the driving wheels, required to overcome driving resistance forces, final drive losses, and braking losses, is calculated using Eq. (2.6), as previously defined in Chapter 2 for the conventional configurations.

3.3.2 Control Strategy for Energy Management System

The developed constrained rule-based EMS in the power-split hybrid configuration incorporates two modes: 1) speed coupling, where engine speed is adjusted by control-

Table 3.3: Summary of hybrid operating modes resulting from the combination of speed coupling and torque coupling in the power-split hybrid electric configuration.

ID	Speed Coupling Mode (Table 3.2)	Torque Coupling Mode	Hybrid Mode Description
1	1	MG1 motoring ($T_{MG1} > 0$)	Parallel full traction mode
2	1	MG1 generating ($T_{MG1} < 0$)	Parallel traction + charging mode
3	2	MG1 motoring ($T_{MG1} > 0$)	All-electric full traction mode
4	2	MG1 generating ($T_{MG1} < 0$)	All-electric traction + charging mode
5	3	MG1 motoring ($T_{MG1} > 0$)	Power-split full traction mode
6	3	MG1 generating ($T_{MG1} < 0$)	Power-split traction + charging mode
7	MG2 alone generating or turned off	MG1 generating ($T_{MG1} < 0$)	Regenerative braking mode

ling the MG2 speed at a given wheel speed, and 2) torque coupling, where the torque transmitted from the planetary gearset's ring gear is combined with the MG1 torque to maintain the engine torque within its high-efficiency zone [140, 143]. These two modes can operate independently or simultaneously, depending on driving conditions and the predefined control rules.

In this study, as part of the rule-based EMS developed for the power-split hybrid design, the vehicle speed range is divided into three regions using the low (V_L) and high (V_H) speed thresholds, as defined in Eq. (3.11) and Eq. (3.12) [105, 139]. These regions are: 1) the low-speed region, where the vehicle speed is equal to or less than V_L , 2) the medium-speed region, where the vehicle speed exceeds V_L but is below V_H , and 3) the high-speed region, where the vehicle speed is equal to or greater than V_H .

$$V_L = \frac{\pi k_{yr} r_w \omega_{e,min}}{30 n_g n_{fd}} \quad (3.11)$$

$$V_H = \frac{\pi k_{yr} r_w \omega_{e,max}}{30 n_g n_{fd}} \quad (3.12)$$

Where r_w represents the wheels radius. $\omega_{e,min}$ and $\omega_{e,max}$ denote the minimum and maximum engine speeds, respectively, ensuring the engine operates within its high-efficiency speed range. n_{fd} corresponds to the final drive gear ratio.

The constrained rules for the speed coupling control mode are outlined in Table 3.4. When the requested vehicle speed falls within the medium-speed range, the MG2 remains off, and the engine speed is directly proportional to the wheel speed. However, when the requested speed is in the high-speed or low-speed region, speed of

the MG2 is adjusted to maintain the engine operation within its high-efficiency speed range. In Table 3.4, P_{MG2} and T_w denote the output power of MG2 and the driving torque at the wheels, respectively.

Table 3.4: Constrained rules for speed coupling mode of the energy management system in the power-split hybrid electric configuration in this study.

Requested Speed	Constraints	Speed Coupling Rules
Low Speed Region ($v \leq V_L$)	$\omega_e = \omega_{e,\min}$ $\omega_{\text{MG2}} > 0$	$\omega_{\text{MG2}} = k_{ys}(\omega_{e,\min} - \frac{30 n_g n_{fd} v}{\pi k_{yr} r_w})$ $P_{\text{MG2}} = \frac{2\pi}{60} T_e \omega_{e,\min} - \frac{n_g n_{fd}}{k_{yr} r_w} T_e v$
Medium Speed Region ($V_L < v < V_H$)	$\omega_{\text{MG2}} = 0$ $T_{\text{MG2}} = 0$	$\omega_e = \frac{\omega_w n_g n_{fd}}{k_{yr}}$ $T_e = \frac{T_w k_{yr}}{n_g n_{fd} \eta_{yr} \eta_g \eta_{fd}}$
High Speed Region ($v \geq V_H$)	$\omega_e = \omega_{e,\max}$ $\omega_{\text{M/G}} < 0$	$\omega_{\text{MG2}} = k_{ys}(\omega_{e,\max} - \frac{30 n_g n_{fd} v}{\pi k_{yr} r_w})$ $P_{\text{MG2}} = \frac{n_g n_{fd}}{k_{yr} r_w} T_e v - \frac{2\pi}{60} T_e \omega_{e,\max}$

The constrained rules of the torque coupling control mode in this study for the power-split hybrid design are outlined in Table 3.5. These control rules determine the engine operating torque lines, consistent with those previously shown in Figure 3.3 for the parallel hybrid design, based on the driver's requested speed (v), requested torque (T_{req}), and the battery SOC. Various hybrid operating modes are achieved by controlling the planetary gearset's electronic lock system, which determines whether the ring, sun, or yoke gears are fixed or free to operate [105, 140]. This flexibility enables the powertrain to switch among series, parallel, series-parallel hybrid modes, as well as the all-electric mode. When the driver's requested speed is in the low-speed region ($v \leq V_L$), the ICE is constrained to operate at the minimum allowable speed ($\omega_{e,\min}$), while the MG2 functions in the generator mode (Table 3.5). For high-speed regions ($v \geq V_H$), the ICE operates at its maximum allowable speed ($\omega_{e,\max}$), and the MG2 switches to propelling mode. In the medium-speed range ($V_L < v < V_H$), the MG2 remains off, and the vehicle operates in parallel mode, as previously detailed in Table 3.1. When the vehicle operates in propulsion mode, the MG1 supplements the torque demand and manages regenerative energy by storing excess energy in the battery, according to the SOC thresholds and the requested torque levels detailed in Table 3.5. The torque contribution of the MG1 under different conditions is calculated using the previously driven Eq. (3.10). In Table 3.5, P_{chg} represents the total battery charging power, while $P_{\text{MG1,chg}}$ denotes the power from the MG1 directed to the battery for charging. ω_{MG1} denotes the rotational speed of the MG1.

Table 3.5: Constrained rules for torque coupling mode of the energy management system in the power-split hybrid electric configuration in this study.

Requested Speed	Requested Torque	SOC Level			
		Low (SOC ≤ SOC _L)	Medium (SOC _L < SOC < SOC _H)	High (SOC ≥ SOC _H)	
$v \leq V_L$ ($\omega_e = \omega_{e,\min}$) ($\omega_{MG2} > 0$)	$T_{\text{req}} > T_{\text{opt}}$ (high torque zone)	$T_e = T_{\text{high}}$	$T_e = T_{\text{opt}}$	$T_e = T_{\text{opt}}$	
		$T_{MG1} > 0, \omega_{MG1} > 0$	$T_{MG1} > 0, \omega_{MG1} > 0$	$T_{MG1} > 0, \omega_{MG1} > 0$	
		$\omega_{MG2} > 0$	$\omega_{MG2} > 0$	$\omega_{MG2} > 0$	
	$T_{\text{req}} \leq T_{\text{opt}}$ (low torque zone)	$P_{\text{chg}} = P_{MG2}$	$P_{MG2} = P_{MG2}$	$P_{\text{chg}} = 0$	
		$T_e = T_{\text{opt}}$	$T_e = T_{\text{req}}$	$T_e = 0$	
		$T_{MG1} < 0, \omega_{MG1} > 0$	$T_{MG1} = 0$	$T_{MG1} = T_{\text{req}}, \omega_{MG1} > 0$	
$V_L < v < V_H$ ($\omega_{MG2} = 0$)	At any of the points A, B, C, or D in Figure 3.3	$\omega_{MG2} > 0$	$\omega_{MG2} > 0$	$\omega_{MG2} = 0$	
		$P_{\text{chg}} = P_{MG2} + P_{MG1,\text{chg}}$	$P_{\text{chg}} = P_{MG2}$	$P_{\text{chg}} = 0$	
		Operating in the parallel mode (Table 3.1)			
	$v \geq V_H$ ($\omega_e = \omega_{e,\max}$) ($\omega_{MG2} < 0$)	$T_{\text{req}} > T_{\text{opt}}$ (high torque zone)	$T_e = T_{\text{high}}$	$T_e = T_{\text{opt}}$	$T_e = T_{\text{opt}}$
			$T_{MG1} < 0, \omega_{MG1} > 0$	$T_{MG1} > 0, \omega_{MG1} > 0$	$T_{MG1} > 0, \omega_{MG1} > 0$
			$\omega_{MG2} < 0$	$\omega_{MG2} < 0$	$\omega_{MG2} < 0$
$T_{\text{req}} \leq T_{\text{opt}}$ (low torque zone)		$P_{\text{chg}} = P_{MG1,\text{chg}}$	$P_{\text{chg}} = 0$	$P_{\text{chg}} = 0$	
		$T_e = T_{\text{opt}}$	$T_e = T_{\text{opt}}$	$T_e = 0$	
		$T_{MG1} < 0, \omega_{MG1} > 0$	$T_{MG1} < 0, \omega_{MG1} > 0$	$T_{MG1} = T_{\text{req}}, \omega_{MG1} > 0$	
		$\omega_{MG2} < 0$	$\omega_{MG2} = 0$	$\omega_{MG2} = 0$	
		$P_{\text{chg}} = P_{MG1,\text{chg}}$	$P_{\text{chg}} = P_{MG1,\text{chg}}$	$P_{\text{chg}} = 0$	

According to the constrained speed coupling and torque coupling control rules, the power-split hybrid configuration can simultaneously propel the vehicle and charge the battery through coordinated engagement of the electrical driveline, which incorporates two MGs. In addition, recuperated regenerative energy does not necessarily need to be stored in the battery; it can be directly transferred through the DC link to the second MG (operating in motoring mode) to assist propulsion. This operational flexibility reduces intermediate energy conversion losses compared to parallel hybrid configurations.

3.4 Series Plug-in Hybrid Electric Truck

In this section, the component-level governing equations and the powertrain energy balance equation for the series plug-in hybrid Class 8 truck model are derived. An optimization-based EMS control strategy is then formulated to manage powertrain operation across different hybrid modes. For this purpose, a DP-based Pontryagin's Minimum Principle (PMP) [152, 210] approach is developed based on a blended-mode battery charge-depletion strategy to minimize net powertrain energy consumption in the plug-in hybrid configuration. The powertrain configuration of the series plug-in hybrid electric design is shown in Figure 3.6.

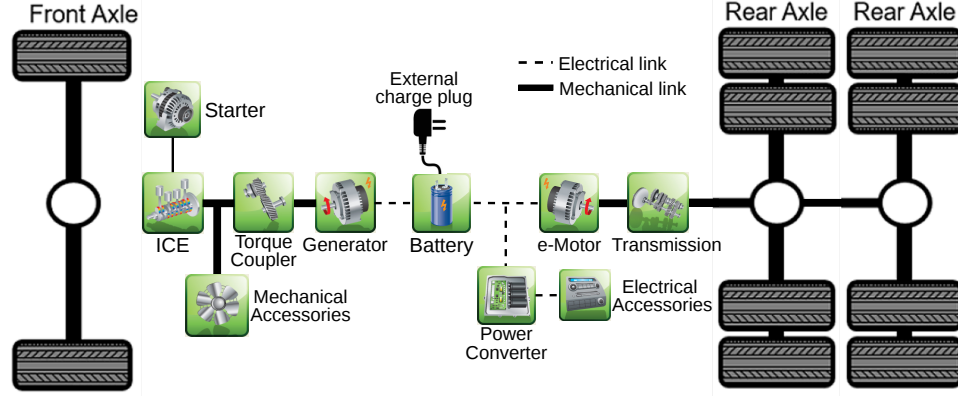


Figure 3.6: Powertrain configuration of the series plug-in hybrid Class 8 regional-haul truck in this study.

3.4.1 Powertrain Modeling

The injected fuel mass flow rate of the engine is determined from the diesel-equivalent BSFC map based on the engine operating point, defined by engine speed and torque, as previously described in Eq. (2.1) in Chapter 2. A PMSM, coupled to the engine via a fixed-ratio torque coupler, is used as the generator to convert the engine's mechanical energy into electrical energy for charging the battery throughout the drive cycle. The generator is modeled based on the torque-speed efficiency map, as shown in Figure 3.7, which is representative of manufacture-provided data for HD truck applications [201, 206]. Its ability to deliver more than 98% of its peak power over a wide speed range of 3000–8000 rpm makes it well suited for series plug-in hybrid configurations, enabling efficient engine operation with stable electrical power generation. The generator power can be expressed as a function of its torque and rotational speed as follows:

$$T_{\text{gen}} = \eta_{\text{tc}} \eta_{\text{gen}} n_{\text{tc}} \left(T_e - \frac{P_{\text{mech}}}{\omega_e} \right), \quad \omega_{\text{gen}} = \frac{\omega_e}{n_{\text{tc}}} \quad (3.13)$$

where T_{gen} is the generator output torque and ω_{gen} is the generator rotational speed. T_e and ω_e are the engine torque and engine speed, respectively. P_{mech} denotes the mechanical accessories power extracted from the engine and n_{tc} corresponds to the fixed gear ratio between the engine and the generator. η_{gen} is the generator efficiency and η_{tc} is the torque coupler efficiency.

The e-motor and electrical accessories models used in the series plug-in hybrid configuration are identical to those previously developed for the P₂ hybrid configuration, as described in Eq. (3.1) and Eq. (3.5), respectively. The governing equations of the

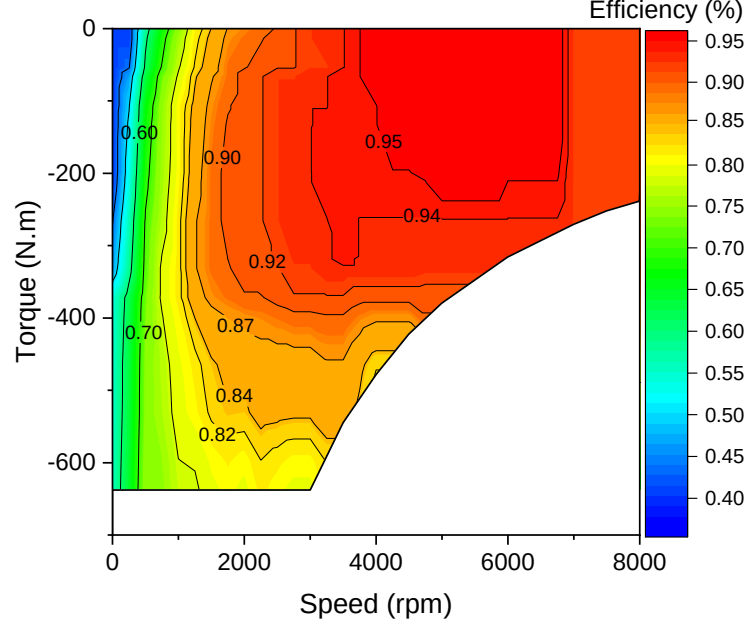


Figure 3.7: Torque–speed efficiency map of the permanent magnet synchronous motor (PMSM) operating in generating mode, used as the generator in the series plug-in hybrid electric configuration [201, 206]. The baseline rated power of the generator is 200 kW.

high-voltage battery are identical to those in Eq. (3.2) for the P_2 hybrid configuration; however, its design parameters are selected based on models of high-energy-density NMC Li-ion battery suitable for plug-in hybrid applications [192, 211]. The battery pack consists of multiple parallel strings, each containing 140 cells in series, with a single-cell energy capacity of 124.1 Wh.

A model of Eaton’s 4-speed automatic transmission, specifically designed for electrified HD truck applications, is selected from the Autonomie library for the series plug-in hybrid configuration and is coupled to the e-motor output shaft. In this hybrid configuration, the engine is mechanically decoupled from the wheels, allowing it to operate near its optimal efficiency region. Consequently, the traction demand is entirely supplied by the e-motor, which inherently provides a wide torque–speed operating range, thus reducing the need for a large number of gear ratios. Therefore, a reduced number of transmission gears is sufficient to maintain the e-motor operation close to its high-efficiency region in both highway and urban driving conditions and across different cargo loads [104, 152]. The gearbox output torque ($T_{g,out}$) and rotational speed ($\omega_{g,out}$) are expressed as follows:

$$T_{g,out} = \eta_g n_g T_m, \quad \omega_{g,out} = \frac{\omega_m}{n_g} \quad (3.14)$$

where η_g denotes the transmissions mechanical efficiency and n_g represents the

selected gear ratio. T_m and ω_m correspond to the e-motor output torque and rotational speed, respectively.

The energy balance equation of the series plug-in hybrid configuration using the blended charge depletion strategy is formulated as follows [212]:

$$[\dot{m}_{f,de}(\omega_e, T_e) \text{LHV}_{\text{diesel}} \text{BTE}_e \eta_{tc} \eta_{gen} \eta_b \eta_{pc} \eta_{m,pr} \eta_{g,pr} \eta_{fd,pr} \eta_{w,pr}]_{ls1} + [E_b \eta_b \eta_{pc} \eta_{m,pr} \eta_{g,pr} \eta_{fd,pr} \eta_{w,pr}]_{ls2} = [E_{tr}]_{rs1} + [E_{aux}]_{rs2} - [(E_{brk} - \frac{E_{w,brk}}{\eta_{w,brk}}) \eta_{w,brk} \eta_{fd,brk} \eta_{g,brk} \eta_{m,brk} \eta_{pc} \eta_b]_{rs3} \quad (3.15)$$

where the first term on the left-hand side (ls1) represents the energy needed for the propulsion from the engine, while ls2 corresponds to the energy needed for propulsion from the battery. On the right-hand side (rs), rs1 and rs2 denote the energies required for traction at the driving wheels and electrical accessories, respectively. rs3 accounts for the recaptured energy from regenerative braking from the wheels to the battery. $\dot{m}_{f,de}$ denotes the diesel-equivalent fuel mass flow rate of the engine as a function of engine speed (ω_e) and torque (T_e). $\text{LHV}_{\text{diesel}}$ and BTE_e represent the lower heating value of diesel fuel and the brake thermal efficiency of the engine, respectively. E_b and E_{aux} represent the total energy consumed by the battery and auxiliary accessories. E_{brk} and $E_{w,brk}$ denote the total braking energy during the drive cycle and the braking energy on the wheels, respectively. The terms η_b , η_{pc} , η_m , η_{tc} , η_g , η_{fd} , and η_w represent the efficiency of the battery, power converter (PC), e-motor, torque coupler, gearbox, final drive, and driving wheels, respectively. The subscripts ‘brk’ and ‘pr’ refer to the powertrain braking and propulsion modes, respectively. The traction energy at the wheels required to overcome the driving resistance forces is calculated using Eq. (2.6), as previously defined in Chapter 2 for the conventional configurations.

3.4.2 Control Strategy for Energy Management System

To improve the performance of the designed series plug-in hybrid configuration over predefined routes, a DP-based optimal control strategy with a blended charge-depletion approach is employed. The control strategy coordinates the engine-generated electrical power and battery usage such that the battery SOC decreases approximately linearly with respect to a known traveled distance [149]. This approach avoids early battery depletion and prevents transitions into charge-sustaining (CS) operation, thereby enabling more efficient utilization of the available electrical energy over the entire drive cycle [104, 148].

To formulate the optimization problem, the Pontryagin's Minimum Principle (PMP) method [210] is implemented in MATLAB to minimize a Hamiltonian cost-function. The optimization considers two objectives: minimizing the engine fuel consumption and depleting the battery SOC to follow a blended charge-depletion trajectory. The Hamiltonian cost-function and the target linear SOC depletion trajectory are formulated as follows:

$$H = \sum_{i=0}^{N-1} \left[100 \times (\text{SOC}_{\text{ref}}(i) - \text{SOC}(i))^2 + (\dot{m}_{\text{f,de}}(\omega_e, T_e) \text{LHV}_{\text{diesel}} \text{BTE}_e \eta_{\text{tc}} \eta_{\text{gen}})^2 \right] \quad (3.16)$$

$$\text{SOC}_{\text{ref}} = \sum_{i=0}^{N-1} \left[\frac{\text{SOC}_{\text{in}} - \text{SOC}_{\text{min}}}{D} X(i) + \text{SOC}_{\text{min}} \right] \quad (3.17)$$

where N denotes the total duration of the drive cycle with a fixed time interval of 0.1 s, $\text{SOC}(i)$ is the instantaneous battery charge level that should follow the reference, and $\dot{m}_{\text{f,de}}$ is the diesel-equivalent fuel mass flow rate of the engine. $\text{LHV}_{\text{diesel}}$ and BTE_e correspond to the lower heating value of diesel and the thermal efficiency of the engine, respectively. η_{tc} and η_{gen} represent the conversion efficiency of the torque coupler and the generator, respectively. Two control inputs are the engine speed (ω_e) and torque (T_e) that must be optimized. SOC_{in} and SOC_{min} denote the initial battery state of charge at the beginning of the trip and the minimum allowable battery state of charge, respectively. D is the total drive-cycle distance before recharging the battery from an external source (e.g., the grid), and X is the cumulative distance traveled at a given time. The values for SOC_{in} and SOC_{min} are assumed as 0.9 and 0.2, respectively.

To minimize the Hamiltonian cost-function defined in Eq. (3.16) over the reference drive cycle, a set of constraints is imposed to ensure that the system operates within feasible limits. These constraints are defined as follows:

$$\begin{aligned} 0 &\leq \omega_e \leq \omega_{e,\text{max}}, \\ 0 &\leq \omega_{\text{gen}} \leq \omega_{\text{gen,max}}, \\ 0 &\leq \omega_m \leq \omega_{m,\text{max}}, \\ 0 &\leq T_e \leq T_{e,\text{max}}, \\ T_{\text{gen,min}} &\leq T_{\text{gen}} \leq T_{\text{gen,max}}, \\ T_{m,\text{min}} &\leq T_m \leq T_{m,\text{max}}, \\ 0.2 &\leq \text{SOC} \leq 0.9. \end{aligned} \quad (3.18)$$

where ω_e , ω_{gen} , and ω_m denote the rotational speeds of the engine, generator, and e-motor, respectively. $\omega_{e,\text{max}}$, $\omega_{\text{gen,max}}$, and $\omega_{m,\text{max}}$ are their corresponding maximum allowable speeds. T_e , T_{gen} , and T_m represent the torques of the engine, generator,

and e-motor, respectively. $T_{e,\max}$, $T_{\text{gen},\min}$, $T_{\text{gen},\max}$, $T_{m,\min}$, and $T_{m,\max}$ denote their respective minimum and maximum allowable torque limits.

To solve the constrained nonlinear optimization problem, the ‘fmincon’ function from MATLAB’s Optimization Toolbox is employed using the sequential quadratic programming (SQP) algorithm. SQP is selected because it is well suited for constrained nonlinear optimization problems with bounded control variables and path constraints [213], as defined in Eq. (3.18). The quadratic penalty terms in the Hamiltonian cost function improve numerical conditioning by penalizing deviations from the target SOC trajectory and high fuel-energy consumption. However, because the full powertrain model includes nonlinear component efficiency maps, SOC dynamics, and torque–speed constraints, the overall optimization problem is not strictly guaranteed to be globally convex. Therefore, the obtained solution is interpreted as a feasible near-optimal control trajectory for the predefined drive cycle, rather than a mathematically guaranteed global optimum.

3.5 Summary of Chapter

This chapter derived the fundamental powertrain equations for modeling full hybrid electric configurations, which demonstrate a high potential to reduce fuel consumption and emissions in Class 8 HD truck applications. The selected hybrid configurations include: 1) a pre-transmission parallel (P_2) hybrid with a single e-motor, 2) a complex power-split hybrid incorporating a planetary gearset coupled to a two-speed automatic gearbox, and 3) a series plug-in hybrid configuration. In addition, control strategies for the energy management system (EMS) are designed to manage feasible hybrid modes for each configuration, aiming to enhance the net powertrain energy efficiency across different operating conditions. Modified rule-based control strategies were designed for the P_2 and power-split hybrid configurations due to their lower computational requirements and effectiveness in charge-sustaining operation. In contrast, a dynamic programming (DP)-based strategy was designed for the plug-in hybrid configuration, as the availability of a predefined drive cycle enables global optimization of energy usage, particularly for managing the battery SOC trajectory under a blended charge-depletion strategy.

The designed hybrid powertrain architectures were refined to better accommodate the high torque and energy demands of Class 8 HD truck applications. These modifications include the addition of fixed-ratio torque couplers between the e-motors and the driveline in the P_2 and plug-in hybrid configurations, enabling more suitable

torque-speed operation and improving e-motor efficiency independent of the driving wheels. In the power-split configuration, a multi-speed transmission is coupled to the planetary gearset to extend the effective driveline gear ratio range and improve torque delivery and efficiency under high-load operating conditions.

The critical variables of the designed hybrid configurations, including those related to the sizing of powertrain components and the EMS, will be optimized in the next chapter as part of the component sizing framework, aiming to minimize net energy consumption over a real-world regional-haul drive cycle.

Chapter 4

Optimal Powertrain Component Sizing and Performance Evaluation¹

The powertrain architectures of the hybrid Class 8 truck configurations considered in this study, along with their control strategies for the energy management system (EMS), were previously modeled in Chapter 3. The present chapter develops a multi-stage component sizing framework to optimize the sizes of the primary powertrain components, as well as to tune the key EMS-related variables for the designed hybridized configurations. This framework aims to minimize net powertrain energy consumption over the real-world Highway-2 regional-haul drive cycle while maintaining propulsion system performance comparable to that of the conventional diesel baseline model. To determine the variation ranges of the optimization variables within this framework, a preliminary component sizing procedure is formulated for each proposed hybrid configuration. Finally, to assess the fuel consumption and emissions of the optimized hybrid electric configurations over a broader range of operating conditions, additional standard highway and urban drive cycles suitable for HD truck applications are introduced.

4.1 Introduction and Background

Optimal sizing of powertrain components and tuning of control parameters are critical in hybrid electric vehicle design, as they directly affect fuel consumption, emissions, and overall system performance [124]. In the literature, dynamic programming (DP) has been widely used as a benchmark optimization method due to its ability to

¹This chapter is partially based on the published journal papers in [2, 4].

determine globally optimal control trajectories over predefined drive cycles[54, 214]. However, its high computational cost and reliance on known driving conditions limit its applicability to large-scale design optimization problems with multiple decision variables [195]. These limitations become more pronounced in HD truck applications, where variations in cargo load and the high dimensionality of the design space increase computational complexity.

To address these limitations, metaheuristic optimization methods such as genetic algorithms (GA) [215] have been widely applied for component sizing and parameter tuning of hybrid powertrains. These methods are well-suited for handling nonlinear, multi-variable optimization problems and have been effectively used to optimize parameters such as engine size and transmission gear ratios [199, 216]. Additionally, design of experiments (DOE) techniques [128, 198, 217] have been employed to systematically explore the design space and identify the influence of key variables on system-level performance, while reducing computational burden compared to exhaustive search approaches.

The multi-stage component sizing framework in this study for the proposed Class 8 hybrid electric truck configurations adopts a combined DOE- and GA-based optimization approach. The DOE method is used to define and explore feasible ranges of key design variables associated with component sizing and EMS-related parameters, while GA is employed to optimize transmission gear ratios. This integrated approach provides a balance between computational efficiency and global search capability, making it suitable for the optimal design of hybrid powertrains in Class 8 HD truck applications.

4.2 Propulsion System Performance Criteria

As part of the optimal powertrain component sizing for a newly designed Class 8 hybrid truck configuration, it is essential to ensure that key propulsion system performance criteria remain comparable to those of conventional counterparts. These performance criteria for conventional Class 8 trucks are specified by the SAE J2807 standard [165] and are summarized in Table 4.1. The criteria include: 1) minimum acceleration times on a flat road under full cargo load (0-48 km/h, 0-96 km/h, and 64-96 km/h), 2) a maximum vehicle speed of 120 km/h on a flat highway under full cargo load, and 3) gradeability performance, defined as the ability to maintain a constant speed of 56 km/h for 20 minutes of continuous driving over a constant 7% road grade. This 7% road grade is derived from the steepest 19.7 km segment of the Davis

Dam (Arizona-68) in the United States.

Table 4.1: Summary of propulsion system performance criteria specified by the SAE J2807 standard [165] for conventional Class 8 HD trucks.

Class 8 Truck Category	Daily Driving Range (km)	0 → 48 km/h Acceleration Time (s)	0 → 96 km/h Acceleration Time (s)	Flat Road Top Speed (km/h)	Gradeability Performance Metric
Long-haul	600–800	20	70	120	Capability to maintain a constant speed of 56 km/h for 20 minutes over a constant 7% road grade
Regional-haul	400–600	18	65	120	

* Level road acceleration and top speed criteria must be satisfied when the truck is fully loaded.

The Class 8 conventional diesel truck model used as the baseline in this study, as previously designed in Chapter 2, satisfies the acceleration and top speed performance criteria under the full cargo load condition, as listed in Table 4.1. The 0–48 km/h acceleration time is 16 s and the 0–96 km/h acceleration time is 62 s. However, the diesel baseline model satisfies the gradeability performance criterion under 34% of its full cargo capacity (full cargo capacity: 23.6 tonne).

Throughout the optimal powertrain component sizing of the proposed hybridized Class 8 truck configurations in this study, the propulsion system performance criteria are required to be maintained at levels comparable to those of the baseline diesel model. In addition, the maximum carryable cargo load of the hybridized configurations must be maintained at the same level as that of the baseline diesel model.

4.3 Multi-Stage Optimization Framework

A multi-stage optimization framework is developed in this study for optimal powertrain component sizing and tuning of EMS-related parameters for the designed Class 8 hybrid electric truck configurations. The framework aims to satisfy two main objectives: 1) ensuring that the propulsion system performance criteria are comparable to or better than those of the conventional diesel baseline model, and 2) minimizing fuel consumption over a real-world regional-haul drive cycle (Highway-2 in this study) under the full cargo load condition. Satisfaction of the first objective is treated as a prerequisite for achieving the second objective. The optimization process employs a full-factorial DOE in combination with GA. The GA is employed to optimize the transmission gear ratios in each iteration after selecting a set of DOE variables for the proposed hybrid electric configurations. The schematic of the multi-stage optimization framework is shown in Figure 4.1.

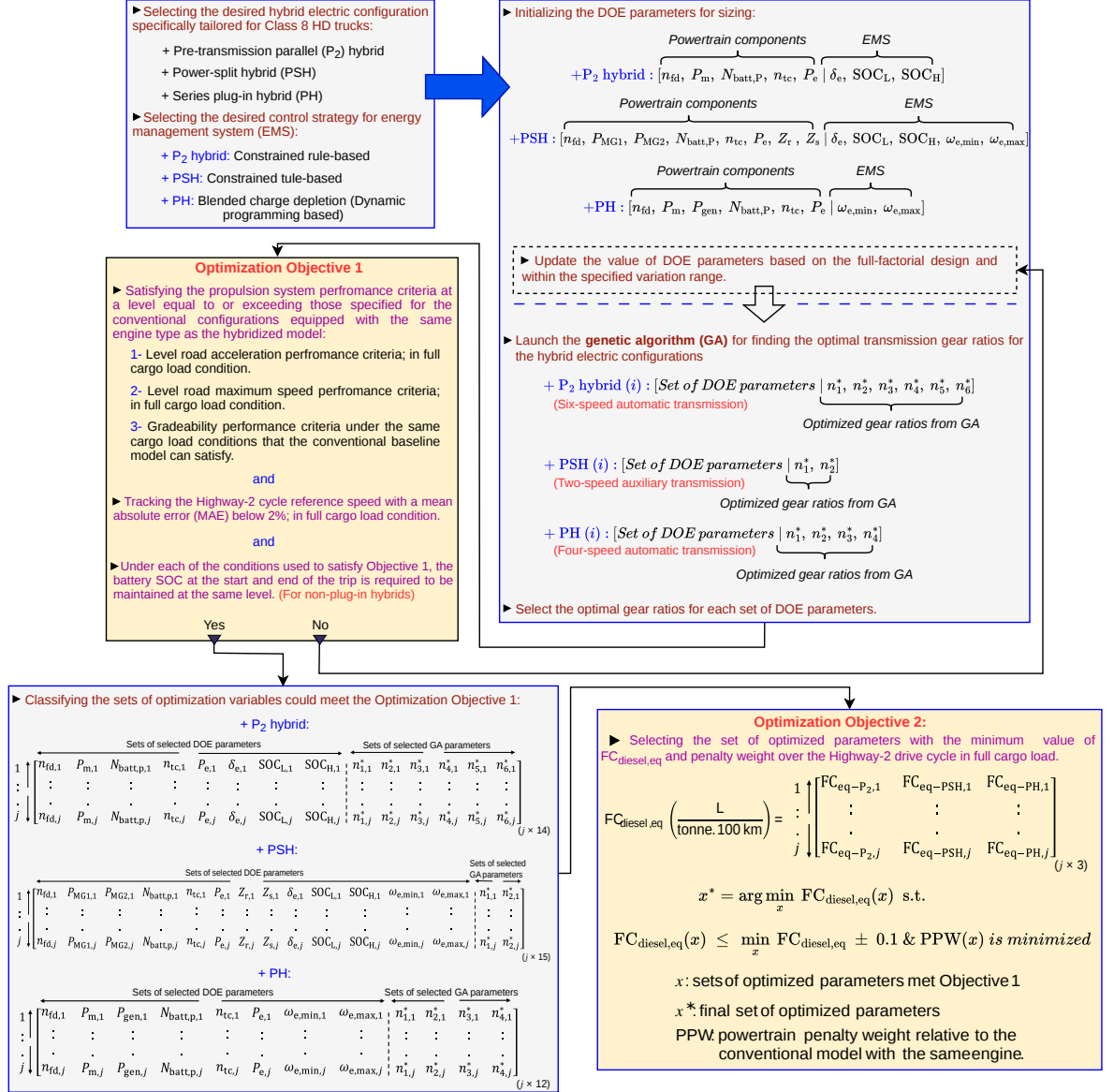


Figure 4.1: Schematic of the developed multi-stage optimization framework for optimal powertrain component sizing and tuning of EMS-related design parameters for the Class 8 hybrid electric truck configurations in this study. The hybrid configurations include pre-transmission parallel (P₂) hybrid, power-split hybrid, and series plug-in hybrid.

Satisfying the first optimization objective (Figure 4.1) involves three sub-constraints: 1) meeting the comparable propulsion system performance criteria, outlined in Section 4.2, as the conventional diesel baseline model, 2) tracking the Highway-2 drive cycle reference speed profile with a mean absolute error (MAE) of less than 2% under full cargo load condition, and 3) enforcing charge-sustaining operation for the non-plug-in hybrid electric configurations by constraining the battery SOC at the end of the drive cycle to match its initial value. Following the satisfaction of these constraints, the second optimization objective is addressed by selecting the configuration that yields the minimum fuel consumption from the set of DOE-generated candidates that meet all the first-objective requirements, which is then identified as the optimized configuration. The diesel-equivalent fuel consumption of the hybridized configurations equipped with either a diesel engine, or low-carbon dual-fuel diesel/NG or diesel/H₂ engines is calculated as follows:

$$\text{FC}_{\text{e,de}} \left(\frac{\text{L}}{\text{tonne}_{\text{cargo}} \cdot 100 \text{ km}} \right) = \frac{10^5 (M_{\text{diesel}} + M_{\text{LCF}} \text{LHV}_{\text{LCF}} / \text{LHV}_{\text{diesel}} + E_{\text{batt,net}} / \text{LHV}_{\text{diesel}})}{D \rho_{\text{diesel}} M_{\text{cargo}}} \quad (4.1)$$

where M_{diesel} and M_{LCF} represent the total mass of diesel and low-carbon fuels (e.g., NG or H₂) consumed by the ICE, respectively, expressed in grams. $E_{\text{batt,net}}$ denotes the net electrical energy consumption of the high-voltage battery over the drive cycle for the hybridized configurations, expressed in kJ. D corresponds to the total drive cycle distance in km, and ρ_{diesel} is the diesel fuel density, taken as 0.835 kg/L [218] in this study. The lower heating values (LHVs) of diesel, NG, and H₂ are assumed to be 42.6 MJ/kg, 48.9 MJ/kg, and 120 MJ/kg, respectively. M_{cargo} represents the transported cargo mass in metric tonnes. In the hybrid configurations, E_{batt} is first converted to an equivalent diesel fuel mass using the $\text{LHV}_{\text{diesel}}$ and then combined with the fuel directly consumed by the ICE. Consequently, the diesel-equivalent fuel consumption ($\text{FC}_{\text{e,de}}$) over the drive cycle is obtained, enabling a consistent comparison of energy consumption among the designed conventional and hybridized truck models.

After selecting each set of DOE variables for the considered hybrid electric configurations, as detailed in Figure 4.1, a GA is employed to optimize the transmission gear ratios. The GA flowchart and its optimization objectives, as part of the multi-stage optimal component sizing process, are presented in Figure 4.2. To initialize the population of candidate gear ratios, default gear ratio sets are adopted from representative HD transmissions, including the Fuller FSO-8406 6-speed [182], the Fuller AT-1202 2-speed [209], and the Allison MT-643 4-speed [204] automatic

transmissions for the parallel, power-split, and series plug-in hybrid configurations, respectively. The default gear ratios for the selected transmissions are provided in Figure 4.2. Subsequently, the GA generates random populations of gear ratios, ensuring they satisfy the performance criteria outlined as the first objective in Figure 4.1. The GA objective function (J) is then updated to compute the equivalent fuel consumption over the Highway-2 drive cycle. The best-performing individuals from the random populations are selected and recombined through crossover, selection, and mutation processes [213, 219]. Finally, a convergence check is performed on the most successful populations to identify the optimal gear ratios for each proposed hybrid configuration, as shown in Figure 4.2.

The optimization objective in Eq. (4.1) is limited to diesel-equivalent fuel consumption because it represents the net powertrain energy consumption of the vehicle over the drive cycle. For a given fuel, CO₂ emissions are directly proportional to fuel consumption; therefore, reducing fuel consumption also reduces CO₂ emissions. NO_x is not included in the objective function because the studied CI engines are assumed to use dedicated NO_x-control aftertreatment systems, such as SCR, which substantially reduce tailpipe NO_x. In addition, methane slip from HPDI NG engines is expected to be limited compared with premixed NG combustion due to direct injection and diesel-like diffusion-controlled combustion. Therefore, fuel consumption is used as the main optimization objective, while CO₂, NO_x, and CH₄ emissions are evaluated separately in the results.

4.3.1 Variation Ranges of DOE Parameters

The potential DOE variables for each of the proposed hybrid electric configurations, as denoted in Figure 4.1, are selected based on the critical powertrain design parameters and the key EMS-related parameters previously detailed in Chapter 3. The variation ranges of these parameters used in the DOE-based optimization process are determined based on the preliminary powertrain component sizing for each hybridized configuration. The procedure for preliminary component sizing in this study follows the standard methodology proposed in [105].

Pre-transmission parallel (P₂) hybrid

The selected DOE variables for optimal powertrain component sizing of the P₂ hybrid electric configuration, along with their corresponding variation ranges, are summarized in Table 4.2.

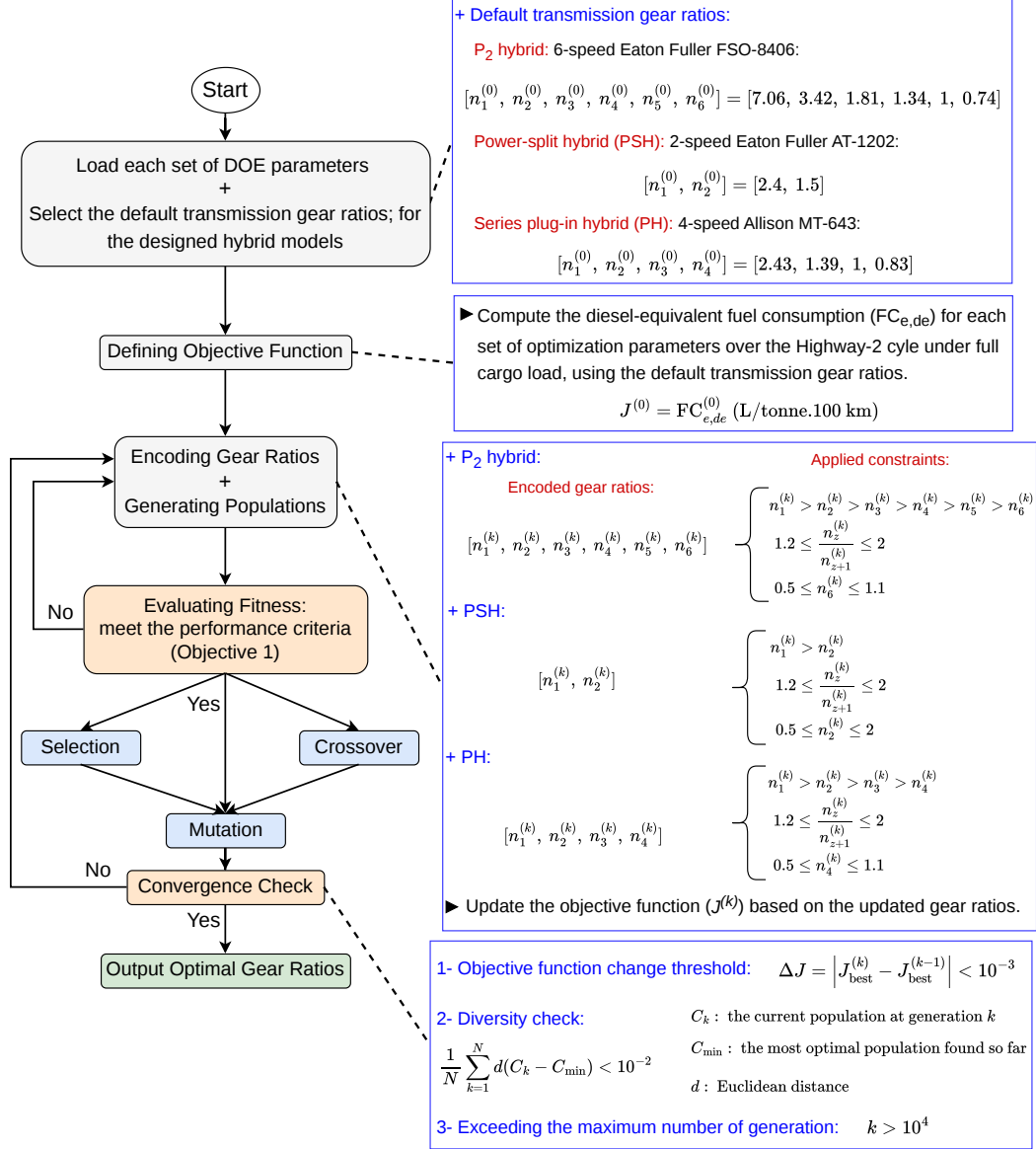


Figure 4.2: Flowchart of the developed genetic algorithm in this study as a part of the multi-stage optimal component sizing process for optimizing the automatic transmission gear ratios in the pre-transmission parallel (P₂) hybrid, power-split hybrid, and series plug-in hybrid configurations.

Table 4.2: Range of the DOE variables through the optimal component sizing framework for the designed Class 8 pre-transmission parallel (P₂) hybrid electric truck configuration in this study.

Variables (Unit)	Index	Variation Range	Increment Change
Final drive ratio (-)	n_{fd}	2-6	0.1
e-motor rated power (kW)	P_m	Preliminary Sizing	1
Torque coupler ratio (-)	n_{tc}	1-4	0.1
Number of parallel battery modules (-)	$N_{batt,p}$	Preliminary Sizing	1
Engine rated power (kW)	P_e	Preliminary Sizing	1
Engine efficiency threshold for EMS (%)	δ_e	0-0.03	0.001
Low battery SOC threshold for EMS (%)	SOC _L	0.2-0.6	0.01
High battery SOC threshold for EMS (%)	SOC _H	0.5-0.9	0.01

Throughout the DOE-based optimization, the variation range of the final drive ratio is assumed to be 2 to 6. This range is selected to comprehensively cover the spectrum of real-world HD truck applications, from fuel-efficient long-haul operations to high-torque heavy-haul and gradeability-demanding conditions [220, 221]. The variation range of the fixed-ratio torque coupler, which connects the e-motor output shaft to the driveline upstream of the gearbox, is assumed to be 1 to 4. This range enables flexibility in scaling the e-motor torque contribution, ensuring effective torque matching with the engine and the transmission while maintaining feasible rotational speed limits for the e-motor [124, 188].

The variation range of the engine efficiency threshold (δ_e), defined as an EMS-related variable within the DOE-based optimization, is specified between 0 and 0.03 (Table 4.2). This variable is used to determine the offset of the engine's low and high operating torque lines from its optimal torque line within the constrained rule-based control strategy previously described in Chapter 3 (Section 3.2.2). The initial battery SOC is set to 0.6, and charge-sustaining operation is enforced by requiring the SOC at the end of the drive cycle to return to the same level for the non-plug-in hybrid configurations. Accordingly, the variation ranges for the lower and upper battery SOC thresholds are defined to be 0.2 to 0.6 and 0.5 to 0.9 (Table 4.2), respectively. These ranges provide sufficient flexibility for the P₂ hybrid configuration to regulate engine on/off events over the drive cycle as part of the constrained rule-based control strategy.

Two approaches are considered in this study for sizing the propulsion system in the P₂ hybrid configuration: 1) using the same engine rated power and displacement volume as those of the conventional configurations detailed in Chapter 2, and 2) using a downsized engine as the conventional models. In the first approach, the engine

rated power is treated as a fixed parameter, identical to that of the conventional configurations, and is therefore excluded from the set of DOE variables.

For the design of Class 8 parallel hybrid electric truck configurations equipped with a downsized ICE, the engine can be sized to meet only the continuous load requirement during steady-state operation over a flat highway. The electrical driveline, including the high-voltage battery and the e-motor, supplements the engine during transient operating conditions such as acceleration, grade climbing, and peak power demands. Accordingly, the lower bound of the engine rated power variation range for the DOE-based optimization is determined based on the maximum tractive power required to overcome aerodynamic drag and rolling resistances over the reference Highway-2 drive cycle, as expressed in Eq. (4.2). The upper bound is set equal to the rated engine power of the corresponding conventional configuration.

$$P_{e,\min} = \max_{t \in [t_0, t_f]} \left\{ \frac{v(t)}{\eta_{t,e}} \left[\frac{1}{2} \rho_a C_d A_f v(t)^2 + M_v g (f_{r1} + f_{r2} v(t)) \right] \right\} \quad (4.2)$$

Where $P_{e,\min}$ denotes the minimum required engine power for steady-state operation, obtained from the maximum tractive power demand over the drive cycle. $v(t)$ is the vehicle speed profile over the Highway-2 drive cycle, and $\eta_{t,e}$ represents the overall drivetrain efficiency from the engine to the wheels, assumed as 0.9. ρ_a is the air density, C_d is the aerodynamic drag coefficient, and A_f is the frontal area of the vehicle. g is the gravitational acceleration, and f_{r1} and f_{r2} are the rolling resistance coefficients. M_v denotes the gross vehicle mass, assumed to be 39.5 tonne, consistent with the fully loaded conventional diesel baseline configuration in this study. The values of ρ_a , C_d , A_f , f_{r1} , and f_{r2} are selected to be the same as those of the conventional diesel configuration previously presented in Chapter 2 (Section 2.2).

The lower bound of the e-motor rated power variation range in the DOE-based optimization is determined to ensure coverage of the traction power requirements during acceleration events over the reference Highway-2 drive cycle, as expressed in Eq. (4.3). The upper bound is determined based on the e-motor power required to achieve acceleration of 0 to 100 km/h under full cargo load conditions over a flat highway, as expressed in Eq. (4.4).

$$P_{m,\min} = \frac{1}{t_f - t_0} \int_{t_0}^{t_f} \frac{v(t)}{\eta_{t,m}} \left[\delta_{mass} M_v \left| \frac{dv(t)}{dt} \right| \right] dt \quad (4.3)$$

$$P_{m,\max} = \frac{\delta_{mass} M_v}{2 \eta_{t,m} t_a} (V_f^2 + V_b^2) \quad (4.4)$$

In Eq. (4.3), $P_{m,\min}$ denotes the minimum required e-motor power and $\eta_{t,m}$ represents the overall drivetrain efficiency from the e-motor to the wheels, assumed as 0.9. δ_{mass} is the rotational mass factor accounting for drivetrain inertia, assumed as 1.04 [105]. In Eq. (4.4), t_a denotes the acceleration time for the 0–100 km/h test, assumed to be 65 s as specified in Table 4.1 for regional-haul HD trucks. V_f is the final vehicle speed corresponding to 100 km/h, and V_b is the base vehicle speed, representing the transition point between the constant torque and constant power operating regions of the e-motor, assumed to be 35 km/h [200, 206].

The minimum required battery energy capacity can be estimated based on the minimum e-motor rated power over the reference Highway-2 drive cycle under the full cargo load condition, as expressed in Eq. (4.5). Accordingly, the lower bound of the number of parallel battery modules in the DOE-based optimization is determined from Eq. (4.6). The upper bound of the number of parallel battery modules is set to 65, corresponding to a total battery capacity of 100 kWh [1, 131] for the Li-ion battery used in the P₂ hybrid electric configuration in this study. This upper bound is selected based on battery capacity ranges commonly assumed in previously published studies for non-plug-in hybrid HD truck applications [203, 211].

$$E_{\text{batt},\min} = \frac{1}{\Delta\text{SOC}_{\text{usable}}} \int_{t_0}^{t_f} \frac{P_{m,\min}}{\eta_{\text{batt}-m}} dt \quad (4.5)$$

$$N_{\text{batt},p,\min} = \frac{E_{\text{batt},\min}}{N_{\text{batt},s} Q_{\text{batt},\text{cell}}} \quad (4.6)$$

In Eq. (4.5), $E_{\text{batt},\min}$ denotes the minimum required battery energy capacity, $\Delta\text{SOC}_{\text{usable}}$ represents the allowable battery SOC window which is assumed from 0.2 to 0.9. $\eta_{\text{batt}-m}$ denotes the overall efficiency between the battery and the e-motor, accounting for inverter and power electronics losses, assumed to be 0.85. In Eq. (4.6), $N_{\text{batt},p,\min}$ represents the minimum required number of parallel battery modules, $N_{\text{batt},s}$ is the number of cells connected in series within each module, and $Q_{\text{batt},\text{cell}}$ denotes the energy capacity of an individual battery cell. In this study, as previously described in Chapter 3, for the P₂ hybrid electric configuration, each parallel module consists of 98 cells connected in series, with an individual cell capacity of 15.8 Wh.

Power-split hybrid

The selected DOE variables for optimal powertrain component sizing of the power-split hybrid electric configuration, along with their corresponding variation ranges, are summarized in Table 4.3.

Table 4.3: Range of the DOE variables through the optimal component sizing framework for the designed Class 8 power-split hybrid electric truck configuration in this study.

Variables (Unit)	Index	Variation Range	Increment Change
Final drive ratio (-)	n_{fd}	2-6	0.1
MG1 rated power (kW)	P_{MG1}	Preliminary Sizing	1
MG2 rated power (kW)	P_{MG2}	Preliminary Sizing	1
Torque coupler ratio (-)	n_{tc}	1-4	0.1
Number of parallel battery modules (-)	$N_{batt,p}$	Preliminary Sizing	1
Engine rated power (kW)	P_e	Preliminary Sizing	1
Ring gear number of teeth (-)	Z_r	75-200	1
Sun gear number of teeth (-)	Z_s	20-60	1
Engine efficiency threshold for EMS (%)	δ_e	0-0.03	0.001
Low battery SOC threshold for EMS (%)	SOC_L	0.2-0.6	0.01
High battery SOC threshold for EMS (%)	SOC_H	0.5-0.9	0.01
Engine minimum efficient speed for EMS (rpm)	$\omega_{e,min}$	Preliminary Sizing	10
Engine maximum efficient speed for EMS (rpm)	$\omega_{e,max}$	Preliminary Sizing	10

The variation ranges of the following variables in the DOE-based optimization process are identical to those of the P₂ hybrid electric configuration: the final drive ratio, the fixed-gear torque coupler ratio, the engine efficiency threshold for the EMS, and the lower and upper battery SOC thresholds for the EMS. In addition, the variation ranges for the engine rated power and the number of parallel battery modules are determined through the preliminary component sizing following the same procedure described for the P₂ hybrid electric configuration. The variation range of the MG1 rated power is determined using the same approach as that applied to the single e-motor in the parallel hybrid configuration.

For determining the variation ranges of the ring and sun gear tooth numbers in the planetary gearset, ranges are selected based on values reported in the published literature for HD truck applications (Table 4.3). The sun and ring gears are considered with a gear module of 1.75, consistent with commercially available specifications used in the Toyota Hybrid System (THS) for power-split hybrid electric vehicles [222, 223].

In the power-split hybrid configuration, the rated power of MG2, which is connected to the sun gear of the planetary gear set, is determined based on the extreme operating conditions of the power-split device. The maximum generating and motoring power requirements of MG2 are evaluated to define the variation range of its rated power within the DOE-based optimization. The maximum generating power of MG2 occurs at zero vehicle speed, where the engine power is entirely transferred to the electrical path through the planetary gear set [105, 140]. Under this condition, the generating power of MG2 is expressed as follows:

$$P_{MG2,gen}^{max} = T_{e,max} \omega_{e,min} \quad (4.7)$$

where $P_{\text{MG2,gen}}$ and $T_{\text{e,max}}$ denote the maximum generating power of MG2 and the maximum engine torque, respectively. $\omega_{\text{e,min}}$ corresponds to the minimum engine speed within the high-efficiency operating range. In this study, $\omega_{\text{e,min}}$ is defined as the minimum engine speed at which the engine torque reaches more than 98% of its maximum value.

The maximum motoring power of MG2 occurs at the maximum vehicle speed, where MG2 assists in meeting the total traction power demand at the wheels [105, 140]. This power is determined from the difference between the traction power required at the wheels and the power provided by the engine, considering the drivetrain gear ratios, as follows:

$$P_{\text{MG2,mot}}^{\text{max}} = \left| \frac{n_g n_{\text{fd}}}{(1 + Z_s/Z_r) r_w} T_{\text{e,max}} V_{\text{max}} - T_{\text{e,max}} \omega_{\text{e,max}} \right| \quad (4.8)$$

where $P_{\text{MG2,mot}}$ and V_{max} denote the maximum motoring power of MG2 and the maximum vehicle speed, assumed to be 120 km/h, respectively. $\omega_{\text{e,max}}$ corresponds to the maximum engine speed within the high-efficiency operating range. n_g and n_{fd} denote the gearbox and final drive ratios, respectively. Z_r and Z_s represent the number of teeth of the ring and sun gears of the planetary gearset, respectively. r_w is the effective wheel radius. As the initial guess for the maximum motoring power of MG2 in Eq. (4.8), n_g , n_{fd} , and r_w are assumed to be 1, 2.85, and 0.5 respectively. In addition, Z_r and Z_s are assumed to be 78 and 30, respectively, based on the available market data for planetary gearsets used in power-split hybrid electric vehicles [105, 222, 223].

Accordingly, the lower and upper bounds of the MG2 rated power are determined as the minimum and maximum of the generating and motoring power limits, respectively, as expressed in Eq. (4.9) and Eq. (4.10). These bounds are used to define the feasible search space for MG2 sizing within the DOE-based optimization framework.

$$P_{\text{MG2,min}} = \min(P_{\text{MG2,gen}}^{\text{max}}, P_{\text{MG2,mot}}^{\text{max}}) \quad (4.9)$$

$$P_{\text{MG2,max}} = \max(P_{\text{MG2,gen}}^{\text{max}}, P_{\text{MG2,mot}}^{\text{max}}) \quad (4.10)$$

The variation ranges of $\omega_{\text{e,min}}$ and $\omega_{\text{e,max}}$, as EMS-related DOE variables, are determined as expressed in Eq. (4.11) and Eq. (4.12), respectively.

$$\omega_{\text{e,min}} = \omega_{\text{e,min}}^{\text{ref}} + \Delta\omega, \quad \Delta\omega \in [-100, 100] \text{ rpm} \quad (4.11)$$

$$\omega_{e,\max} = \omega_{e,\max}^{\text{ref}} + \Delta\omega, \quad \Delta\omega \in [-100, 100] \text{ rpm} \quad (4.12)$$

where $\omega_{e,\min}^{\text{ref}}$ and $\omega_{e,\max}^{\text{ref}}$ represent the reference values of the minimum and maximum efficient speeds of the engine, respectively. $\Delta\omega$ denotes the variation applied to these reference speeds as a DOE parameter, bounded within ± 100 rpm.

Series plug-in hybrid

The selected DOE variables for optimal powertrain component sizing of the series plug-in hybrid electric configuration, along with their corresponding variation ranges, are summarized in Table 4.4.

Table 4.4: Range of the DOE variables through the optimal component sizing framework for the designed Class 8 series plug-in hybrid electric truck configuration in this study.

Variables (Unit)	Index	Variation Range	Increment Change
Final drive ratio (-)	n_{fd}	2-6	0.1
e-motor rated power (kW)	P_m	Preliminary Sizing	1
Generator rated power (kW)	P_{gen}	Preliminary Sizing	1
Torque coupler ratio (-)	n_{tc}	0.5-4	0.1
Number of parallel battery modules (-)	$N_{batt,p}$	Preliminary Sizing	1
Engine rated power (kW)	P_e	Preliminary Sizing	1
Engine minimum efficient speed for EMS (rpm)	$\omega_{e,\min}$	Preliminary Sizing	10
Engine maximum efficient speed for EMS (rpm)	$\omega_{e,\max}$	Preliminary Sizing	10

The variation ranges of the final drive ratio and the downsized engine rated power within the DOE-based optimization are determined following the same procedure previously detailed for the P₂ hybrid electric configuration. In addition, the variation ranges of the engine minimum and maximum efficient speeds are determined using the same procedure previously detailed for the power-split hybrid electric configuration. Due to the higher reliance of series plug-in hybrid vehicles on the electrical power source (i.e., battery and e-motor), a downsized engine is adopted for the series plug-in hybrid configuration in this study, compared to the full-sized engine used in the conventional baseline models.

There is no standardized sizing procedure for determining the high-voltage battery capacity of plug-in hybrid HD trucks based on their target all-electric range. Prior studies in [19, 104] suggest that the battery size for plug-in hybrid Class 8 trucks can be determined to cover approximately half of the driving range on a single battery charge in pure electric mode. In this study, it is assumed that the battery of the plug-in hybrid regional-haul configuration is fully charged after completing a distance equivalent to one Highway-2 drive cycle (265 km). For a total DDR of 530 km,

this corresponds to two charging events per day. Accordingly, the battery is sized to provide an all-electric range of 133 km, representing half of the expected driving range per charge, as follows:

$$E_{\text{batt,min}} = \frac{1}{2} \sum_{i=1}^N \left[\frac{1}{2} \rho_a C_d A_f v_i^3 + (f_{r1} + f_{r2} v_i) M_v g v_i \cos \theta_i + M_v g v_i \sin \theta_i + M_v v_i \frac{v_i - v_{i-1}}{\Delta t} \right] \Delta t \quad (4.13)$$

where N denotes the total number of discrete time steps over the drive cycle, and θ_i represents the road grade angle at each time step i . The summation is performed over the Highway-2 drive cycle with one repetition (265 km), and the resulting value is used to estimate the minimum battery capacity corresponding to half of the total traction energy demand over the full driving range. This is referred to as the minimum battery capacity because the total vehicle mass (M_v) is based on the fully loaded conventional diesel baseline model as 39.5 tonne, without accounting for the additional mass of the high-voltage battery pack. The values of ρ_a , C_d , A_f , f_{r1} , and f_{r2} are selected to be the same as those of the conventional diesel configuration previously presented in Chapter 2 (Section 2.2).

The lower bound of the number of parallel battery modules within the DOE-based optimization is calculated using the same equation as previously defined in Eq. (4.6) for the P₂ hybrid electric configuration, but with different parameter values. In the series plug-in hybrid configuration, the Li-ion battery is configured with 140 cells in series per module, each with a higher single-cell capacity of 124.1 Wh. The upper bound is selected by assuming that the total vehicle mass in Eq. (4.13) is increased by 5% to account for the additional weight of the high-voltage battery pack. This additional mass assumption is based on the available studies in the literature reporting the weight penalty associated with battery integration in plug-in hybrid HD trucks [19, 104].

The e-motor in the series plug-in hybrid configuration is required to satisfy the maximum traction power demand at the wheels over the drive cycle. The engine is not mechanically coupled to the driving wheels and operates solely to charge the battery through the generator. The lower bound of the e-motor rated power within the DOE-based optimization is calculated as expressed in Eq. (4.14) to overcome the applied road resistive forces as well as the traction power required for acceleration. This lower bound is evaluated over the reference Highway-2 drive cycle under the full cargo load condition. The fully loaded vehicle mass is considered as 41.5 tonne, which is calculated based on the fully loaded mass of the conventional diesel baseline model

(39.5 tonne) with an additional 5% weight penalty due to the larger battery. The upper bound of the e-motor rated power is determined from Eq. (4.14) under a constant speed of 56 km/h on a 7% constant-grade road, as specified by the gradeability criterion previously detailed in Table 4.1. For this condition, the gross vehicle mass is assumed at 34% of the full cargo load, consistent with the gradeability performance of the conventional diesel baseline model.

$$P_{m,\min} = \max_{t \in [t_0, t_f]} \left\{ \frac{v(t)}{\eta_{t,m}} \left[\frac{1}{2} \rho_a C_d A_f v(t)^2 + M_v g (f_{r1} + f_{r2} v(t)) \cos \theta(t) + M_v g \sin \theta(t) + M_v \frac{dv(t)}{dt} \right] \right\} \quad (4.14)$$

The generator in the series plug-in hybrid configuration is sized according to the engine rated power, since the engine is not mechanically coupled to the driving wheels and operates only to charge the battery through the generator. A fixed-gear torque coupler exists between the engine and the generator to adjust the torque and speed of the generator, enabling operation within its high-efficiency region. The lower bound of the generator rated power within the DOE-based optimization is determined as expressed in Eq. (4.15), based on the maximum mechanical power delivered from the engine to the generator over the drive cycle. This lower bound is evaluated over the reference Highway-2 drive cycle under the full cargo load condition. The upper bound of the generator rated power is set equal to the rated power of the downsized engine, assuming a direct coupling between the engine and the generator and neglecting coupling losses.

$$P_{\text{gen},\min} = \max_{t \in [t_0, t_f]} (T_{\text{gen}}(t) \omega_{\text{gen}}(t)) = \eta_{tc} \max_{t \in [t_0, t_f]} (T_e(t) \omega_e(t)) \quad (4.15)$$

Where $P_{\text{gen},\min}$ denotes the minimum required generator rated power, and T_{gen} and ω_{gen} represent the generator torque and speed as functions of time, respectively. η_{tc} denotes the mechanical efficiency of the fixed-gear torque coupler between the engine and the generator, assumed as 0.98. The variation range of the torque coupler ratio is assumed between 0.5 and 4 to allow sufficient flexibility in adjusting the generator speed and torque for efficient operation across the engine operating range.

4.3.2 Estimation of Powertrain Penalty Weight

After determining the variation ranges of the selected DOE variables for the proposed Class 8 hybrid electric truck configurations in this study, the multi-stage optimization process, can be performed. Throughout the optimal component sizing process, as different DOE variable sets are evaluated, the gross vehicle weight of the designed

hybridized configurations is estimated. This section presents a simplified approach to estimate the powertrain penalty weight of the designed hybridized configurations relative to the conventional diesel baseline configurations previously developed in Chapter 2.

In this study, it is assumed that the designed hybrid electric truck configurations maintain the same maximum cargo capacity and trailer configuration as the conventional diesel baseline model. Therefore, the inclusion of additional electrical and mechanical powertrain components, as well as modifications to the sizes of certain components, leads to a mass penalty in the tractor of the hybridized configurations relative to the conventional counterparts. To estimate this mass penalty, the weight densities of the primary hybrid components are summarized in Table 4.5.

Table 4.5: Powertrain components weight estimation for the designed pre-transmission parallel (P_2), power-split, and series plug-in hybrid electric configurations in this study. The numbers in this table are taken from [52, 143, 182, 188, 200, 203, 206, 207, 209, 211, 221].

Component	Weight	Component	Weight
Electric Machine (MG1, MG2, gen)	power (kW) $\in$ [50, 125] $\rightarrow$ 63 kg power (kW) $\in$ [125, 215] $\rightarrow$ 105 kg power (kW) $\in$ [215, 320] $\rightarrow$ 145 kg power (kW) $\in$ [320, 410] $\rightarrow$ 185 kg power (kW) $\in$ [410, 550] $\rightarrow$ 230 kg	Planetary Gearset	0.37 kg/teeth ²
High-Voltage Li-Ion Battery +Packaging	5 kg/kWh	Direct Current Power Converter	buck converter $\rightarrow$ 4 kg bidirectional converter $\rightarrow$ 30 kg
Automatic Transmission	28.4 kg/gears ¹	Fixed-Gear Torque Coupler	10 kg/gear

¹ ‘gears’ refers to the total number of gears in the automatic transmission.

² ‘teeth’ refers to the total number of teeth from both the sun (Z_s) and ring (Z_r) gears in the planetary gearset.

The weight of the electric machines, including the e-motor in the P_2 hybrid, MG1 and MG2 in the power-split hybrid, and the generator and e-motor in the series plug-in hybrid configurations, is determined based on their rated power ranges using PMSM manufacturer data for electrified vehicles [200, 206, 208]. The weight density of the high-voltage battery, including packaging, is specified in Table 4.5 based on the weight densities of commercially available NMC Li-ion batteries for electrified vehicles [52, 203, 211]. The weight of the multi-speed automatic transmissions in the hybridized truck configurations is estimated on the basis of a gear-dependent weight density. A value of 28.4 kg/gear is assumed, which is consistent with commercial data

reported for HD truck gearboxes [182, 209, 220]. The weight density of the planetary gearset in the power-split hybrid configuration is estimated based on the total number of teeth of the sun and ring gears, using commercially available data for power-split devices [143, 223]. The weight density of the mechanical torque coupler is estimated based on a gear-dependent weight density [124, 188, 221]. Since the torque couplers in the designed hybridized configurations are single-gear, a mass of 10 kg is assumed for each configuration.

According to power converter manufacturer data [203, 207], the mass of buck converters used to supply 2–6 kW of electrical accessory power at a constant 12 V is assumed to be 4 kg, as summarized in Table 4.5. However, the mass of bidirectional boost converters used between the high-voltage battery and e-motors in the power-split hybrid configuration is assumed as 30 kg. These converters regulate the 400 V battery output (up to 300 kW) to the required 200–500 V voltage range for e-motor operation in both propulsion and regenerative modes.

The engine is another key contributor to the difference in powertrain weight between the hybrid electric configurations and the conventional diesel baseline model. In the proposed hybridized configurations in this study, the baseline diesel engine is replaced with low- and zero-carbon-fueled engines with different specifications, displacement, and fuel supply systems. Depending on the level of engine downsizing and fuel system requirements, this replacement can lead to either an increase or a reduction in weight relative to the baseline model. The dry weight of these engines, including aftertreatment and fuel supply systems, will be specified in the following chapters to estimate the total powertrain penalty weight.

4.4 Standard Drive Cycles for Performance Assessment

The equivalent fuel consumption of the designed Class 8 hybrid electric truck configurations is minimized over the real-world Highway-2 drive cycle within the optimal component sizing framework. After determining the optimal powertrain component sizes, the performance of the designed configurations is further evaluated under broader driving conditions, including both urban and highway scenarios. For this purpose, two standard drive cycles suitable for Class 8 HD trucks are selected [1]: 1) the Heavy Heavy-Duty Diesel Truck (HHDDT-65) cycle for highway driving, and 2) the City Suburban Heavy Vehicle Route (CSHVR) cycle for urban driving. The speed-time profiles of these standard drive cycles which neglect road grade variations

are shown in Figure 4.3.

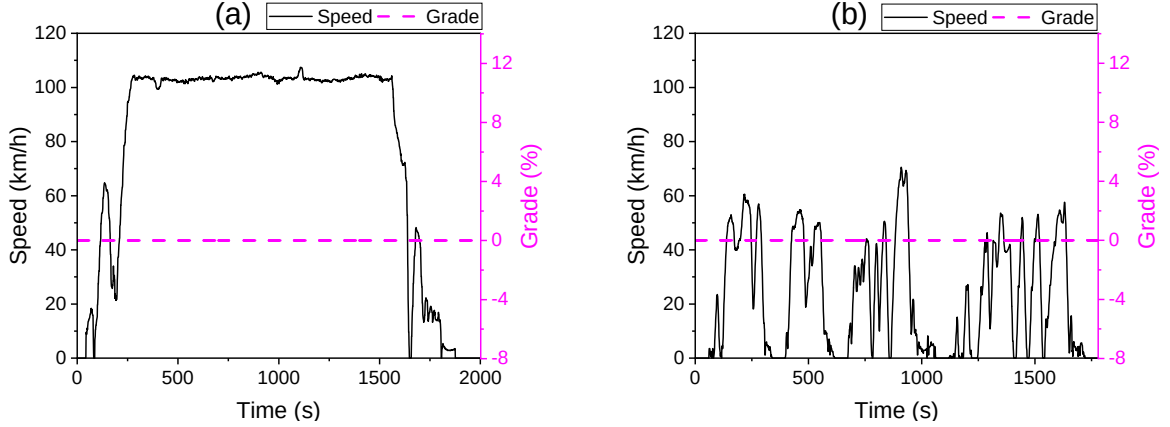


Figure 4.3: Selected standard drive cycles for testing the performance of the designed Class 8 regional-haul truck configurations in this study: (a) HHDDT-65 (highway) and (b) CSHVR (urban).

Based on standard thresholds used for classifying driving modes in HD drive cycles [179, 180], the HHDDT-65 cycle is dominated by cruising operation (79.7%), with smaller contributions from idling (4.5%), acceleration (8.6%), and deceleration (7.2%), a maximum speed of 107.3 km/h, and total length of 42.4 km. The CSHVR driving modes include: 27.4% cruising, 20.6% idling, 28.7% acceleration, 23.3% deceleration along with a maximum speed of 70.6 km/h and total length of 10.9 km.

4.5 Summary of Chapter

This chapter developed a multi-stage component sizing framework to optimize the sizes of primary powertrain components, as well as to tune the key EMS-related variables for the designed hybrid Class 8 truck configurations. The developed optimization framework aims to minimize net powertrain energy consumption over the real-world Highway-2 regional-haul drive cycle while maintaining propulsion system performance comparable to that of the conventional diesel baseline model in this study. This framework employs a full-factorial design of experiments (DOE) in combination with a genetic algorithm (GA). The GA is employed to optimize the gear ratios of the automatic transmission, while the DOE is used to optimize the sizes/ratios of the primary components and the EMS-related variables for each hybridized configuration.

The set of critical variables used in the optimal component sizing framework was defined for each of the proposed hybrid electric truck configurations based on their

powertrain architectures and the corresponding control strategies, as previously developed in Chapter 3. This selection ensures an accurate representation of the key design and operational characteristics in the optimization process. To determine the feasible variation ranges of the optimization variables within the framework, a preliminary component sizing formulation was presented for each proposed hybridized configuration.

To assess the performance of the optimized hybridized configurations (after completing the component sizing framework), under broader driving conditions, two standard drive cycles suitable for Class 8 HD trucks were introduced. These include the HHDDT-65 drive cycle for highway driving and the CSHVR drive cycle for urban driving.

In the next chapter, the existing gradeability performance criterion, used as part of the optimal component sizing framework to ensure that the hybridized truck configurations maintain propulsion system performance comparable to the conventional baseline model, is revised. The existing criterion, developed exclusively for conventional Class 8 trucks, evaluates gradeability over a constant 7% road grade at a constant speed of 56 km/h. This criterion is extended for hybrid electric truck configurations by incorporating varying road grade profiles derived from real-world steep highways. This approach evaluates the feasibility of engine downsizing in hybridized Class 8 truck configurations while maintaining performance comparable to their conventional counterparts.

Chapter 5

Real-World Steep Drive Cycles for Gradeability Testing of Class 8 Hybrid Electric Trucks¹

Gradeability is a key determinant for powertrain component sizing of class 8 HD trucks; thus, knowing real-world steep roads along with their grade profiles are essential for design and performance analysis of commercial HD trucks. This chapter refines the traditional gradeability criterion established by the SAE J2807 standard for conventional HD trucks, tailoring it to address the testing requirements of hybridized HD trucks with downsized ICEs. To achieve this, fifteen real-world steep highways suitable for HD truck operations are identified from different regions, including the U.S., Canada, Europe, and China. An algorithm is developed to obtain grade versus distance profiles of these newly identified steep highways using GPS data from Google Earth Pro[®], including latitude, longitude, and altitude. These profiles serve as benchmarks for assessing the gradeability performance of both conventional and pre-transmission parallel (P₂) hybrid class 8 truck models. The optimal powertrain component sizing, previously proposed in Chapter 4, is then refined for both conventional and hybrid models using each of the identified steep highways as the gradeability criterion. Then, the diesel-equivalent fuel consumption of the hybridized and conventional truck models, with the refined component sizing, is compared over the representative Highway-2 drive cycle. This provides insight into the feasibility of engine downsizing in hybridized Class 8 trucks while maintaining propulsion system performance comparable to conventional counterparts, while enabling improved fuel economy over a regional-haul drive cycle. This superiority is attributed to the hybrid's ability to maximize engine efficiency, reduce engine-on duration, and utilize

¹This chapter is fully based on the published journal paper in [3].

regenerative braking.

5.1 Introduction and Background

Standard highways suitable for HD commercial trucks are classified by the institute of transportation engineers (ITE) based on the maximum allowable speed and traffic flow [224]. These highways are categorized as: 1) single-lane undivided highways, have one lane in each direction without a median barrier, 2) single-lane divided highways, use a median barrier for added safety, 3) multi-lane undivided highways, offering two or more lanes per direction without median separation, 4) multi-lane divided highways, featuring multiple lanes in each direction separated by a median barrier for high-speed and high-traffic use (e.g., 80 vehicles per km), and 5) controlled access highways (expressways), with limited entry and exit points and a median barrier separating opposing traffic, designed specifically for high-speed travel. The U.S. federal highway administration (FHWA) defines maximum speed limits for the majority of single-lane highways as 80-110 km/h, for multi-lane highways as 90-120 km/h, and for controlled access highways as 120-130 km/h. Minimum allowable speed limits for HD trucks may vary depending on jurisdiction, safety, road conditions, and regulations, and may not always be explicitly defined [225]. According to FHWA, on North American highways equipped with shoulders for emergency stopping or passing, the minimum speed limit for HD trucks ranges from 50 km/h to 80 km/h to maintain consistency with traffic flow [226]. For Class 8 tractor-trailers, gradeability performance should be evaluated based on the grade profiles of standard steep highways, typically with maximum grades exceeding 6%, while maintaining vehicle speeds within the specified minimum allowable limits (i.e., 50-80 km/h) [165].

According to the SAE J2807 standard [165], the gradeability of conventional HD trucks is evaluated at a constant speed of 56 km/h over the initial steepest segment (19.7 km) of the 44.4 km Davis Dam route, located in Arizona, U.S., which has a maximum grade of 7%. In evaluating the gradeability of conventional trucks, the critical parameter is the maximum road gradient. If the ICE, as the sole power generation source, can meet the power demand at the maximum road grade, it inherently has sufficient capability to operate under lower grade conditions. Therefore, instead of using the real-world varying road grade profile, studies in the literature [19, 82, 104, 167] adopt a constant 7% grade profile for gradeability assessment of conventional HD trucks.

In contrast, in hybrid propulsion systems, which incorporate an additional power

source such as a high-voltage battery alongside the ICE, the electrical propulsion system must be able to supply a portion of the road demanding power. In addition, in non-plug-in hybrid configurations, the battery SOC at the end of the trip should be maintained at the same level as at the beginning. This ensures that the performance criteria remain at least on par with conventional trucks [104, 154]. Therefore, for testing gradeability of hybrid electric HD trucks, besides considering the maximum road gradient, other crucial parameters come into play, including: 1) the length of the longest continuous distance with a slope surpassing the ICE rated power to meet road power demands, and 2) portion of the regenerative brake opportunities throughout the road by predetermining the trip's starting and ending points. As a result, for gradeability evaluation of hybridized configurations, particularly when engine downsizing is the approach, the entire varying grade profile of the real-world steep routes must be considered to capturing both uphill and downhill segments.

5.2 Performance of the Conventional Baseline Model Over the Davis Dam Road

The Class 8 conventional diesel baseline model in this study was developed and validated in Chapter 2 using the real-world Highway-2 regional-haul drive cycle. In this section, gradeability of the baseline vehicle is tested according to the SAE J2807 standard. For gradeability assessment, it's recommended to focus on the initial 19.7 km of the Davis Dam road, encompassing its steepest portion with the maximum and minimum grade of 7% and -1.2%, respectively [165]. This segment begins at the junction of Arizona-SR-68 and McCormick Boulevard (Bullhead Parkway) and concludes at the intersection with Old Kingman Highway (Union Pass), exhibiting an average grade of 4.8% [165, 227].

The performance of the conventional baseline model over the first 19.7 km of the Davis Dam road is shown in Figure 5.1. The grade versus distance profile was converted to the grade versus time, assuming the constant speed of 56 km/h as stipulated by the SAE J2807 standard. The baseline model, equipped with the engine rated power of 336 kW is capable of meeting the most power demanding segment of the Davis Dam road (7% grade), with a cargo load of 8,045 kg, representing 34% of its total cargo capacity. The simulated vehicle speed versus the drive cycle reference speed, engine power fluctuations throughout the drive cycle, and the corresponding gear selection at each interval are shown in Figure 5.1.

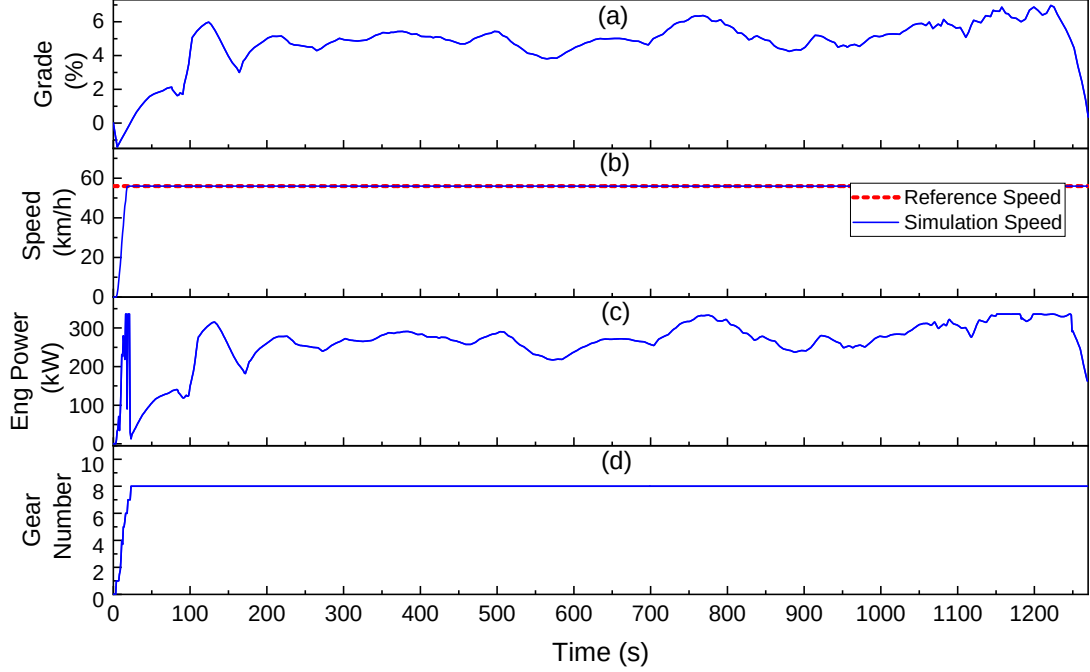


Figure 5.1: Class 8 regional-haul conventional model performance over the steepest 19.7 km section of the Davis Dam road at a constant speed of 56 km/h and with the cargo load of 8,045 kg (GVWR: 23,885 kg). (a) Road grade vs. time, (b) Simulation and reference speeds vs. time, (c) Engine power vs. time, and (d) Gear number vs. time.

5.3 Developing Grade vs. Distance Profiles of the World Steepest Highways

In this section, extending beyond the Davis Dam road, several additional steep highways around the world, that are used by HD trucks frequently, are identified. Next, a methodology is represented for computing grade versus distance profiles of the proposed steep roads utilizing GPS data on latitude, longitude, and altitude.

5.3.1 Classification of Steep Highways

When assessing hybridized powertrains' gradeability with a focus on optimum powertrain component sizing, considerations must extend beyond just including the maximum road gradient. Other factors such as continuous distance with slopes exceeding the ICE rated power and the proportion of downhill sections for regenerative braking are also crucial. Therefore, it is imperative to consider these additional criteria across representative real-world grade profiles when testing the gradeability of hybridized powertrains that use both an ICE and an electric source, such as battery, for power generation.

To identify real-world steep grade profiles for gradeability testing of both conventional and electrified Class 8 HD trucks, this study proposes four criteria, as detailed in Table 5.1. Highways frequently used by HD trucks can be identified as suitable for gradeability testing and classified into three groups based on steepness: Tier 1, Tier 2, and Tier 3 (Table 5.1). Tier 1 includes the steepest highways according to the selected criteria, while Tier 2 and Tier 3 comprise steep highways with lower level of steepness compared to Tier 1, respectively.

Table 5.1: Criteria for classifying steep highways for Class 8 HD trucks in this study.

Steep Highway Classification	Criteria for Identifying Steep Highways			
	Max. Grade (%)	Percent of Downhill Distance (%)	Longest Continuous Distance with Grade $\geq 4.8\%$ (km)	Cumulative Elevation Gain (m)
Tier 1	≥ 8	≤ 45	> 4.3	≥ 1400
Tier 2	≥ 7	≤ 55	≥ 2	≥ 1000
Tier 3	≥ 6	≤ 55	≥ 1	≥ 500

The criteria for classifying the steep highways in this paper (Table 5.1) include: 1) maximum road grade, 2) total percent of downhill distance, 3) longest continuous distance with a slope exceeding the 4.8% average grade of the initial 19.7 km of the Davis Dam road, and 4) cumulative elevation gain. The fourth criterion is used as a metric for assessing the difficulty and energy requirements for vehicle gradeability testing [228]. It refers to the total increase in elevation encountered over a route, reflecting the sum of all positive altitude changes. The cumulative elevation gain (G) for a particular route can be calculated as follows:

$$G = \sum_{j=1}^{h-1} \max(\text{Altitude}_{j+1} - \text{Altitude}_j, 0) \quad (5.1)$$

where, Altitude_{j+1} and Altitude_j represent the elevations of two consecutive points along the route, and h is the total number of elevation points obtained from GPS data.

For the initial 19.7 km of the Davis Dam, the criteria shown in Table 5.1 can be calculated from the available GPS data in [165]. The maximum road grade is 7%, the total percentage of downhill sections is 1.4%, the longest continuous distance with a slope exceeding 4.8% is 4.3 km, and the elevation gain is 1021.7 m. Thus, the 19.7-km-long Davis Dam road is classified as a Tier 2 steep highway based on the

classification presented in Table 5.1.

To establish the acceptable range for each of the four criteria in Table 5.1 for classifying suitable steep highways into three tiers, two key constraints were applied: 1) the selected steep highways must exceed the demands of the Highway-2 cycle in each of the four specified criteria, and 2) the Tier 2 criteria ranges were set so that the grade profile of the 44.4-km-long Davis Dam road falls within this tier. In the case of the Highway-2 cycle, the maximum road grade is 5.3%, the percent of downhill distance is 55%, the longest continuous distance with grade upper than 4.8% is less than 1 km, and the cumulative elevation gain is 496.7 m. Accordingly, the criteria ranges for Tier 3 in Table 5.1 were set to be more demanding than the Highway-2 cycle. For the entire 44.4-km-long Davis Dam road, the corresponding ranges are 7%, 25.9%, 4.3 km, and 1192.2 m, respectively. As a result, the criteria ranges for Tier 2 steep highways in Table 5.1 were determined to encompass the characteristics of the Davis Dam road. The average road grade over the first 19.7 km of the Davis Dam road, which is the most power demanding segment, is 4.8%. Therefore, this value was chosen as the threshold for the longest continuous distance with a high grade, as specified in the third criterion. The Tier 1 criteria ranges are established to include the selected steep roads with profiles that are more demanding than the Davis Dam road, which is classified in Tier 2.

5.3.2 Identified Real-World Steep Highways

There might be other steep roads worldwide used by HD commercial trucks that have higher maximum gradients, longer uphill sections, and greater elevation gains than the Davis Dam. Hence, to determine optimal component sizing for hybridized Class 8 trucks on a predetermined route, assessing gradeability across a broader range of steep roads, rather than relying solely on a portion of the Davis Dam road, provides higher assurance. This section identifies 15 steepest highways, including the entire 44.4-km-long Davis Dam road, from regions such as the U.S., Canada, Europe, and China that meet the four deterministic criteria, detailed in Table 5.1. The identified steep roads were ensured to be selected from categorized highways suitable for HD truck operation. Additionally, they were selected from mountainous regions across different countries from three continents in the world. These include sample highways with steep grade profiles. It is not claimed that the selected list encompasses all steep highways or represents the steepest roads worldwide. However, it is claimed that these highways are comparable to the Davis Dam road, the only available standard, and are more power demanding than the Highway-2 drive cycle, as a representative regional-

haul drive cycle for Class 8 HD trucks. Schematic route profiles of the selected steep highways are shown in Figure 5.2, and their geographical information is provided in Table 5.2.

1- U.S. Steep Highways

Arizona-68 Highway (Davis Dam): This route is a multi-lane divided highway which is located in the State of Arizona, connecting the Bullhead City and Kingman County. Although the initial 19.7 km segment of this road was chosen according to the SAE J2807 standard for testing the gradeability of conventional trucks, the entire 44.4 km stretch along this route is designated for hybridized configurations in this study.

Victory Highway (Route-40): This route is classified as a multi-lane undivided highway that passes through multiple U.S. states including Maryland, Pennsylvania, West Virginia, Ohio, Indiana, Illinois, Missouri, Kansas, Colorado, Utah, and Nevada. The steepest segment that was selected for this study starts from the intersection of 32nd state road and continuous to the Lincoln Highway at North Snyderville Basin, in Utah. The selected route's total length is 20.8 km.

Highway-14: This route runs through several U.S. states including Minnesota, South Dakota, Wyoming, and Montana. The steepest part that was selected for this study is classified as a single-lane undivided highway, starts from the eastern part of Bighorn River near the Greybull City and continues to the Burgess Junction at the southern parts of Bull Elk Park natural area, in Wyoming. The total length of the selected route is 75.1 km.

Highway-160: This route stretches across multiple U.S. states including Arizona, New Mexico, Colorado, Kansas, and Missouri. The steepest part that was chosen for this study is classified as a multi-lane undivided highway, begins from the eastern part of Pagosa Springs City and continuous to the South Fork Town at the southern parts of Rio Grande National Forest. The total length of the selected route is 64.6 km.

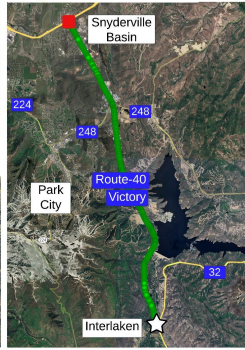
Highway-149: This route is categorized as a single-lane undivided highway which is located in the State of Colorado, connecting the towns of South Fork and Creede. The steepest segment that was selected for this study starts from the Lake City Bridge over the Gunnison River and continues southward to the northwestern parts of Spar Town across the Rio Grande River. The selected route's total length is 134.1 km.

The United States of America

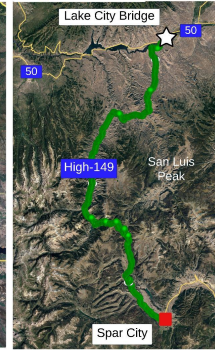
(a) AR-68 (Davis Dam), Length: 44.4 km



(b) Victory (Route-40), Length: 20.8 km



(c) High-149, Length: 134.1 km

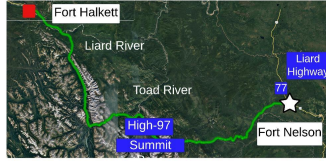


(d) High-14, Length: 75.1 km (e) High-160, Length: 64.6 km



Canada

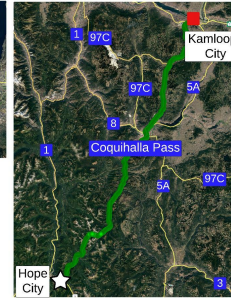
(a) Summit Pass, Length: 305.2 km



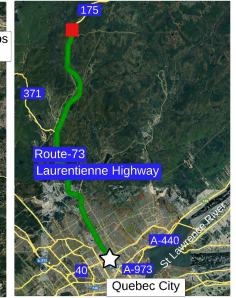
(b) Okanagan (High-97C), Length: 85.2 km



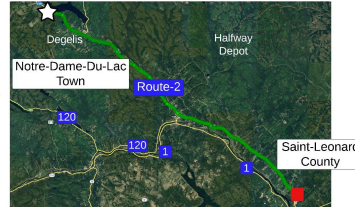
(c) Coquihalla Pass, Length: 184.2 km



(d) Laurentienne (Route-73), Length: 34.1 km

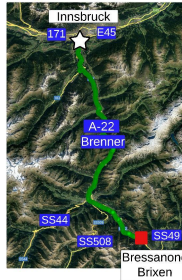


(e) Claude-Bechard (Route-2), Length: 79.1 km

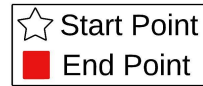


Europe

(a) Brenner (A-22), Length: 65.2 km



(b) N-432, Length: 42.2 km



China

(a) G-6, Length: 141.3 km



(b) G-0613, Length: 47.5 km



(c) G-309, Length: 82.7 km



Figure 5.2: Geographical maps of the identified steep highways from the U.S., Canada, Europe, and China in this study. Reprinted from my previously published work in [3], licensed under CC BY 4.0.

2- Canadian Steep Highways

Coquihalla Pass: This route is classified as a controlled access expressway which is located in the Province of British Columbia, connecting the cities of Hope and Kamloops. The steepest section of this road which was selected for this study starts from the Coquihalla Canyon provincial park in the eastern parts of the Hope district and continues to the Kamloops City. The total length of the selected route is 184.2 km.

Summit Pass: This route is a part of Alaska Highway-97 that crosses the summit of the Rockies mountains in the Province of British Columbia. The steepest section that was selected for this study is categorized as a single-lane undivided highway, begins from the Fort Nelson region at the intersection with Liard Highway and continues to the west to reach Fort Halkett beside the Smith River. Total length of the selected route is 305.2 km.

Okanagan Highway: This highway, also known as Highway-97C, is classified as a multi-lane divided highway which connects the City of Merritt in the west to the City of Kelowna in the east in the Province of British Columbia. The steepest part which was selected for this study starts from the eastern part of Merritt City at the end of Princeton Kamloops Highway and continues to the east to Kelowna City. The total length of the selected route is 85.2 km.

Laurentienne Highway (Route-73): This route is classified as a multi-lane divided highway which is located in the Province of Quebec. The steepest part which was chosen for this study starts from the Saint-Roch region at the intersection with Felix-Lecrec Highway in Quebec City and continues to the beginning part of Antonio Talbot Highway. The total length of the selected route is 34.1 km.

Claude-Becharde Highway (Route-2): This route is classified as a multi-lane divided highway which is located in New Brunswick Province. The steepest part which was selected for this study starts from the eastern part of Notre-Dame-Du-Lac Town and continues towards southeast to Saint-Leonard County. The total length of the selected route is 79.1 km.

3- European Steep Highways

Brenner Highway (A-22): This route is categorized as a controlled access expressway which runs primarily through Austria and Italy, in the Alps region. The steepest part which was selected for this study begins at the southern part of Innsbruck in Austria and continues to the Brixen City in South Tyrol, northern Italy. The total

length of the selected route is 65.2 km.

N-432 Highway: Carretera de Badajoz a Granada (N-432) runs from Badajoz City in the west to Granada City in the southeast of Spain. The steepest part which was selected for this study is categorized as a single-lane undivided highway, begins at the northern part of Cordoba City and continues to the Espiel City. The total length of the selected route is 42.2 km.

4- Chinese Steep Highways

G-6 Highway: Beijing–Lhasa (G-6) road is classified as a controlled access expressway that passes through several provinces and regions, including Hebei, Shanxi, Henan, Shaanxi, Hubei, Chongqing, Sichuan, and Tibet. The steepest part which was selected for this study begins in Xining City in the western China and continues to Hainan Tibetan in northeastern Qinghai Province. The total length of the selected route is 141.3 km.

G-309 Highway: This road passes through several provinces, including Qinghai, Gansu, Ningxia Hui Autonomous Region, Shaanxi, and Henan with a total length of approximately 2,200 km. The steepest part which was selected for this study is classified as a single-lane undivided highway, begins in Xiji County in Guyuan Province and continues to Huining County in Gansu Province. The total length of the selected route is 82.7 km.

G-0613 Highway: Xining-Lijiang (G-0613) road is categorized as a multi-lane undivided highway that runs through the Province of Sichuan and connects several cities and towns in the region. The steepest part which was selected for this study connects Gonghe County to Yushu County. The total length of the selected route is 47.5 km.

Table 5.2: Main characteristics of the identified steep highways in this study for HD trucks in the U.S., Canada (CA), Europe (EU), and China (CN).

Highways Name	Location	Highway Classification	Length (km)	Elevation (m) Min. Max.	Start Point Location Lat. Lon.	End Point Location Lat. Lon.
Arizona-68 (Davis Dam)	U.S., Arizona	Multi-lane divided	44.4	160.7 1100.1	35.173 -114.565	35.220 -114.128
Victory (Route-40)	U.S., Utah	Multi-lane undivided	20.8	1746.2 2133.1	40.560 -111.427	40.731 -111.497
Highway-14	U.S., Wyoming	Single-lane undivided	75.1	1163.1 2753.1	44.488 -108.044	44.770 -107.518
Highway-160	U.S., Colorado	Multi-lane undivided	64.6	2183.2 3310.1	37.277 -106.984	37.664 -106.645
Highway-149	U.S., Colorado	Single-lane undivided	134.1	2293.2 3523.3	38.487 -107.059	37.726 -107.034
Coquihalla	CA, British Columbia	Multi-lane expressway	184.2	192.2 1448.3	49.366 -121.355	50.664 -120.457
Summit (High-97)	CA, British Columbia	Single-lane undivided	305.2	382.6 1306.1	58.901 -123.130	59.553 -126.478
Okanagan (High-97C)	CA, British Columbia	Multi-lane divided	85.2	426.4 1733.1	49.951 -120.617	49.828 -119.637
Laurentienne (Route-73)	CA, Quebec	Multi-lane divided	34.1	15.8 363.3	46.835 -71.262	47.100 -71.345
Claude-Bechar (Route-2)	CA, New Brunswick	Multi-lane divided	79.1	141.5 306.6	47.593 -68.758	47.210 -67.931
Brenner (A-22)	EU, Austria/Italy	Multi-lane expressway	65.2	630.5 1380.2	47.251 11.404	46.804 11.587
N-432	EU, Spain	Single-lane undivided	42.2	125.8 651.1	37.906 -4.754	38.175 -5.013
G-6	CN, Beijing-Tibet	Multi-lane expressway	192.2	1261.8 2870.5	40.382 115.763	40.555 116.751
G-309	CN, Gansu	Single-lane undivided	82.7	1762.1 2121.9	35.986 105.645	35.983 105.047
G-0613	CN, Gonghe	Multi-lane undivided	47.5	2237.7 2461.8	36.252 101.564	35.986 101.099

5.3.3 Calculation of the Grade vs. Distance Profiles

This section outlines the method for calculating the grade versus distance profiles of the 15 identified highways in Section 5.3.2, using GPS data including longitude, latitude, and altitude.

The road grade calculation is conducted following the steps depicted in Figure 5.3. The initial step is to gather GPS data including latitude and longitude along the route, obtained from Google Earth Pro. Afterward, the acquired latitude and longitude coordinates serve a dual purpose. First, they are used in a digital elevation model (DEM) [229] to determine the elevation of each point along the route. Second, they are employed in the Haversine formula, shown in Eq. (5.2) to calculate the horizontal distance between consecutive points [230]. Finally, the grade is calculated using Eq. (5.3), which incorporates both the elevation data from the DEM and the horizontal distances obtained from the Haversine formula. The detailed sources of the DEM sources utilized in this paper for calculating the selected roads grade profiles are provided in Appendix B.

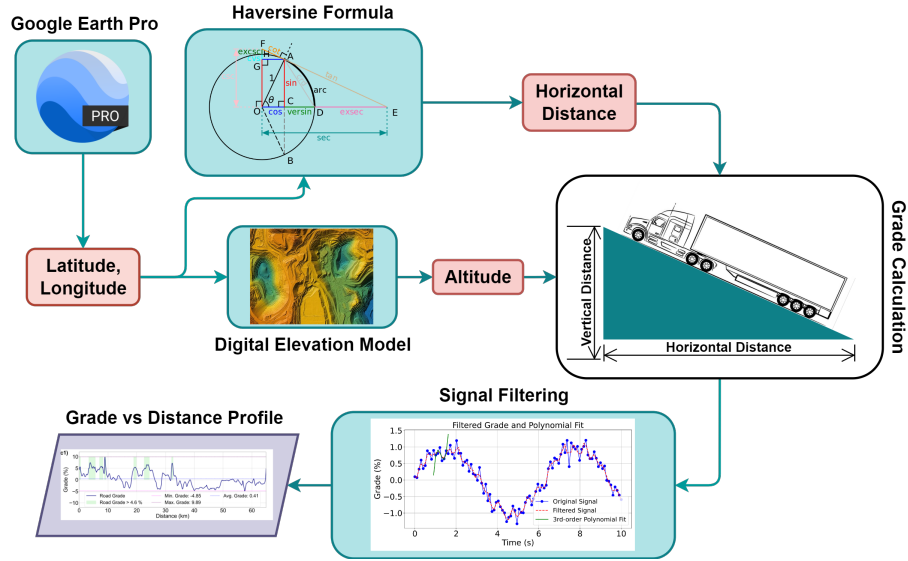


Figure 5.3: Road grade profile calculation process for the identified steep highways in this study.

$$\left\{ \begin{array}{l} a = \sin^2 \left(\frac{\Delta \text{lat}}{2} \right) + \cos(\text{lat}_1) \cos(\text{lat}_2) \sin^2 \left(\frac{\Delta \text{lon}}{2} \right), \end{array} \right. \quad (5.2a)$$

$$\left\{ \begin{array}{l} c = 2 \cdot \text{atan2}(\sqrt{a}, \sqrt{1-a}), \end{array} \right. \quad (5.2b)$$

$$\left\{ \begin{array}{l} d = R \cdot c \end{array} \right. \quad (5.2c)$$

$$\text{Road Grade (\%)} = \left(\frac{\text{Altitude Change}}{\text{Horizontal Distance}} \right) \times 100 \quad (5.3)$$

Where d is the horizontal distance, Δlat represents the difference in latitude between the two points, Δlon is the difference in longitude between the two points, lat_1 and lat_2 are the latitudes of the two points, and R is the Earth radius, considered as 6,371,000 meters [231]. The study in [232] found that calculating distance using Equation (5.2) resulted in an average error of 0.088 meters compared to the Google Maps distance.

5.4 Design of a Class 8 Pre-Transmission Parallel Hybrid Electric Truck

To evaluate the impact of real-world gradeability criteria on the optimal component sizing of Class 8 hybrid electric trucks, a P_2 hybrid configuration is designed in this section. The key powertrain components of the P_2 hybrid configuration are optimally sized using the multi-stage optimization framework previously presented in Chapter 4 (Section 4.3). As the first step, the gradeability performance criterion, as part of the first optimization objective within the component sizing framework, is evaluated over the entire 44.4-km-long Davis Dam road at a constant speed of 56 km/h. Next, a comparison is made between the performance of the designed hybrid model across the entire 44.4-km Davis Dam road and when considering only the initial steepest 19.7 km segment.

The procedure for developing the powertrain configuration of a Class 8 P_2 hybrid electric truck in Autonomie is carried out based on the methodology described in Chapter 3 (Section 3.2.1). The constrained rule-based control strategy for the EMS is also designed following the same procedure as previously described in Chapter 3 (Section 3.2.2). The engine used in the P_2 hybrid model is of the same type as that of the conventional diesel baseline model. However, in the hybrid configuration, the engine can be downsized in terms of rated power and displacement. The critical DOE variables used for optimal powertrain component sizing, along with their variation ranges, are presented in Table 5.3. The variation ranges of these variables are determined based on the preliminary sizing procedure previously described in Chapter 4 (Section 4.3.1) for the P_2 hybrid configuration with a downsized engine.

Following the selection of the 44 km Davis Dam highway for testing the gradeability performance, and upon satisfying the optimization objectives defined in Chapter 4

Table 5.3: Variation range of the critical DOE variables within the optimal component sizing framework for the designed Class 8 pre-transmission parallel (P₂) hybrid electric truck configuration. The hybrid configuration employs a downsized diesel engine of the same type as the conventional baseline model in this study.

Variables (Unit)	Index	Variation Range	Increment Change
Final drive ratio (-)	n_{fd}	2-6	0.1
e-motor rated power (kW)	P_m	115-304	1
Torque coupler ratio (-)	n_{tc}	1-4	0.1
Number of parallel battery modules (-)	$N_{batt,p}$	5-65	1
Engine rated power (kW)	P_e	225-336	1
Engine efficiency threshold for EMS (%)	δ_e	0-0.03	0.001
Low battery SOC threshold for EMS (%)	SOC _L	0.2-0.6	0.01
High battery SOC threshold for EMS (%)	SOC _H	0.5-0.9	0.01

(Section 4.3) using the specified DOE variables, the final component sizing for the P₂ hybrid configuration is derived and presented in Table 5.4.

To approximate the total powertrain penalty weight of the parallel hybrid design under the same cargo load as the conventional baseline, the weight densities of the e-motor, battery, power converter, torque coupler, and transmission are estimated based on Table 4.5 previously presented in Chapter 4. The downsized engine penalty weight, compared to the Cummins X15 engine in the conventional baseline design, is calculated based on the proposed power ranges and displacements shown in Figure 5.4. According to the engine manufacturer’s published data [160, 162, 176], when engine displacement remains constant, rated power can vary within a range depending on tuning and technology, with no significant change in mass. For instance, with a rated power between 301-450 kW, the displacement is assumed to be 15 L, and the engine dry weight matches the 2019 Cummins X15 at 1430 kg (Figure 5.4). When rated power is between 202-300 kW, the displacement drops from 15L to 12L, reducing the dry weight from 1430 kg to 1017 kg. For power ratings between 451-580 kW, displacement increases to 16/17 L, raising the dry weight to 1650 kg.

5.4.1 Performance over the Davis Dam Road

The Class 8 P₂ hybrid configuration was designed with the Davis Dam as the sole criterion for assessing gradeability. In this section, the performance of the parallel hybrid model equipped with a downsized diesel engine, both across the initial 19.7-km segment and the entirety of the Davis Dam road (44.4-km), is evaluated.

The grade versus time profile and performance of the designed parallel hybrid Class 8 configuration on the Davis Dam road are shown in Figure 5.5. The downsized 231

Table 5.4: Component specifications of the conventional diesel and the pre-transmission parallel (P₂) hybrid Class 8 truck configurations. The component sizes in the hybridized configuration are optimized considering the entire 44.4 km Davis Dam route as the gradeability criterion within the optimal component sizing process.

Component/Module		Conventional Diesel Baseline	Parallel Hybrid
Engine	Type	Cummins X15 diesel	Downsized 12 L diesel
	Max. power (kW)	336	231
	Max. torque (N.m)	2330	1602
	Max. speed (rpm)	1900	1900
Fuel system	Type	Single fuel diesel	Single fuel diesel
	Per tank capacity (L)	400 (diesel)	400 (diesel)
	Number of tanks	2	2
Traction motor	Max. power (kW)	-	200 (@1100 rpm)
	Max. torque (N.m)	-	1600
	Max. speed (rpm)	-	5550
High-voltage battery	Type	-	Li-Ion
	Number of parallel modules	-	39
	Number of series cells	-	98
	Single cell capacity (Wh)	-	15.8
	Total capacity * (kWh)	-	60.8
Multi-speed transmission	Type	10-Speed manual	6-Speed automatic
	Gear ratios	Table 2.2 (Chapter 2)	$[n_1, n_2, n_3, n_4, n_5, n_6] =$ $[6.93, 4.1, 2.67, 1.45, 1, 0.82]$
Power converter	Voltage range (V)	-	12 const.
	Power range (kW)	-	0 to 3.1
Torque coupler	Single-gear ratio	-	1.8
Final drive ratio	-	2.85	2.43
Constrained on/off EMS	Engine efficiency threshold (δ_e)	-	0.008
	Low battery SOC threshold (SOC _L)	-	0.30
	High battery SOC threshold (SOC _H)	-	0.55
Powertrain penalty weight compared to the Conv. diesel baseline		-	-88.6 kg

* The total battery capacity spans an SOC range of 0% to 100%, while the usable capacity is limited to the range of 20% to 90% SOC.

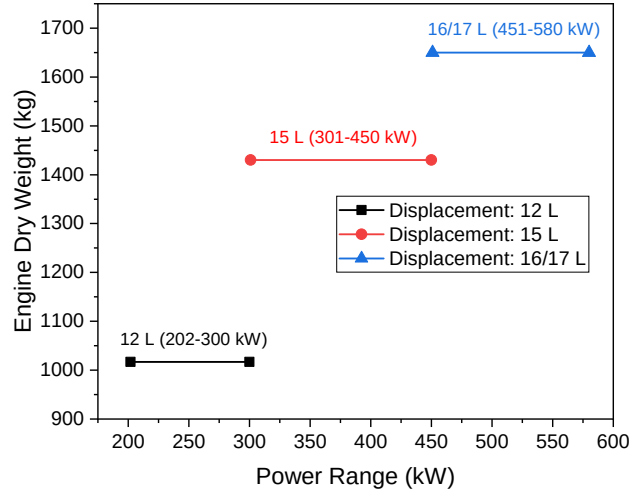


Figure 5.4: Dry weight estimation of diesel engines based on power range and displacement [160, 162, 176].

KW diesel engine in the hybridized model can generate continuous propelling power on a road with a constant grade of 4.8% while carrying 34% of its total cargo load, operating independently of the e-motor and battery. In hybrid configurations where external battery charging is not feasible, the battery SOC should be maintained at a fixed level at the beginning and end of the trip. However, due to the absence of regeneration braking opportunities along the initial 19.7 km of the steepest section of the Davis Dam road, coupled with the use of a downsized engine, the battery SOC cannot be sustained by the end of this segment. Conversely, when assessing the gradeability of the entire 44.4 km route, the presence of downhill segments in the latter part of the road and a less power-demanding profile enable the battery SOC to be recharged to the initial 60% level (Figure 5.5). This highlights the necessity of using the whole cycle profile for gradeability testing of electrified powertrains when the real-world driving route is predefined.

When designing the parallel hybrid electric configuration with a downsized engine, it is crucial to ensure that the propulsion system's performance is comparable to its conventional counterparts. In this study, the conventional diesel baseline model demonstrated the capability to maintain a constant speed of 56 km/h over the most power demanding section of the Davis Dam road when carrying a cargo load equivalent to 34% of its total capacity, as outlined in Section 5.2. Since the SAE J2807 standard does not specify the cargo load that the truck should carry, this paper assumes that both the cargo load (34%) and the constant speed (56 km/h) are maintained for evaluating the gradeability performance of both the conventional and parallel hybrid

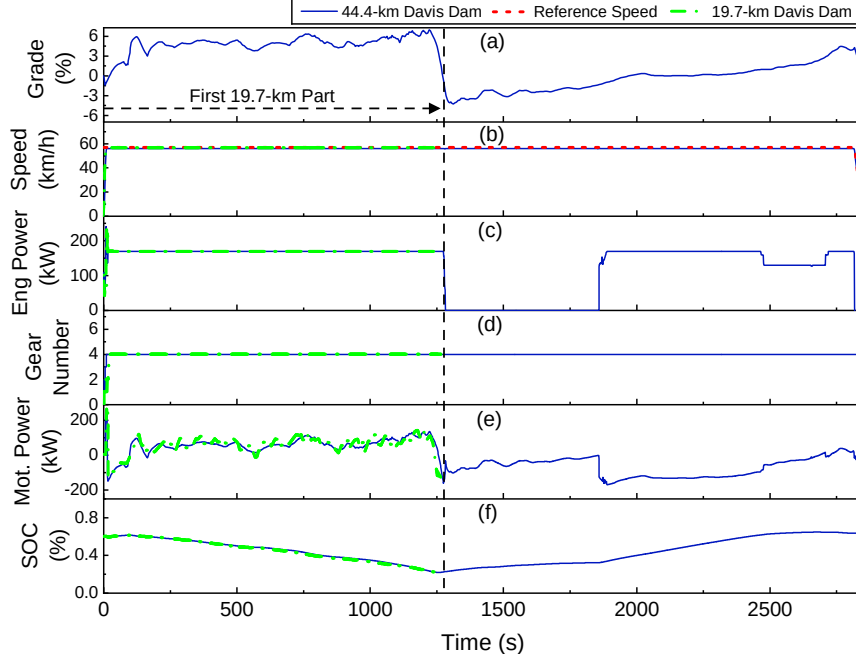


Figure 5.5: Performance of the designed parallel hybrid model over the Davis Dam road at a constant speed of 56 km/h, with 34% cargo load. (a) Road grade, (b) Simulation and reference vehicle speeds, (c) Engine power, (d) Gear number, (e) e-Motor power, and (f) Battery SOC level, versus time.

designs. This assumption is applied not only to the Davis Dam road but also to the newly proposed steep highways identified in Section 5.3.2. The rated power of the power-generating systems is then optimized to follow the grade profile of each identified steep road, which is selected as part of the performance criteria within the optimal component sizing process. Notably, selecting a different conventional baseline model with a different engine rated power necessitates adjusting the assumed cargo load to align with the baseline model's gradeability performance over the Davis Dam road.

5.5 Results and Discussion

This section presents the calculated grade versus distance profiles of identified steep highways, categorized by the proposed steepness classification (Tiers 1, 2, and 3). Subsequently, the optimal powertrain component sizing for the conventional diesel and parallel hybrid Class 8 designs is modified using each proposed steep highway as the gradeability criterion, besides the previously used Davis Dam (Arizona-68) road. Following this, the primary factors contributing to the superior fuel conversion efficiency of the modified parallel hybrid designs compared to their conventional

counterparts over the reference Highway-2 regional-haul drive cycle are examined.

5.5.1 Identified Steep Highways Grade vs. Distance Profiles

Using the latitude and longitude GPS data from Google Earth Pro and the altitude data collected from digital elevation models (DEMs), the grade versus distance profiles of the identified steep highways are computed following the methodology outlined in Section 5.3.3. Figure 5.6 shows the calculated grade versus distance profiles for the 15 identified steep highways in the U.S., Canada, Europe, and China. These profiles can be transformed into grade versus time profiles under the assumption of constant vehicle speed, set at 56 km/h in this study according to the SAE J2807 standard for gradeability testing of Class 8 HD trucks [165]. In Figure 5.6, sections of the route where the grade exceeds 4.8% are highlighted in green color. This number (4.8%) is the average grade over the initial 19.7 km of the Davis Dam road. Supplementary GPS information for each identified steep highways including latitude, longitude, altitude, and grade versus distance profile, is provided in Appendix B and shared publicly via this paper for use by the research community.

The optimal component sizing process of the parallel hybrid design, employing the 44.4 km Davis Dam road as the gradeability criterion, resulted in sizing the engine's rated power to 231 kW. Under 34% cargo load and the constant speed of 56 km/h, the 231 kW engine delivers continuous propulsion power for grades equal to or lower than 4.8%. However, for grades upper than 4.8%, the e-motor supplements the required power demands. Consequently, over the sections with grades exceeding 4.8%, the 231 kW engine in the parallel hybrid design cannot charge the battery, necessitating sufficient energy in the battery to support the e-motor and maintain gradeability performance comparable to the conventional diesel baseline model.

The main characteristics of the grade profiles for the identified steep highways are summarized in Table 5.5. These include the four criteria used in Section 5.3.1 to classify highways into three tiers (Tier 1, 2, and 3) based on the steepness level that considered: 1) the maximum road grade, 2) total percent of downhill distance, 3) the longest continuous distance with a grade $> 4.8\%$, and 4) cumulative elevation gain. The minimum and average grades of each route are shown in the last two columns of Table 5.5. The rows in Table 5.5 highlight identified routes in Tier 1, Tier 2, and Tier 3, marked by red, purple, and green, respectively.

With the exception of the G-6 and G-0613 highways in China, showcasing maximum road grades of 6.2% and 6.9%, respectively, all other identified routes surpass the Davis Dam's 7% maximum road grade (Table 5.5). Highway-14 and Highway-149 in

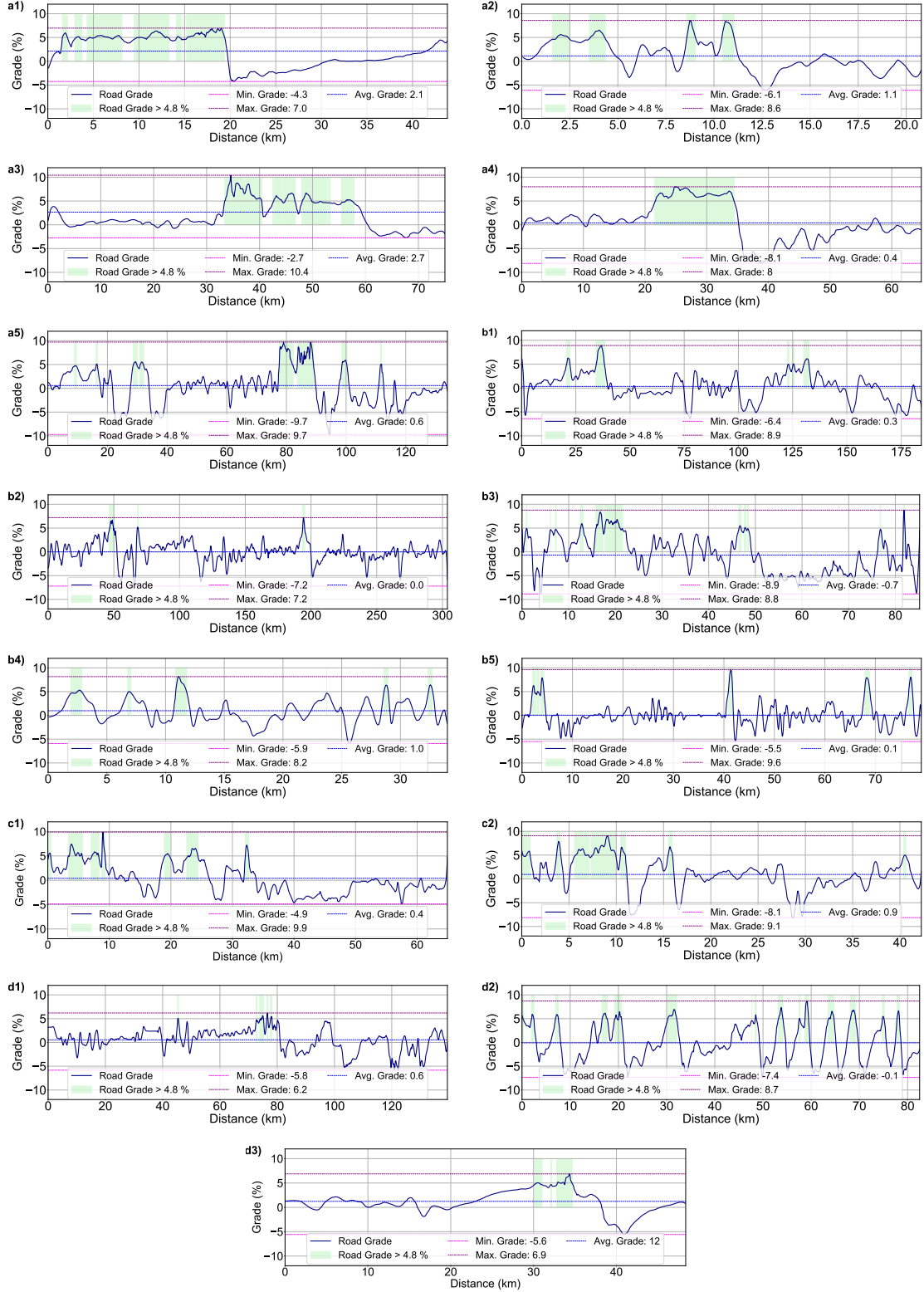


Figure 5.6: Grade versus distance profiles of the identified steep highways in U.S., Canada, Europe, and China. U.S.: a₁) Davis Dam (Arizona-68), a₂) Victory (Route-40), a₃) Highway-14, a₄) Highway-160, a₅) Highway-149; Canada: b₁) Coquihalla, b₂) Summit (High-97), b₃) Okanagan (High-97C), b₄) Laurentienne (Route-73), b₅) Claude-Bechar (Route-2); Europe: c₁) Brenner (A-22), c₂) N-432; China: d₁) G-6, d₂) G-309, d₃) G-0613.

the U.S., and Brenner highway in Europe exhibit the most formidable maximum road grades among the selected highways, reaching 10.4%, 9.7%, and 9.9%, respectively. Concerning the longest continuous distance with a grade exceeding 4.8%, Highway-14, Highway-160, and Highway-149 from the U.S. secure the top positions. From Table 5.5, Summit (High-97), G-0613, Davis Dam (Arizona-68), and Highway-14 exhibit the smallest proportion of downhill section relative to the total drive cycle distance, with the values of 4.9%, 24.4%, 25.9%, and 28.3%, respectively.

5.5.2 Modifying the component sizing to meet the identified steep drive cycles power demands

In Sections 5.2 and 5.4.1, the Davis Dam road was used to evaluate the gradeability of the conventional diesel and parallel hybrid Class 8 regional-haul models. The evaluation was conducted at a constant speed of 56 km/h, as specified by the SAE J2807 standard. This section presents the updated component sizing results for the conventional diesel and parallel hybrid designs, incorporating each newly proposed steep highway as gradeability criteria alongside the Davis Dam. Then, the primary factors lead to the lower fuel consumption and higher average engine brake thermal efficiency (BTE) of the updated parallel hybrid designs, compared to the conventional models, over the Highway-2 drive cycle are discussed.

The conventional diesel X15 baseline model, equipped with an engine rated power of 336 kW, successfully followed the Davis Dam drive cycle at a constant speed of 56 km/h, carrying a cargo load of 8,045 kg (34% cargo load). However, when assessing other proposed steep highways with the maximum gradients larger than the 7% of Davis Dam, an increase in engine rated power is necessary to maintain the same vehicle speed and cargo load. Figure 5.7 shows the modified engine rated power for the conventional diesel configuration when each proposed steep highway is selected, alongside the Davis Dam, as the gradeability criteria. Moreover, Figure 5.7 presents the extra powertrain penalty weight, indicated in green color, due to the increased engine displacement compared to the 15 L Cummins baseline. The engine penalty weight is determined by the rated power range, achievable across three displacement options: 12L, 15L, and 16/17L, as shown in Figure 5.4. The most substantial increase in engine rated power, from 336 kW to 500 kW, occurs for Highway-14 road with a maximum road gradient of 10.4%. This engine size increase leads to an additional powertrain penalty weight of 220 kg compared to the conventional diesel X15 baseline.

The optimal component sizing of the parallel hybrid configuration, over the 44.4-km-long Davis Dam road as the sole gradeability criterion, was detailed in Section

Table 5.5: Main characteristics of the identified steep highways for HD trucks in this study in the U.S., Canada (CA), Europe (EU), and China (CN).

Highway Name (Region)	Steepness Classification (Table 5.1)	Max. Grade (%)	$\frac{\text{Downhill Distance}}{\text{Total Road Distance}} \times 100$	Longest Continuous Distance with Grade > 4.8% (km)	Cumulative Elevation Gain (m)	Min. Grade (%)	Avg. Grade (%)
Davis Dam (U.S.)	Tier 2	7.0	25.9	4.3	1192.2	-4.3	2.1
Victory (U.S.)	Tier 3	8.6	43.3	1.0	508.1	-6.1	1.1
Highway-14 (U.S.)	Tier 1	10.4	28.3	7.0	2123.8	-2.7	2.7
Highway-160 (U.S.)	Tier 1	8.0	44.1	12.9	1461.9	-8.1	0.4
Highway-149 (U.S.)	Tier 1	9.7	44.3	5.5	2830.8	-9.7	0.6
Coquihalla (CA)	Tier 1	8.9	44.9	4.4	3352.0	-6.4	0.3
Summit (CA)	Tier 2	7.2	4.9	3.2	4274.8	-7.2	0.0
Okanagan (CA)	Tier 2	8.8	52.8	3.9	1457.7	-8.9	-0.7
Laurentienne (CA)	Tier 3	8.2	39.7	1.0	686.3	-5.9	1.0
Claude-Bechar (CA)	Tier 3	9.6	52.6	1.0	955.6	-5.5	0.1
Brenner (EU)	Tier 2	9.9	53.8	2.4	1127.4	-4.9	0.4
N-432 (EU)	Tier 2	9.1	34.8	2.7	1058.0	-8.1	0.9
G-6 (CN)	Tier 3	6.2	33.8	1.4	2239.5	-5.8	0.6
G-309 (CN)	Tier 2	8.7	53.7	2.0	2140.0	-7.4	-0.1
G-0613 (CN)	Tier 3	6.9	24.4	1.9	961.5	-5.6	1.2

* Downhill distance refers to the total road distance where the grade is < 0%.

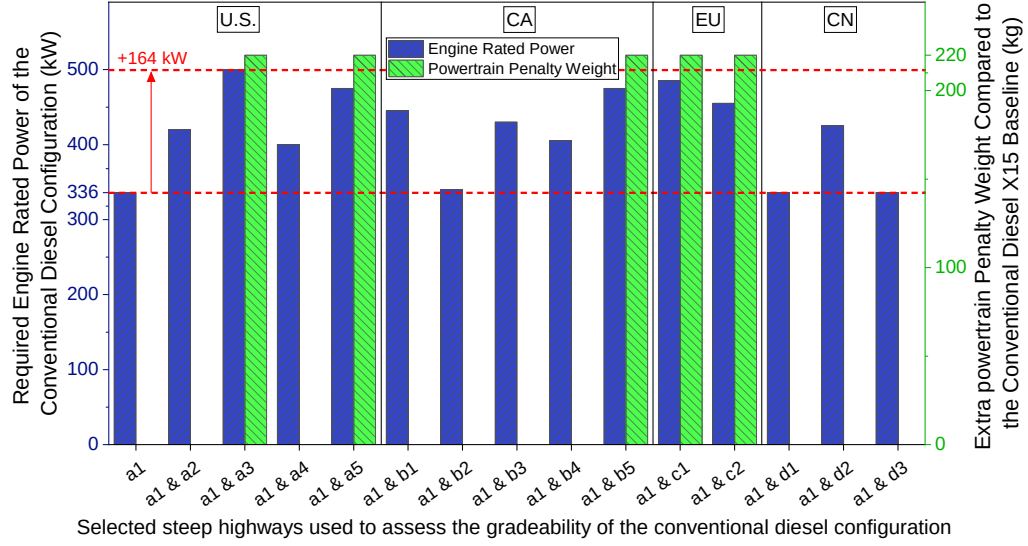


Figure 5.7: Modified engine rated power and extra powertrain penalty weight of the conventional diesel configuration, resulted from selecting each proposed steep highway, alongside the Davis Dam, as criteria for assessing gradeability. The extra powertrain penalty weight is shown compared to the conventional 336 kW baseline diesel configuration. The cargo load in all cases is 34% (8,045 kg) of total cargo capacity. U.S.: a₁) Davis Dam (Arizona–68), a₂) Victory (Route–40), a₃) Highway–14, a₄) Highway–160, a₅) Highway–149; Canada: b₁) Coquihalla, b₂) Summit (High–97), b₃) Okanagan (High–97C), b₄) Laurentienne (Route–73), b₅) Claude–Bechard (Route–2); Europe: c₁) Brenner (A–22), c₂) N–432; China: d₁) G–6, d₂) G–309, d₃) G–0613.

5.4. This design achieved the dual objectives of optimization, resulting in an engine rated power of 231 kW, e-motor rated power of 200 kW, final drive ratio of 2.4, and battery energy capacity of 60.8 kWh, as previously detailed in Table 5.4. However, incorporating each of the identified steep highways alongside the previously selected Davis Dam for gradeability testing necessitates adjustments in optimal component sizing. Figure 5.8 illustrates the refined component sizing outcomes for the parallel hybrid electric design, while incorporating each of the identified steep highways besides the Davis Dam as the gradeability criteria. Each pair of gradeability criteria is substituted as the first optimization objective, as previously outlined in Chapter 4, Figure 4.1.

In the designed parallel hybrid model, the summation of e-motor and engine rated powers correlates directly with the maximum road gradient. As depicted in Figure 5.8, higher maximum gradients in the identified steep drive cycles correspond to higher required summations of e-motor and engine rated powers. For instance, the Highway-14 drive cycle in the U.S. (Figure 5.8. a₃) exhibits the highest maximum road gradient (10.4%) among the selected steep highways in Table 5.5, leading to a maximum summation of e-motor and engine rated power of 545 kW. Following Highway-14, the

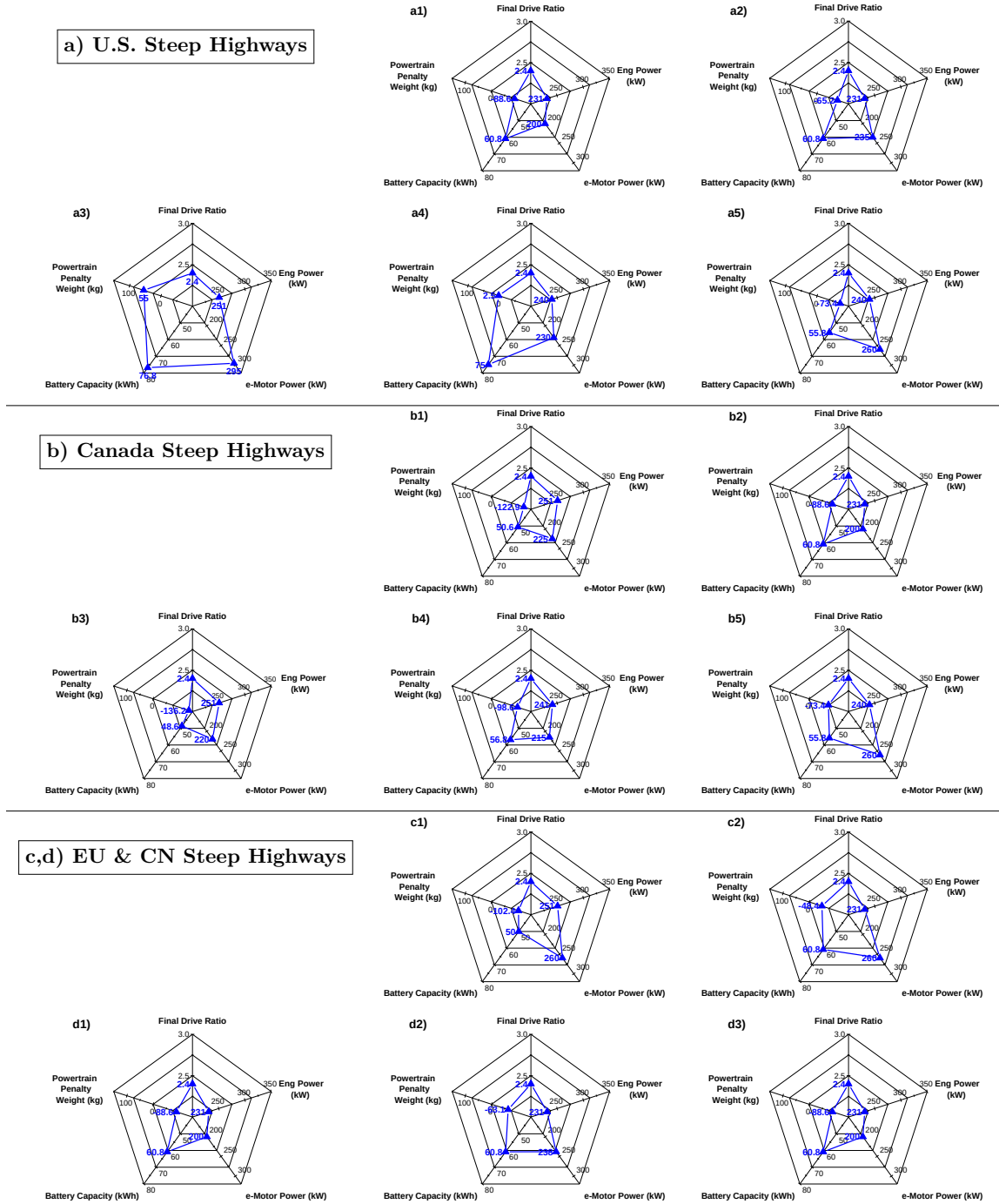


Figure 5.8: Updated component sizing results for the parallel hybrid design, incorporating each of the identified steep highways besides the previously utilized Davis Dam as the gradeability criteria. a1) Davis Dam (Arizona-68), a2) Victory (Route-40), a3) Highway-14, a4) Highway-160, a5) Highway-149, b1) Coquihalla, b2) Summit (High97), b3) Okanagan (High-97C), b4) Laurentienne (Route-73), b5) Claude-Bechard (Route-2), c1) Brenner (A-22), c2) N-432, d1) G-6, d2) G-309, d3) G-0613.

Brenner (A-22) (Figure 5.8. c1), Highway-149 (Figure 5.8. a5), and Claude-Bechard (Route-2) (Figure 5.8. b5) roads display the next highest summations at 511 kW, 500 kW, and 500 kW, respectively. The main reason for the higher required summation of e-motor and engine rated powers in the parallel hybrid design is the limitation of the e-motor in reaching its maximum rated power at speeds less than 1200 rpm. Across all component sizing results in Figure 5.8, the optimal final drive ratio remains 2.4. This choice ensures that the diesel engine operates within the optimum speed range (1100-1400 rpm), to minimize fuel consumption over the Highway-2 drive cycle.

The optimal component sizing results depicted in Figure 5.8 reveal that the highest battery capacities are associated with Highway-14 and Highway-160, amounting to 76.8 kWh and 75 kWh, respectively. For all other cases, the optimal battery capacities either match the 60.8 kWh capacity observed when only the Davis Dam was considered as the gradeability criterion (as shown in Table 5.4), or are lower. In each scenario presented in Figure 5.8, the combination of e-motor and engine rated powers was optimally achieved to ensure that the battery SOC at the start is sustained at the end of the trip. This was achieved through the usage of a constrained rule-based EMS for the P₂ hybrid configuration, as previously detailed in Chapter 3 (Section 3.2.2).

When each of the identified steep highways, along with the Davis Dam, are considered for assessing the gradeability of the parallel hybrid Class 8 design, the total powertrain penalty weight varies. It ranges from -136.2 kg to +55 kg relative to the conventional baseline model (Figure 5.8). In contrast, for the conventional Class 8 design, this value was between 0 kg and +220 kg (Figure 5.7). This observation highlights not only the feasibility of engine downsizing for the parallel hybrid model but also the advantage of a lower total powertrain weight compared to the conventional model.

5.5.3 Performance over a Real-World Regional-Haul Drive Cycle

While it is critical that a vehicle powertrain meets gradeability expectations, in most cases, fuel consumption during regular use has a greater impact on operational cost and net emissions. The effects of the different conventional and parallel hybrid vehicle powertrain sizes (Figure 5.7 and 5.8) over the Highway-2 drive cycle at 23,660 kg payload (full cargo capacity) are compared in Figure 5.9. Different powertrain component sizes were achieved by selecting each proposed steep highway as the gradeability criterion besides the Davis Dam. As would be expected, the higher power engines used to meet the most demanding drive cycles with a conventional powertrain result

in higher fuel consumption (and GHG emissions) over the Highway-2 drive cycle. Surprisingly, the parallel hybrid designs do not show any significant sensitivity to powertrain component sizing. In all cases shown in Figure 5.9, fuel consumption of the parallel hybrid design, in liter per 100 kilometers, is less than the conventional diesel model in 100% cargo load. This fuel consumption reduction is 3.4% when assessing gradeability criteria by combining the Davis Dam highway with Summit (High-97), G-6, or G-0613. The fuel consumption reduction increases to 8.9% when assessing gradeability based on the combination of Davis Dam and Highway-14.

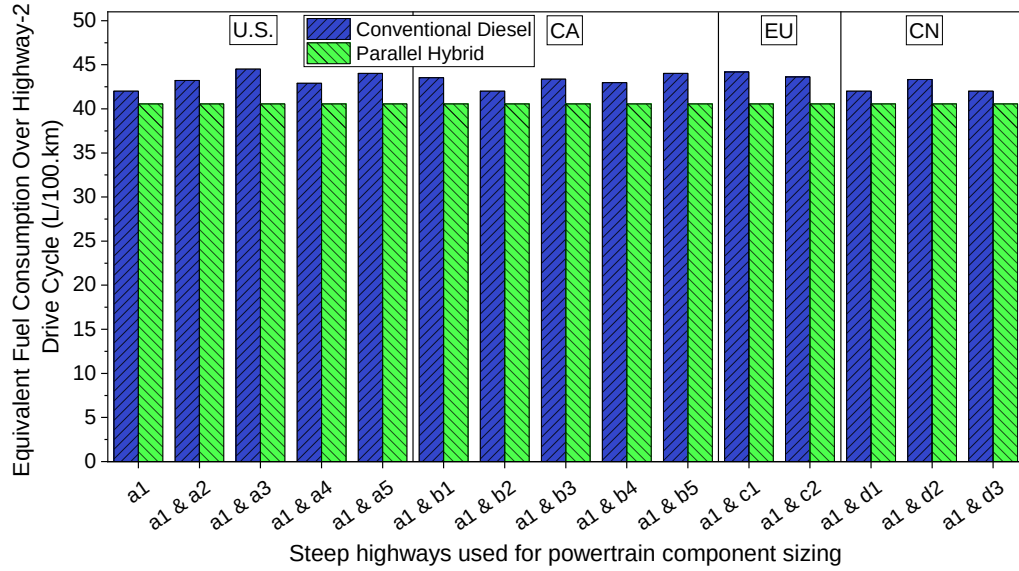


Figure 5.9: Equivalent fuel consumption comparison among the designed parallel hybrid and conventional diesel Class 8 models over the reference Highway-2 regional-haul drive cycle in 100% cargo load. The powertrain component sizing is modified based on the selection of the proposed steep highways, alongside the Davis Dam, as criteria for gradeability assessment. a1) Davis Dam (Arizona-68), a2) Victory (Route-40), a3) Highway-14, a4) Highway-160, a5) Highway-149, b1) Coquihalla, b2) Summit (High97), b3) Okanagan (High-97C), b4) Laurentienne (Route-73), b5) Claude-Bechar (Route-2), c1) Brenner (A-22), c2) N-432, d1) G-6, d2) G-309, d3) G-0613.

Despite the diverse optimal component sizing in Figure 5.9 for parallel hybrid model, fuel consumption remains almost constant at 40.5 liters per 100 kilometers over the Highway-2 drive cycle with a 100% cargo load. In contrast, for the conventional diesel model, fuel consumption not only varies among different combinations of steep highways as gradeability criterion but also increases over the Highway-2 drive cycle with an increase in required engine rated power. The difference between conventional and parallel hybrid models can be explained by contrasting the engine average BTE of the engines across the Highway-2 drive cycle, as shown in Figure 5.10. Across all powertrain component sizing configurations in Figure 5.10, the engine average

BTE in the parallel hybrid design stays at 44.4% and consistently outperforms its conventional diesel counterpart by 0.8% to 2.5% over the Highway-2 drive cycle in 100% cargo load.

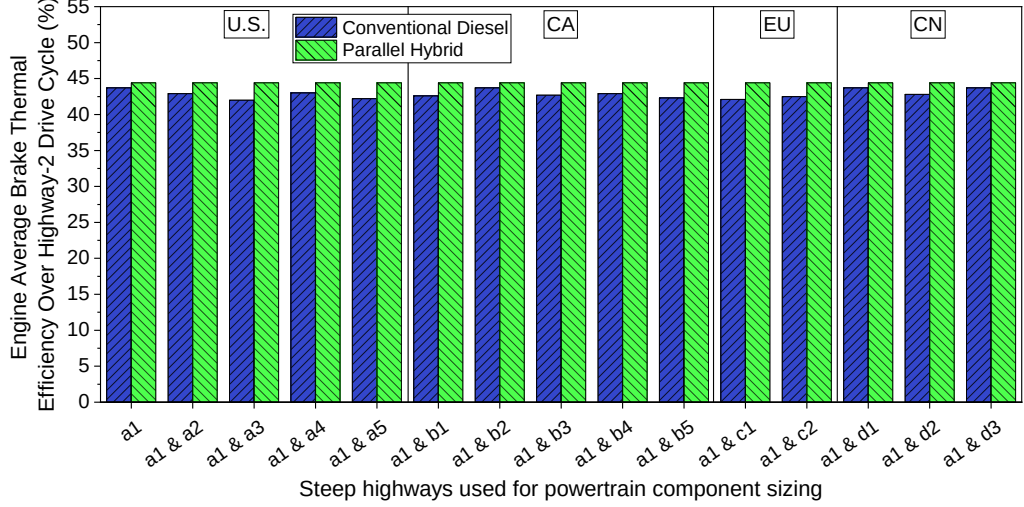


Figure 5.10: Engine average brake thermal efficiency (BTE) comparison among the designed parallel and conventional diesel Class 8 models over the reference Highway-2 regional-haul drive cycle in 100% cargo load. The powertrain component sizing is modified based on the selection of the proposed steep highways, alongside the Davis Dam, as criteria for gradeability assessment. a1) Davis Dam (Arizona-68), a2) Victory (Route-40), a3) Highway-14, a4) Highway-160, a5) Highway-149, b1) Coquihalla, b2) Summit (High97), b3) Okanagan (High-97C), b4) Laurentienne (Route-73), b5) Claude-Bechard (Route-2), c1) Brenner (A-22), c2) N-432, d1) G-6, d2) G-309, d3) G-0613.

For gradeability assessment on steep highways with maximum grades exceeding that of Davis Dam, the engine size of the conventional diesel baseline must be increased to meet the higher power demands. This augmented power, however, surpasses the maximum needed for the Highway-2 reference drive cycle. Consequently, the engine operating points for the conventional design over Highway-2 shift to lower torque regions with reduced BTE values. Figure 5.11 shows the engine operating points for Class 8 conventional diesel and parallel hybrid regional-haul configurations over Highway-2 at full cargo load in three conditions: 1) with Davis Dam and Highway-14, 2) with Davis Dam and Laurentienne (Route-73), and 3) with only Davis Dam as gradeability criteria. Increasing the engine rated power in the conventional diesel design for the mentioned conditions from 336 kW to 500 kW decreases the average BTE over Highway-2 from 43.7% to 41.9%. Table 5.6 indicates that increasing the engine rated power in the conventional design reduces the percentage of engine operation within the highest-efficiency region ($\text{BSFC} \leq 190 \text{ g/kWh}$) over Highway-2. This percentage drops from 73.1% when only Davis Dam is used as the gradeability criterion to 48.7% when both Davis Dam and Highway-14 are selected as the

gradeability criterion.

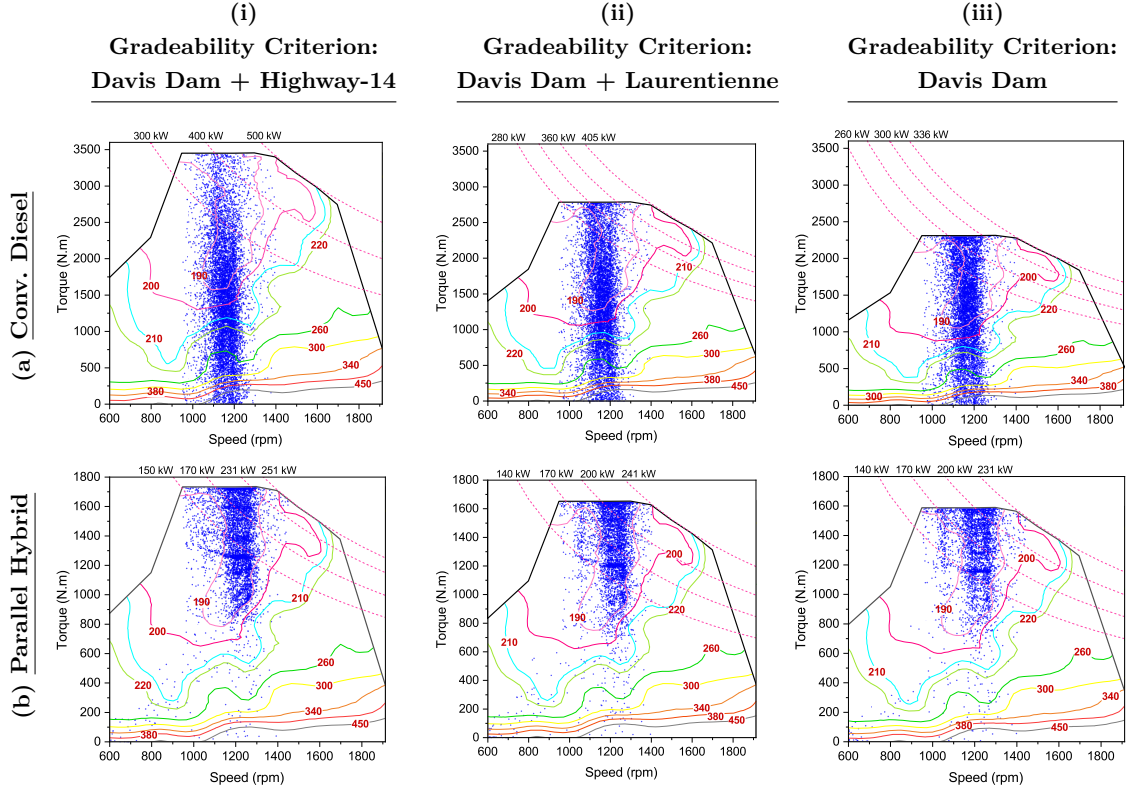


Figure 5.11: Engine operating points for the (a) conventional diesel and (b) the pre-transmission parallel (P_2) hybrid designs on BSFC contour plots over the Highway-2 drive cycle in 100% cargo load. The refined powertrain component sizes were determined by selecting each pair of proposed steep highways as the gradeability criterion: (i) Davis Dam and Highway-14, (ii) Davis Dam and Laurentienne (Route-73), and (iii) Davis Dam alone.

For the parallel hybrid design, as shown in Figure 5.11, the engine mostly operates within the highest-efficiency region ($\text{BSFC} \leq 190 \text{ g/kWh}$), regardless of changes in powertrain component sizing due to different steep roads selected as gradeability criterion. The constrained rule-based control strategy helps keeping the engine off in lower-efficiency regions ($\text{BSFC} \geq 190 \text{ g/kWh}$) and relying on the e-motor and battery to propel the vehicle. Table 5.6 indicates that the percent of engine operation within the highest-efficiency region for the parallel hybrid design over Highway-2 with 100% cargo load is 92.7%, 92.2%, and 91.5% when gradeability criterion is based on Davis Dam alone, Davis Dam and Laurentienne, and Davis Dam and Highway-14 routes, respectively. In contrast, these values for the conventional design are 73.1%, 63.2%, and 48.7%, respectively. Two additional factors contributing to the lower fuel consumption of the parallel hybrid design over Highway-2 with full cargo load are the reduced engine-on duration and the ability to capture regenerative brake energy (Table 5.6).

Table 5.6: Percentage of the time that the engine is ON, percentage of the time that the engine is running at high-efficiency region ($\text{BSFC} \leq 190 \text{ g/kWh}$), and percentage of captured regenerative brake energy per total traction energy at wheels; over Highway-2 in full cargo load. Numbers for the conventional and parallel hybrid designs are shown for three cases when different gradeability criteria are selected: Davis Dam and Highway-14, Davis Dam and Laurentienne (Route-73), and Davis Dam alone.

Operation Over Highway-2 in Full Cargo Load	Gradeability Criteria					
	Davis Dam + Highway-14		Davis Dam + Laurentienne		Davis Dam alone	
	Conv.	Parallel	Conv.	Parallel	Conv.	Parallel
ICE ON percentage (%)	100	91.5	100	92.7	100	94.2
Percent of operation in high-efficiency zone ($\text{BSFC} \leq 190 \text{ g/kWh}$) (%)	48.7	91.5	63.2	92.2	73.1	92.7
Regen energy per total traction energy at wheels (%)	0	7.6	0	7.5	0	7.4

Increasing the rated power of e-motor enhances its capacity to capture regenerative brake energy. Thus, when the e-motor size is raised from 200 kW (with Davis Dam as the sole gradeability criterion) to 295 kW (with Davis Dam and Highway-14 as criteria), the percentage of regenerative energy relative to total traction energy at the wheels over Highway-2 at full cargo load slightly rises from 7.4% to 7.6% (Table 5.6).

5.6 Summary of Chapter

This chapter reported on the significant effect of gradeability criteria on optimal powertrain component sizing in hybridized and conventional Class 8 HD trucks. It emphasized the limitations of the existing gradeability criterion, specified by the SAE J2807 standard for conventional HD trucks, when applied to the gradeability assessment of hybrid electric HD trucks. To address this, fifteen real-world steep highways suitable for HD truck operations were identified across the U.S., Canada, Europe, and China. The effect of each of these highways, along with the previously used Davis Dam route (specified by the SAE standard), as the gradeability criterion was evaluated on the optimal component sizing of Class 8 conventional and parallel hybrid truck configurations. Selecting different pairs of these steep highways as the

gradeability criterion resulted in adjustments to powertrain components, including engine rated power, e-motor rated power, battery capacity, and final drive ratio. The net effect of these variations in component sizing was analyzed for fuel consumption in conventional diesel and parallel hybrid designs over a real-world regional-haul drive cycle (Highway-2). The key findings from this chapter are:

- When evaluating gradeability of Class 8 hybrid electric trucks, which incorporate multiple power sources such as the ICE and the battery, it is essential to consider the entire planned route incorporating the real-world grade profile, especially when aiming for engine downsizing. Therefore, in addition to the maximum road grade, three other deterministic factors are considered: i) the proportion of downhill sections for regenerative braking, ii) the duration of continuous driving on steep sections where the road load demands exceed the ICE rated power, iii) total elevation increase encountered over a route, i.e., the cumulative elevation gain.
- The fifteen identified steep highways are categorized by steepness into Tier 1, 2, and 3 based on the four specified criteria. Tier 1 comprises the most power-demanding roads, including Highway-14 in Wyoming, U.S., Highway-160 and Highway-149 in Colorado, U.S., and the Coquihalla Highway in British Columbia, Canada. In this classification, the Davis Dam road is placed in Tier 2, indicating that it is less power-demanding than the Tier 1 roads. As the steepness level increases, the combined rated power of the e-motor and engine in the parallel hybrid design also increases. For the 15 identified steep highways the data including latitude, longitude, altitude, and grade versus distance profile were tabulated and made publicly available in the Appendix B of this thesis.
- The unique grade profile of each steep road requires tailored component sizing adjustments for the conventional diesel and parallel hybrid truck models. For fuel consumption on daily routes, particularly Highway-2, the parallel hybrid configuration maintains a steady rate of 40.5 L/100 km regardless of component sizing changes. Its hybrid control strategy optimizes efficiency by keeping engine operation within the high-efficiency zone ($\text{BSFC} \leq 190 \text{ g/kWh}$) and relying on the e-motor and battery in less efficient zones. This strategy also reduces engine-on duration and captures regenerative braking energy, resulting in 3.4% to 8.9% greater fuel efficiency than that of conventional diesel counterparts.
- When more power-demanding steep routes are selected for gradeability assess-

ment of the conventional design, engine rated power increases from 336 kW (using only Davis Dam as the criterion) to 500 kW (using Davis Dam and Highway-14). This increase in the engine rated power leads to increasing the powertrain penalty weight from 0 to 220 kg. For the parallel hybrid electric design, with a downsized diesel engine, this penalty ranges from -136.2 to 55 kg. This demonstrates that the parallel hybrid design maintains a lighter total powertrain penalty weight than the conventional design while achieving the same performance criteria.

- For assessing the gradeability of Class 8 conventional trucks, due to the sole reliance on the ICE for power generation, the engine rated power needs to be adjusted to only fulfill the required road power demands at the steepest section of the road with the maximum grade. This needs to be considered if the Class 8 truck is intended for operation in mountainous regions. While, for Class 8 hybridized trucks, an optimal component sizing process is essential to ensure that powertrain elements such as the ICE, e-motor, battery, and final drive are appropriately sized to meet the maximum road demanding power and maximize fuel conversion efficiency over a representative highway driving route.

For gradeability assessment within the optimal component sizing framework in the next chapters, for designing both conventional and hybridized Class 8 truck models equipped with lower-carbon-fueled engines, the entire 44.4 km Davis Dam route will be used. This includes employing the real-world varying road grade profile obtained in this chapter, evaluated at a constant speed of 56 km/h, consistent with the SAE J2807 specification for HD truck applications.

Chapter 6

Performance Assessment of Class 8 Hybrid Electric Trucks Equipped with Low-Carbon Natural Gas Engines¹

The effect of combining full powertrain hybridization with a downsized diesel engine on reducing fuel consumption and GHG emissions, while maintaining propulsion system performance comparable to conventional Class 8 trucks, was studied in the previous chapter. In the present chapter, the integrated effect of powertrain hybridization with low-carbon natural gas (NG)-CI engines is investigated to further reduce tailpipe GHG (including CO₂ and CH₄) and NO_x emissions in Class 8 truck applications. The NG-CI engines considered are categorized as a dual-fuel premixed NG–diesel engine and technology-enhanced variants of the diesel-pilot HPDI NG engine, in both downsized and full-size configurations. The incorporated engine technologies and modifications for the HPDI NG engine include: increasing compression ratio (CR), advancing start of injection (SOI), and integrating a turbo-compound (TC) system. These technologies are implemented to improve engine efficiency, particularly at mid-to-high torque levels, where the engine in hybrid configurations operates predominantly. The fuel consumption and emission maps of the different NG-CI engine variants are also presented.

The three hybrid Class 8 truck powertrains previously developed in Chapter 3 are optimized in the present chapter for each proposed variant of NG-CI engine. The hybrid configurations include the pre-transmission parallel (P₂), the power-split, and the series plug-in hybrid configuration. The diesel-equivalent fuel consumption and

¹This chapter is partially based on the published journal papers in [2, 4] and the journal paper currently under review in [5].

emissions of the optimized hybrid electric truck configurations are then compared with those of the conventional diesel and conventional NG baseline models in this thesis. The comparison is carried out over both regional-haul (highway) and urban drive cycles under different cargo load conditions. Finally, a comparative evaluation is performed between the designed Class 8 conventional and hybrid electric truck models, focusing on engine operating points, component-level energy losses, and net powertrain energy efficiencies.

6.1 Introduction and Background

Hybridization enables the ICE to either remain off or operate more frequently within its high-efficiency region, which is typically associated with mid-to-high torque regions ($T > 0.5 T_{\max}$) in CI engines [1]. Instead, the electrical driveline can propel the vehicle when engine operation would fall outside this high-efficiency region, which commonly occurs during low power demand conditions (e.g., urban driving) or transient high power demands such as rapid acceleration [121]. This results in a higher cycle-average engine brake thermal efficiency (BTE), thereby improving the overall powertrain energy efficiency compared to non-hybrid vehicles [19]. In addition, regenerative braking enables recovery of a portion of the vehicle’s kinetic energy during deceleration, further contributing to overall energy efficiency. However, under highway cruising conditions typical of Class 8 HD trucks, regenerative energy recovery is limited, and the efficiency gains rely primarily on improved engine operation within the high-efficiency region.

The selection of the engine technology plays a key role in determining the effectiveness of hybridization in Class 8 HD trucks. When advanced NG-CI engines, such as diesel-pilot HPDI NG engines with efficiency levels comparable to or higher than diesel, are integrated into hybrid powertrains, both fuel consumption and CO₂ emissions can be reduced relative to conventional diesel trucks [43]. In contrast, the use of lower-efficiency NG engines, such as SI-based configurations, may limit these benefits due to their lower overall efficiency compared to diesel [127, 146]. Under highway-dominant operation, where regenerative braking opportunities are limited, this efficiency gap can result in reduced fuel savings or even increased fuel consumption compared to conventional diesel counterparts.

Incorporating advanced technologies into HPDI NG engines that enhance BTE, particularly at mid-to-high torque levels, leads to greater energy efficiency benefits in hybrid electric HD truck applications. A summary of the potential advancements and

modifications for conventional diesel engines was previously presented in Chapter 1, Table 1.3. Studies in [59, 233, 234] reported that increasing the compression ratio (CR) can yield a theoretical BTE improvement of 2–15% at mid-to-high torque levels for NG-CI engines, primarily due to higher achievable peak cylinder pressures. Studies in [43, 235, 236] reported that advancing the injection timing by 2–4 crank angle degrees can lead to a 1–3% improvement in BTE at upper than mid-torque levels for NG-CI engines, primarily due to improved combustion phasing and increased premixed heat release near top dead center. Additionally, integrating a series turbo-compound (TC) system has been reported to increase BTE by 1.8–4% [1, 43, 237] at mid-to-high torque levels, primarily due to the recovery of exhaust waste heat and its conversion into useful mechanical power. Other advancements, such as organic Rankine cycle (ORC), cylinder deactivation, and advanced turbocharging, have also been shown to provide BTE improvements in the range of 2–4.5% [1, 49, 236] at higher torque regions.

The corresponding fuel consumption and emissions maps for the technology-enhanced NG-CI engine variants in this chapter were developed and provided by the collaborative research team at Simon Fraser University. Further details regarding the selection of these technologies and their modeling approaches are provided in [43].

6.2 Baseline Natural Gas Engines for Hybridization

To develop Class 8 hybrid electric truck configurations equipped with low-carbon NG-CI engines with the same rated power as the diesel engine in the conventional baseline configuration, the 13 L HPDI NG engine previously introduced in Chapter 2 (Section 2.3) is used. To further enhance the fuel economy benefits of the hybridized configurations, several variants of the baseline HPDI NG engine are considered by modifying key engine design parameters or incorporating additional technologies.

The key specifications of the baseline 13 L HPDI NG engine, along with its fuel consumption, CO_2 , and CH_4 emissions maps, were previously presented in Chapter 2, Section 2.3.1. The engine-out brake specific NO_x emissions map of the baseline HPDI NG engine is shown in Figure 6.1.

6.2.1 Engine Technologies to Improve Fuel Consumption

As shown by the engine operating plots in Figure 5.11 of Chapter 5, hybridization shifts the engine operating regime toward high-efficiency torque regions (i.e.,

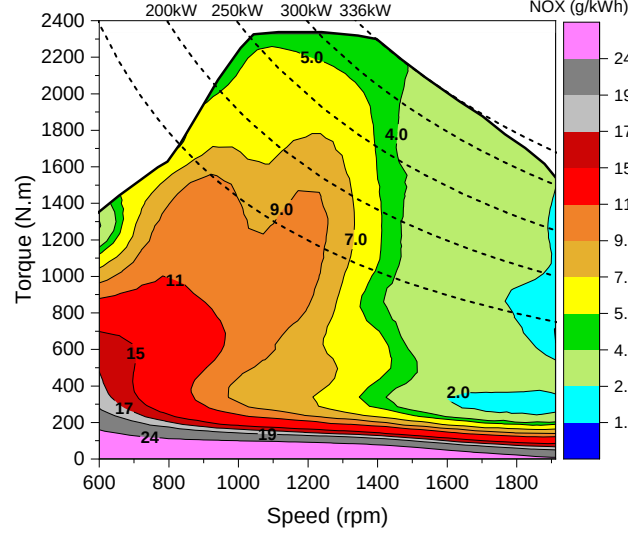


Figure 6.1: Engine-out brake specific NO_x emissions map of the 13 L diesel-pilot HPDI NG engine in this study.

$T > 0.5 T_{\max}$), corresponding to lower BSFC values. The electrical driveline, including the battery and e-motor(s), propels the vehicle during low power demand conditions such as urban driving and allows the engine to operate more frequently within its high-efficiency region at higher torques. This section proposes modified and technologically enhanced variants of the baseline HPDI NG engine to improve efficiency, particularly within the high-torque region. The proposed modifications include increasing the CR, advancing the start of injection (SOI), and incorporating a series TC system as a waste heat recovery technology.

1- Basline HPDI NG engine with increased compression ratio:

The first modified engine parameter is the CR, which is increased from 17 for the baseline HPDI NG engine to 19 [43]. The diesel-equivalent BSFC map of the high-CR 13 L HPDI NG engine is shown in Figure 6.2(a). As shown in Figure 6.2(b), increasing the CR reduces fuel consumption by up to 1% in the mid-to-high torque regions (i.e., $T > 0.5 T_{\max}$), while slightly increasing fuel consumption in the low-torque region relative to the baseline HPDI NG engine. The reduction in fuel consumption at higher torque levels is attributed to improved thermodynamic efficiency; however, the increase in fuel consumption at low torque is primarily due to higher friction losses associated with increased in-cylinder pressure [218].

The BSFC maps of the technology-enhanced 13 L HPDI NG engines were derived from the previously validated GT-SUITE one-dimensional HPDI NG engine model. In the referenced engine-level study [43], the baseline HPDI NG model was first cali-

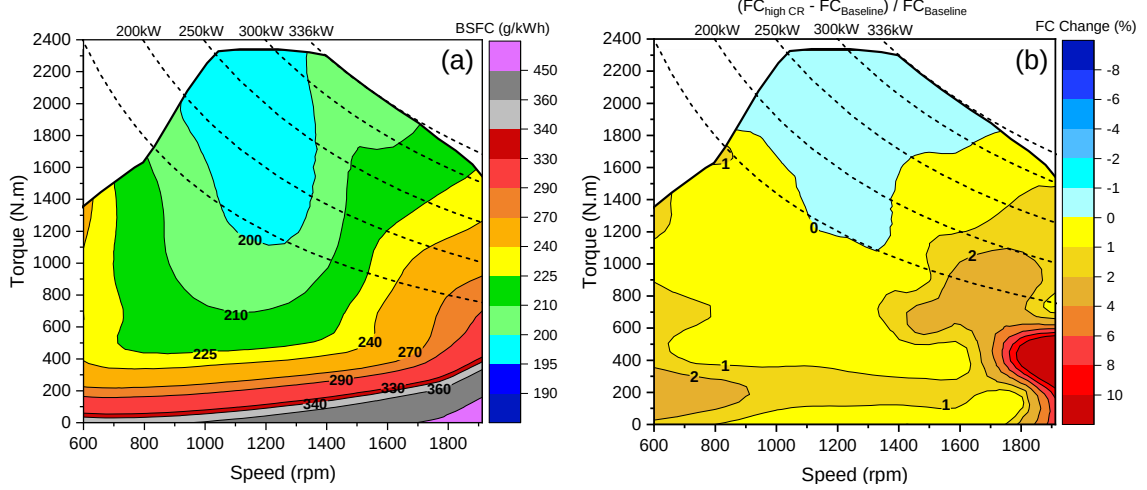


Figure 6.2: (a) Diesel-equivalent brake specific fuel consumption (BSFC) map of the 13 L HPDI NG engine with an increased compression ratio ($CR = 19$), and (b) relative percentage change in fuel consumption, measured in g/s, compared to the baseline HPDI NG engine ($CR = 17$).

brated and verified against experimental engine data, and then the CR, SOI, and TC cases were evaluated over the engine speed–load domain to generate diesel-equivalent BSFC maps. For the high-CR case, a peak cylinder pressure (PCP) threshold of ~ 250 bar was used as the upper hardware limit, and the $CR = 19$ case maintained PCP and pre-turbine exhaust gas temperature within the safe operating limits. Higher CR values were not used because they exceeded the PCP hardware limit. Therefore, only the BSFC map was modified for the high-CR case, while the engine-out NO_x and CH_4 emission maps were assumed unchanged relative to the baseline engine [185].

2- Basline HPDI NG engine with advanced start of injection:

The second modified engine parameter involves advancing the injection timing by 2 degrees of crank angle during full-load operation within the speed range of 1000–1600 rpm [43]. The diesel-equivalent BSFC map of the 13 L HPDI NG engine with advanced SOI is shown in Figure 6.3(a). As shown in Figure 6.3(b), advancing the SOI reduces fuel consumption in the high-torque region by up to 1.2%; however, fuel consumption slightly increases in the mid- and low-torque regions relative to the baseline engine. The efficiency improvements within the speed range of 1000–1400 rpm are particularly important due to the frequent engine operation of Class 8 HD trucks over highway driving conditions.

The SOI advancement was also constrained by the PCP limit. In the referenced engine-level study [43], advancing SOI at the A100 operating point increased PCP

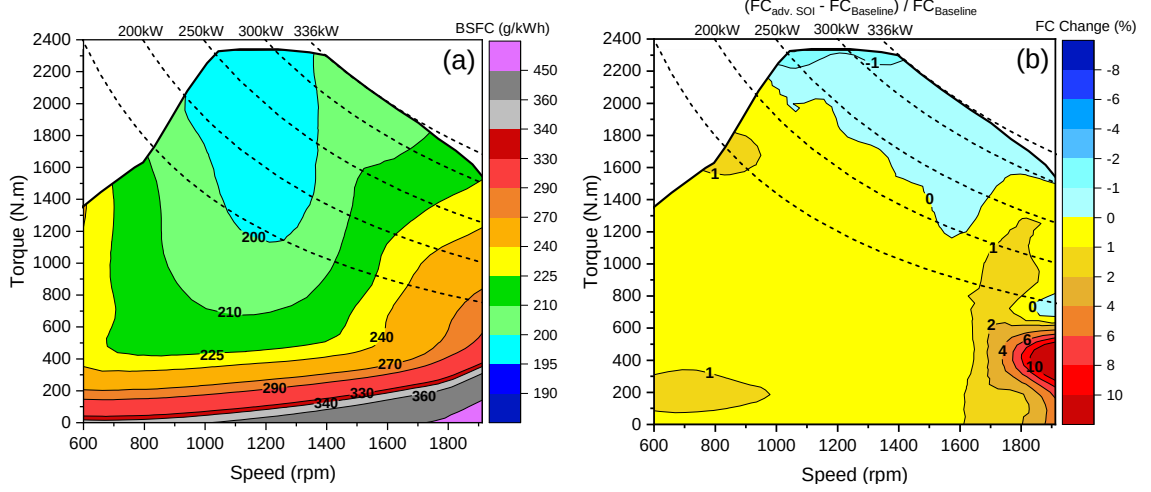


Figure 6.3: (a) Diesel-equivalent brake specific fuel consumption (BSFC) map of the 13 L HPDI NG engine with advanced start of injection (SOI), and (b) relative percentage change in fuel consumption, measured in g/s, compared to the baseline HPDI NG engine.

from 226 bar to 236 bar, corresponding to a 4.5% increase, while remaining below the allowable hardware limit. SOI advancement was limited to avoid exceeding PCP limits, that 1.5°CA SOI advancement increased PCP from 226 to 236 bar at A100, and that $CR = 19$ remained within PCP and EGT limits while CR values higher than 19 exceeded the PCP hardware limit. The 250 bar PCP threshold is also identified as the future HD engine reference threshold used in the downsizing analysis.

Advancing the SOI results in substantially higher engine-out NO_x emissions compared to the baseline injection timing, as shown in Figure 6.4. The NO_x emissions increase by up to 4% in the low-to-mid torque region and rise further to as much as 26.4% at higher loads. As reported in [185], advancing the SOI by 2 degrees of crank angle within the 1000–1600 rpm speed range in HPDI NG engines does not affect engine-out CH_4 emissions.

3- Basline HPDI NG engine with turbo-compound system:

To enhance the efficiency of the baseline HPDI NG engine, particularly in the high-torque region, a series TC system was designed and integrated into the baseline engine as an additional technology [43]. The diesel-equivalent BSFC map of the turbo-compounded 13 L HPDI NG engine is shown in Figure 6.5(a). The technology-enhanced HPDI NG engine achieves up to 3.5% reduction in fuel consumption in the high-torque region and up to 6% at peak power, as shown in Figure 6.5(b). The engine-out NO_x and CH_4 emissions remain unchanged compared to the baseline engine [238].

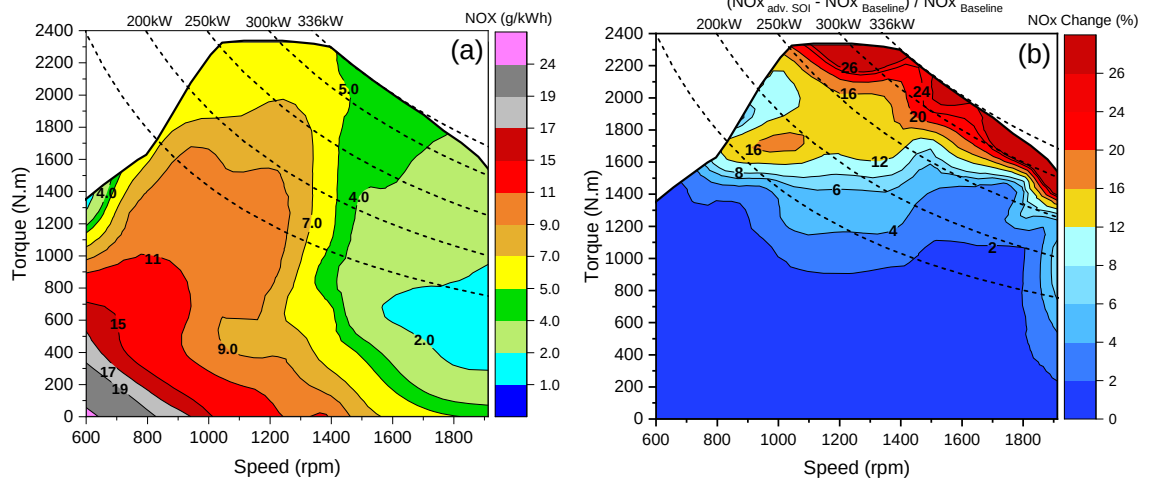


Figure 6.4: (a) Brake specific NO_x emissions map of the 13 L HPDI NG engine with advanced start of injection, and (b) relative percentage change in NO_x emissions, measure in g/kWh, compared to the baseline HPDI NG engine.

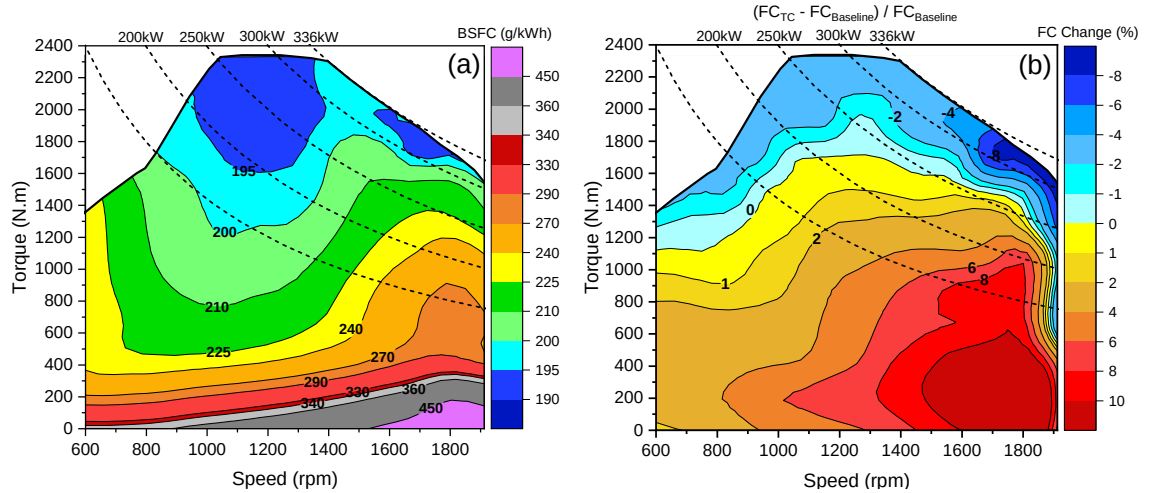


Figure 6.5: (a) Diesel-equivalent brake specific fuel consumption (BSFC) map of the 13 L HPDI NG engine with the turbo-compound system, and (b) relative percentage change in fuel consumption, measured in g/s, compared to the baseline HPDI NG engine.

The greater efficiency improvement of the TC system compared with CR and SOI modification is attributed to direct recovery of exhaust enthalpy rather than only modifying the in-cylinder combustion process. In the referenced engine-level study [43], a series TC system was matched by varying the main-turbine and power-turbine flow capacities and the gear ratio between the power turbine and the crankshaft. The final TC configuration was selected to improve BSFC in the primary operating range of the engine under hybrid operation. At mid-to-high torque regions, the exhaust temperature and pressure are sufficiently high for the power turbine to recover useful shaft power and contribute to the total engine brake power. Although the TC system increases exhaust back-pressure and pumping losses, the recovered turbine power exceeds this penalty at high loads, leading to lower BSFC. At low loads, however, the available exhaust energy is insufficient to compensate for the increased pumping losses, which explains why the TC benefit diminishes or can reverse under low power-demanding conditions.

4- Baseline HPDI NG engine with turbo-compound system and increased compression ratio:

Among the considered variants of the 13 L HPDI NG engine, those with increased CR or a TC system improve engine efficiency in the mid-to-high torque regions without increasing NO_x and CH_4 emissions compared to the baseline engine. Studies in [43, 237] indicates that combining a TC system with a high CR improves efficiency in the mid-to-high torque region more than applying each technology individually. Fuel consumption is reduced by up to 4.2% in the high-torque region ($T > 0.5 T_{\max}$), while it increases by up to 8% within mid-to-low torque regions compared to the baseline HPDI NG engine, as shown in Figure 6.6(b).

Overall, among the investigated 13 L HPDI NG engine variants, the configuration combining a TC system with an increased CR offers the most promising efficiency improvements in the mid-to-high torque region. This aligns well with the operating regime of hybridized powertrains, where the engine is more frequently loaded at higher torque levels. Despite efficiency penalties at lower loads, the combined TC and high CR configuration provides the most favorable trade-off between fuel consumption and emissions. Therefore, it is identified as the most suitable candidate for integration into hybrid electric Class 8 truck configurations.

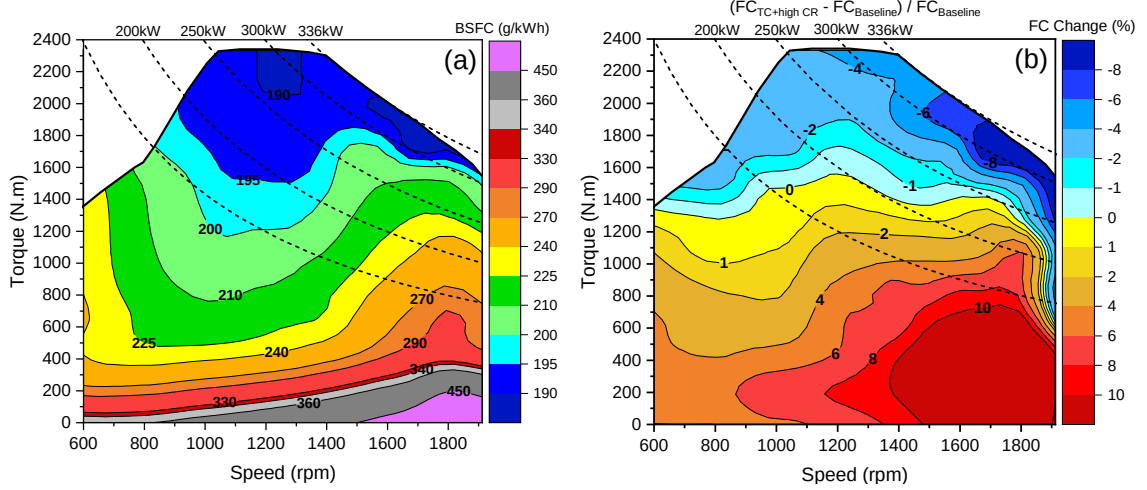


Figure 6.6: (a) Diesel-equivalent brake specific fuel consumption (BSFC) map of the 13 L HPDI NG engine with the turbo-compound system and increased compression ratio ($CR=19$), and (b) relative percentage change in fuel consumption, measured in g/s, compared to the baseline HPDI NG engine.

6.3 Downsized Natural Gas Engines for Hybridization

In this section, following the approach of employing downsized engines for Class 8 hybridized truck configurations, two NG-CI engines with a rated power of 231 kW are introduced. The first engine is a downsized 9 L version of the baseline 13 L HPDI NG engine, developed in [239] by reducing the number of cylinders from six to four while maintaining the per-cylinder displacement and scaling the air-handling system. The second engine is a 6.7 L diesel-pilot premixed (micro-pilot) NG engine, designed in [240] by converting a Cummins ISB diesel engine. The key specifications of these two engines are summarized in Table 6.1. The micro-pilot engine injects NG through the intake manifold as the main fuel, while a small amount of diesel (2–5% of the total fuel energy) is directly injected into the cylinder as a pilot for ignition [241].

Table 6.1: Key specifications of the downsized low-carbon natural gas compression ignition engines in this study [239, 240].

Parameter	9 L HPDI NG	6.7 L Micro-pilot NG
Engine configuration	Inline four-cylinder	Inline six-cylinder
Max. power, kW	231	231
Max. BMEP ¹ , bar	22.8 20.7	
Max. torque, N.m	1,602 @1378 rpm	1,236 @1220 rpm
Max. speed, rpm	1,912	2,000
Fuel injection system	Direct injection, dual concentric needle HPDI (300 bar NG injection pressure)	Port fuel injection of NG at pressure of 3-10 bar
Displacement per cylinder, L	2.2	1.1
Compression ratio (CR)	16.8:1	15:1
Technologies	FGWT/C ²	VGT/C ³ + EGR ⁴ + CAC ⁵
Aftertreatment system	DOC + DPF + SCR	TWC ⁶
Dry weight, kg	866	730

¹ Brake mean effective pressure ² Fixed-geometry wastegate turbocharger ³ Variable geometry turbocharger
⁴ Exhaust gas recirculation system ⁵ Charge air cooler ⁶ Three-way catalyst

As previously concluded in Chapter 5, a Class 8 parallel hybrid truck configuration equipped with a 231 kW downsized engine, along with an e-motor and battery, maintains propulsion system performance comparable to those of the conventional diesel baseline model with a 336 kW engine. Accordingly, the downsized NG engines considered in this section are specified with a rated power of 231 kW to meet the performance requirements of the hybridized powertrain. To compute the engine fuel mass flow rate across different operating speeds and torques, the diesel-equivalent BSFC maps of the proposed downsized engines are depicted in Figure 6.7.

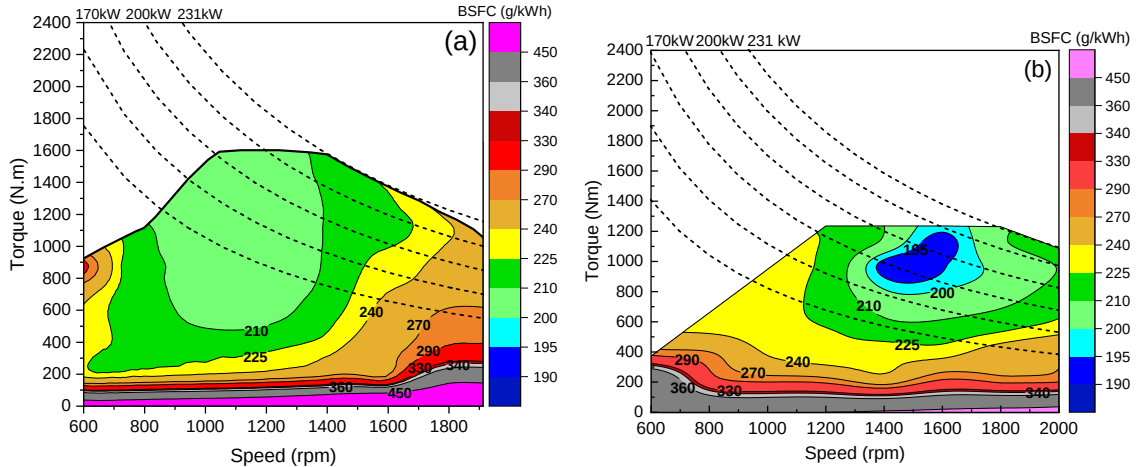


Figure 6.7: Diesel-equivalent brake-specific fuel consumption (BSFC) maps of: (a) the 9 L HPDI NG engine, and (b) the 6.7 L micro-pilot NG engine, both with a rated power of 231 kW.

The fuel consumption map of the downsized 9 L HPDI NG engine was obtained

from the same validated GT-SUITE one-dimensional HPDI NG engine modeling framework used for the 13 L baseline engine. The 9 L engine was developed in [43] by reducing the number of cylinders while maintaining the baseline per-cylinder displacement and carrying over the validated power-cylinder combustion model. The target peak torque was reduced by 30% to match the hybrid powertrain sizing requirement, and the air-handling system was scaled using dimensional analysis to accommodate the lower air flow rate. For each engine speed-load operating point, the NG injection quantity was adjusted using the verified control approach to meet the target torque while respecting the equivalence-ratio constraint.

The downsized 9 L HPDI NG engine offers slight BSFC improvements in the low-torque region (i.e., $T < 0.4 T_{\max}$) relative to the baseline 13 L engine, mainly due to lower friction losses. However, it exhibits higher BSFC values in the mid-to-high torque regions (i.e., $T > 0.5 T_{\max}$), where hybridized configurations operate more frequently. This high-load penalty is attributed to the higher BMEP required from the downsized four-cylinder engine at comparable torque levels, along with increased exhaust enthalpy and pumping-related losses relative to the full-size six-cylinder engine. Therefore, although downsizing can reduce engine mass and improve low-load efficiency, the 9 L HPDI NG engine is less favorable for hybrid operation unless combined with efficiency-enhancing technologies such as TC and high CR.

The engine-out NO_x and CH_4 emissions maps of the downsized HPDI NG engine were previously calculated in [183] and are shown in Figure 6.8. The downsized engine exhibits engine-out emissions comparable to those of the baseline engine at the same load (BMEP) levels. Both engines show approximately 5 g/kWh NO_x and 0.1 g/kWh CH_4 emissions at higher loads. In addition, the downsized engine demonstrates similar contour patterns and gradient distributions in the emissions maps relative to the 13 L baseline engine.

6.3.1 Engine Technologies to Improve Fuel Consumption

To enhance the efficiency of the downsized HPDI NG engine, particularly in the mid-to-high torque region, which represents the sweet spot for hybridization, similar technologies and modifications to those applied to the 13 L engine were evaluated in [43, 239]. The results show that, similar to the 13 L HPDI NG engine, incorporating a TC system together with increasing the CR to 19 leads to the highest efficiency improvement at high loads. As shown in Figure 6.9(b), the incorporation of these modifications results in up to a 5.2% reduction in fuel consumption in the high-torque region ($T > 0.5 T_{\max}$) compared to the downsized HPDI NG engine without them.

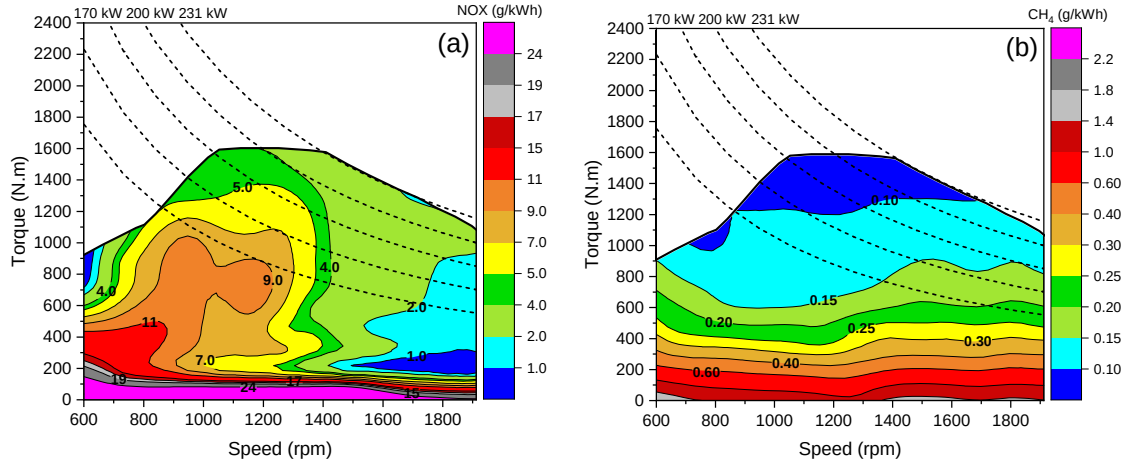


Figure 6.8: Engine-out brake specific emissions maps of the downsized 9 L HPDI NG engine: (a) NO_x , and (b) CH_4 .

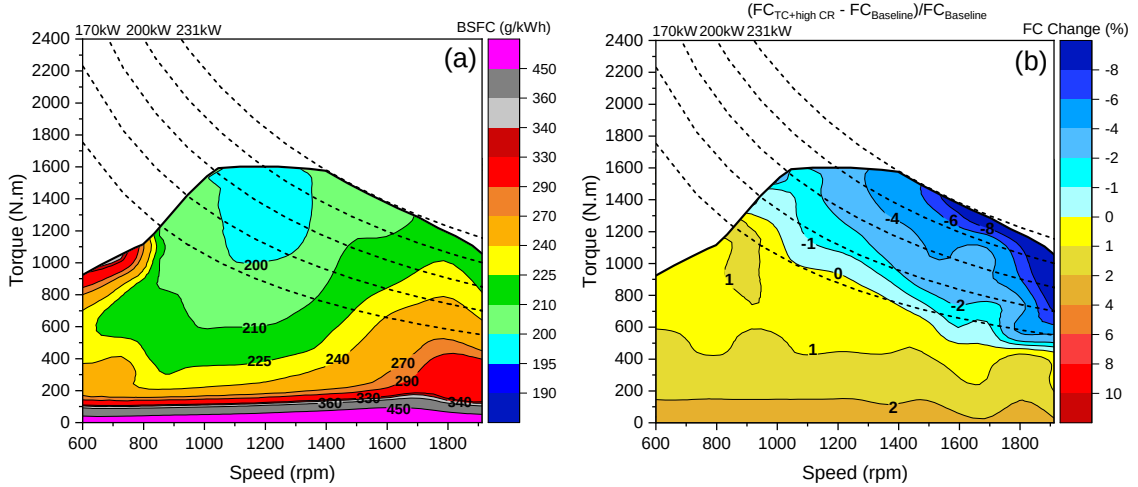


Figure 6.9: (a) Diesel-equivalent brake specific fuel consumption (BSFC) map of the 9 L HPDI NG engine with the turbo-compound system and increased compression ratio ($\text{CR} = 19$), and (b) relative percentage change in fuel consumption, measured in g/s, compared to the 9 L HPDI NG engine without these technologies.

The 13 L HPDI NG engine with a TC system and high CR exhibits higher BTE values in the high-torque region ($T > 0.8 T_{\max}$) compared to the downsized 9 L version with the same technologies. As previously shown in Figure 6.6, the high-efficiency region of the technology-enhanced 13 L engine corresponds to a BSFC range of 188.4 to 190 g/kWh, associated with a maximum BTE of 44.9%. In contrast, the high-efficiency region of the technology-enhanced downsized engine corresponds to a BSFC range of 195.6 to 200 g/kWh, with a maximum BTE of 43.2% (Figure 6.9). Consequently, downsizing the HPDI NG engine leads to a reduction in maximum BTE of up to 2% in the high-torque region and within the active engine operating speed range of 1000–1400 rpm.

The engine-out NO_x and CH_4 emissions of the 9 L HPDI NG engine with a TC system and high CR, across different torque–speed operating regions, are largely identical to those of the 9 L engine without these technologies, as reported in [43]. Therefore, the same brake specific emissions maps as those shown in Figure 6.8 can be used for vehicle modeling purposes.

6.4 Optimal Sizing of Powertrain Components for the Hybridized Configurations

This section presents the optimal sizing of powertrain components for Class 8 hybrid electric truck configurations equipped with different variants of low-carbon NG-CI engines, as introduced earlier in this chapter. For this purpose, the diesel-equivalent fuel consumption and emissions maps of the proposed NG engines are incorporated into the engine model of the hybrid electric truck configurations previously developed in Chapter 3. The hybridized configurations include the P_2 hybrid, the power-split hybrid, and the series plug-in hybrid electric configurations specifically designed for regional-haul HD truck applications. Then, the multi-stage optimization framework, as previously detailed in Chapter 4 (Section 4.3), is applied to each hybrid electric truck model to determine the optimal sizing of powertrain components. The primary objective of the optimization framework is to minimize the fuel consumption of the hybridized models over the Highway-2 drive cycle while maintaining the same propulsion system performance criteria as the conventional diesel baseline model in this study. The performance criteria are defined as those presented in Table 4.1 of Chapter 4, with the exception that the entire 44.4 km Davis Dam road grade profile, as previously created in Chapter 5, is used as the gradeability criterion.

6.4.1 Pre-transmission Parallel Hybrid Electric Truck Configuration

The P_2 hybrid electric configuration is designed based on two different approaches for engine sizing: 1) using the same rated power as the conventional diesel baseline model (336 kW) in combination with a smaller battery, and 2) using a downsized engine with a rated power of 231 kW in combination with a larger battery and greater reliance on the electrical power source. The variation ranges of the DOE variables, as part of the optimal component sizing process, are identical to those previously presented in Table 5.3 in Chapter 5. These ranges are determined based on the preliminary component sizing procedure discussed in Chapter 4, Section 4.3.1. Consequently, the optimal sizes of the primary powertrain components for the P_2 hybrid electric truck configurations equipped with different NG-CI engine variants are determined and summarized in Table 6.2.

The powertrain penalty weights of the designed P_2 hybrid configurations, relative to the conventional diesel baseline model (Table 6.2), are estimated using the procedure described in Section 4.3.2 of Chapter 4. This estimation assumes that the cargo capacity remains unchanged, with no reduction in the maximum payload of the truck. The weight of the dual-fuel system for the NG engines, including the LNG tanks and on-board fuel, is determined as previously discussed in Section 2.3 of Chapter 2 and is assumed to be the same as that of the conventional NG truck model presented in that section.

To improve the ability of the designed P_2 hybrid electric configurations equipped with a six-speed gearbox to maintain engine operation within the efficient speed range of 1100–1400 rpm, two distinct final drive ratios are adopted for highway and urban driving (Table 6.2). One ratio is optimized to minimize fuel consumption over the Highway-2 cycle, while the other is optimized for the same objective over the CSHVR drive cycle. To mitigate the engine on/off hysteresis effect, constraints are applied to ensure the engine operates for a minimum of 60 seconds during highway driving and 5 seconds during urban driving each time it starts. These values are tuned through a trial-and-error process for each of the P_2 hybrid electric configurations as part of the constrained rule-based EMS, as previously detailed in Section 3.2.2 of Chapter 3.

6.4.2 Power-Split Hybrid Electric Truck Configuration

The power-split hybrid electric configuration is designed using low-carbon NG-CI engines with a rated power of 336 kW, consistent with that of the diesel engine in

Table 6.2: Optimal sizes of the primary powertrain components for the Class 8 pre-transmission parallel (P₂) hybrid truck configurations equipped with different low-carbon NG-CI engine variants in this study.

Component/Module		P ₂ Hybrid (full-size 13 L engine)	P ₂ Hybrid (downsized engine)
Engine	Type	HPDI NG (base) HPDI NG (high CR ¹) HPDI NG (adv. SOI ²) HPDI NG (TC ³) HPDI NG (TC + high CR)	9 L HPDI NG (base) 9 L HPDI NG (TC + high CR) 6.7 L micro-pilot NG
	Max. power (kW)	336	231
	Max. torque (N.m)	2330	HPDI: 1602, micro-pilot: 1236
	Type	Dual fuel NG-diesel	Dual fuel NG-diesel
	Per tank capacity (L)	Diesel: 40 L, LNG:1300 L	Diesel: 40 L, LNG:1300 L
Fuel system	Number of tanks	Diesel: 2, LNG: 3	Diesel: 2, LNG: 3
	Max. power (kW)	173 (@1100 rpm)	205 (@1100 rpm)
	Max. torque (N.m)	1384	1640
	Max. speed (rpm)	5550	5550
Traction motor	Type	Li-Ion	Li-Ion
	Number of parallel modules	19	39
	Number of series cells	98	98
	Single cell capacity (Wh)	15.8	15.8
	Total capacity ⁴ (kWh)	29.4	60.8
High-voltage battery	Type	6-Speed automatic	6-Speed automatic
	Gear ratios	$[n_1, n_2, n_3, n_4, n_5, n_6] = [5.94, 3.92, 2.41, 1.34, 1, 0.87]$	$[n_1, n_2, n_3, n_4, n_5, n_6] = [6.93, 4.10, 2.67, 1.45, 1, 0.82]$
Multi-speed transmission	Type	6-Speed automatic	6-Speed automatic
Power converter	Voltage range (V)	12 const.	12 const.
Torque coupler	Power range (kW)	0 to 3.1	0 to 3.1
Final drive ratio	Single-gear ratio	2.0	1.8
Constrained Rule-based EMS	Engine efficiency threshold (δ_e)	Baseline HPDI: 0.009	Baseline HPDI: 0.009
		HPDI (high CR): 0.012	HPDI (high CR): 0.012
		HPDI (adv. SOI): 0.011	HPDI (adv. SOI): 0.011
		HPDI (TC): 0.001	HPDI (TC): 0.001
	HPDI (TC + high CR): 0.008	HPDI (TC + high CR): 0.008	HPDI (TC + high CR): 0.008
Constrained Rule-based EMS	Low battery SOC threshold (SOC _L)	0.28	0.30
	High battery SOC threshold (SOC _H)	0.52	0.55
	Powertrain penalty weight compared to the Conv. diesel baseline (kg)	HPDI without TC: +307.2 HPDI with TC: +397.2	HPDI without TC: +71.2 HPDI with TC: +146.2 Micro-pilot: -64.8

¹ High-pressure direct injection NG engine with high compression ratio.

² High-pressure direct injection NG engine with advanced start of injection.

³ High-pressure direct injection NG engine with turbo-compound technology.

⁴ The total battery capacity spans an SOC range of 0% to 100%, while the usable capacity is limited to the range of 20% to 90% SOC.

the conventional baseline configuration. To this end, two variants of the 13 L HPDI NG engine are considered: the baseline engine and a technology-enhanced version equipped with a TC system and high CR. The variation ranges of the critical DOE variables, as part of the optimal component sizing process for the power-split hybrid configuration, are summarized in Table 6.3. These ranges are determined based on the preliminary component sizing procedure previously discussed in Section 4.3.1 of Chapter 4. The optimal sizes of the primary powertrain components for the power-split hybrid electric truck configurations with the mentioned NG-CI engine variants are determined and summarized in Table 6.4.

Table 6.3: Variation ranges of the critical DOE variables in the optimal component sizing process of the Class 8 power-split hybrid electric truck configuration.

Variables (Unit)	Index	Variation Range	Increment Change
Final drive ratio (-)	n_{fd}	2-6	0.1
MG1 rated power (kW)	P_{MG1}	115-304	1
MG2 rated power (kW)	P_{MG2}	38-234	1
Torque coupler ratio (-)	n_{tc}	1-4	0.1
Number of parallel battery modules (-)	$N_{batt,p}$	5-65	1
Engine rated power (kW)	P_e	Constant 336	-
Ring gear number of teeth (-)	Z_r	75-200	1
Sun gear number of teeth (-)	Z_s	20-60	1
Engine efficiency threshold for EMS (%)	δ_e	0-0.03	0.001
Low battery SOC threshold for EMS (%)	SOC_L	0.2-0.6	0.01
High battery SOC threshold for EMS (%)	SOC_H	0.5-0.9	0.01
Engine minimum efficient speed for EMS (rpm)	$\omega_{e,min}$	870-1070	10
Engine maximum efficient speed for EMS (rpm)	$\omega_{e,max}$	1360-1560	10

Due to the inclusion of two e-motors (MG1 and MG2), multiple power converters, and a slightly larger battery, the powertrain weight of the designed power-split hybrid configurations is approximately 115 kg heavier than that of the parallel hybrid counterparts. In contrast to the parallel hybrid configurations, a single final drive ratio is sufficient for both highway and urban driving in the power-split configurations, owing to the decoupling of engine speed and torque from the wheels via the planetary gearset and the presence of a two-speed automatic transmission.

6.4.3 Series Plug-in Hybrid Electric Truck Configuration

The series plug-in hybrid electric configuration is designed using the downsized low-carbon NG-CI engines previously introduced in Section 6.3. These engines include the 9 L HPDI NG engine equipped with a TC system and high CR, as well as the 6.7 L micro-pilot NG engine, both with a rated power of 231 kW. Using downsized engines is recommended for series plug-in hybrid configurations, as the engine operates solely

Table 6.4: Optimal sizes of the primary powertrain components for the Class 8 power-split and series plug-in hybrid truck configurations equipped with different low-carbon NG-CI engine variants in this study.

Component/Module		Power-Split Hybrid (full-size 13 L engine)	Series Plug-in Hybrid (downsized engine)
Engine	Type	HPDI NG (base) HPDI NG (TC + high CR) ¹	9 L HPDI NG (TC + high CR) 6.7 L micro-pilot NG
	Max. power (kW)	336	231
	Max. torque (N.m)	2330	HPDI: 1602, Micro-pilot: 1236
	Type	Dual fuel NG-diesel	Dual fuel NG-diesel
Fuel system	Per tank capacity (L)	Diesel: 40 L, LNG:1300 L	Diesel: 40 L, LNG:1300 L
	Number of tanks	Diesel: 2, LNG: 3	Diesel: 2, LNG: 3
Motor generator 1 (MG1)	Max. power (kW)	155 (@1100 rpm)	453 (@1100 rpm)
or	Max. torque (N.m)	1240	3624
Traction motor	Max. speed (rpm)	5550	5550
Motor generator 2 (MG2)	Max. power (kW)	130 (@1866 rpm)	210 (@3000 rpm)
or	Max. torque (N.m)	666	669
Generator	Max. speed (rpm)	10,000	10,000
High-voltage battery	Type	Li-Ion	Li-Ion
	Number of parallel modules	22	19
	Number of series cells	98	140
	Single cell capacity (Wh)	15.8	124.1
	Total capacity ² (kWh)	34.6	330
Multi-speed	Type	2-Speed automatic	4-Speed automatic
transmission	Gear ratios	$[n_1, n_2]=$ [2.11, 1.23]	$[n_1, n_2, n_3, n_4]=$ [6.93, 1.45, 1, 0.82]
Planetary gearset	Ring gear number of teeth	148	-
	Sun gear number of teeth	35	-
Power converter	Voltage range (V)	12 const.	12 const.
for electrical acc.	Power range (kW)	0 to 3.1	0 to 3.1
Power converter for e-motor(s)	Voltage range (V)	0 to 500	-
	Power range (kW)	0 to 200	-
Torque coupler	Single-gear ratio	1.9	0.8
Final drive ratio	-	2.6	5.1
EMS-related parameters ³	Engine efficiency threshold (δ_e)	Baseline HPDI: 0.009 HPDI (TC + high CR): 0.008	-
	Low battery SOC threshold (SOC _L)	0.28	-
	High battery SOC threshold (SOC _H)	0.52	-
	Engine min. efficient speed ($\omega_{e,min}$)	1050	HPDI: 1050, Micro-pilot: 1180
	Engine max. efficient speed ($\omega_{e,max}$)	1400	HPDI: 1400, Micro-pilot: 1810
Powertrain penalty weight compared to the Conv. diesel baseline	(kg)	HPDI without TC: +422.3 HPDI with TC: +512.3	HPDI with TC: +1665.4 Micro-pilot: +1454.4

¹ High-pressure direct injection NG engine with the turbo-compound system and high compression ratio.

² The total battery capacity spans an SOC range of 0% to 100%, while the usable capacity is limited to the range of 20% to 90% SOC.

³ Optimized parameters for the constrained rule-based energy management strategy of the power-split hybrid and the dynamic programming-based strategy of the plug-in hybrid.

as a generator to delay battery depletion under the blended charge-depletion strategy. The decoupling of the engine from the wheels allows operation near its optimal efficiency region, while peak power requirements are supplied by the battery and the e-motor. The variation range of the critical DOE variables, as part of the optimal component sizing process for the plug-in hybrid configuration, are summarized in Table 6.5. Accordingly, the optimal sizes of the primary powertrain components are determined and summarized in Table 6.4.

Table 6.5: Variation ranges of the critical DOE variables in the optimal component sizing process of the Class 8 series plug-in hybrid electric truck configuration.

Variables (Unit)	Index	Variation Range	Increment Change
Final drive ratio (-)	n_{fd}	2-6	0.1
e-motor rated power (kW)	P_m	375-480	1
Generator rated power (kW)	P_{gen}	150-231	1
Torque coupler ratio (-)	n_{tc}	0.5-4	0.1
Number of parallel battery modules (-)	$N_{batt,p}$	15-30	1
Engine rated power (kW)	P_e	Constant 231	-

The significantly higher powertrain penalty weight (>1400 kg) of the designed series plug-in hybrid truck configurations, relative to the conventional diesel baseline model, is primarily attributed to the large battery size, which is determined to be 330 kWh (Table 6.4). The battery is sized to cover approximately half of a single repetition of the Highway-2 drive cycle in all-electric mode. It is assumed that the battery is fully recharged after completing each 265 km cycle, corresponding to the length of the Highway-2 route considered in this study. A single final drive ratio for both highway and urban driving conditions is selected for the designed series plug-in hybrid configurations, as shown in Table 6.4. This is because the e-motor solely supplies the tractive power at the wheels with a wide and smooth torque-speed characteristic, while the four-speed automatic gearbox accommodates varying driving conditions.

Despite the use of rule-based control strategies in the parallel and power-split hybrid configurations, a DP-based control method based on the Pontryagin Minimum Principle (PMP) [210], is employed for the EMS in the series plug-in hybrid configuration. The PMP optimization method adopts a blended charge-depletion strategy, in which the battery is discharged linearly as a function of vehicle travel distance, thereby avoiding transitions to charge-sustaining operation. In the blended charge-depletion strategy, the engine consistently contributes to the linear SOC reduction over the predefined driving distance [148]. This approach facilitates operation of the downsized engine within its high-efficiency region, corresponding to lower BSFC val-

ues. Detailed explanations of the DP-based control strategy for the series plug-in hybrid configuration were previously provided in Section 3.4.2 of Chapter 3.

6.5 Results and Discussions

The designed Class 8 hybrid electric truck configurations equipped with low-carbon NG-fueled engines were optimized to minimize fuel consumption over the reference Highway-2 drive cycle under the full cargo load condition. As these HD truck configurations will be used on a variety of routes and under different cargo load conditions, it is essential to compare both fuel consumption and GHG emissions across various drive cycles. Additionally, this section analyzes the engine operating conditions and the distribution of energy flow among the powertrain components in the proposed conventional and hybridized truck models.

6.5.1 Fuel Consumption of the Conventional Truck Models with Different Engines and Technologies

This section compares fuel consumption of the Class 8 conventional truck models equipped with various engine configurations and technology packages. The engines include the Cummins X15 diesel engine and five variants of the 13 L HPDI NG engine introduced earlier in this chapter. The NG engine variants include the baseline HPDI NG configuration, the one with a high CR, the one with advanced SOI, the one with a TC system, and the combined configuration incorporating both TC and high CR.

The conventional diesel baseline model consumes 1.9% to 3.6% less fuel relative to the conventional HPDI NG baseline model over the highway cycles (Highway-2 and HHDDT-65), as shown in Table 6.6. This difference is attributed to the slightly higher BTE of the Cummins diesel engine compared to its NG counterpart, particularly in high-torque regions ($T > 0.5 T_{\max}$). Incorporating the proposed technologies and modifications into the baseline HPDI NG engine reduces fuel consumption by up to 2.6% at 100% cargo load over the highway cycles; however, this benefit drops to below 1% at 35% cargo load condition. Across the urban cycle (CSHVR), the technology-enhanced NG engines can lead to increased fuel consumption, with penalties of up to 6.3% at low cargo loads compared to the baseline HPDI NG engine (Table 6.6).

The differences in fuel consumption among the designed Class 8 conventional truck models are primarily attributed to variations in the average engine BTE across the selected drive cycles and cargo load conditions, as shown in Table 6.7. Under an identical drive cycle and cargo load condition, higher average engine BTE corresponds

Table 6.6: Diesel-equivalent fuel consumption of the designed Class 8 conventional truck configurations equipped with various engines and technology packages, across the selected drive cycles under different cargo load conditions. Absolute fuel consumption values are reported for the conventional diesel and NG baseline models, while relative values are reported for the conventional models equipped with technology-enhanced NG engines, referenced to the conventional NG baseline model.

Designed Conventional Truck Configurations with Different Engines and Technologies	Diesel-Equivalent Fuel Consumption (L/tonne _{cargo} .100 km)								
	Highway-2			HHDDT-65			CSHVR		
	35% Cargo	70% Cargo	100% Cargo	35% Cargo	70% Cargo	100% Cargo	35% Cargo	70% Cargo	100% Cargo
Conv. Diesel Baseline (Absolute value)	4.11	2.41	1.86	4.11	2.31	1.78	5.50	3.48	2.84
Conv. HPDI NG Baseline (Absolute value)	4.22	2.48	1.93	4.19	2.37	1.83	5.41	3.45	2.83
Reduction in Fuel Consumption Relative to the Conv. HPDI NG Baseline Model $\left(\frac{\text{Conv. HPDI NG Baseline} - \text{Conv. HPDI NG with Technology}}{\text{Conv. HPDI NG Baseline}} \times 100 \right)$									
Conv. HPDI NG + CR	0.2	0.4	1.0	0.2	0.4	0.6	-0.7 ¹	-0.6	-0.4
Conv. HPDI NG + SOI	0	0.4	1.0	0.2	0.4	0.6	-0.6	0	0.4
Conv. HPDI NG + TC	0.5	1.2	1.6	-0.7	0.5	1.1	-5.9	-4.9	-1.4
Conv. HPDI NG + TC + CR	0.9	2.0	2.6	-0.5	0.8	2.2	-6.3	-5.2	-1.8

¹ Numbers shown in red color indicate conditions where the fuel consumption of the conventional truck models equipped with the technology-enhanced NG engines is worse than that of the conventional HPDI NG baseline model.

to lower engine fuel consumption. In conventional vehicles, due to the reliance on the ICE as the sole power generation source, engine operation spans a wide range from low-torque regions associated with high BSFC to high-torque regions with low BSFC values. As the tractive demand at the wheels decreases, due to less power-demanding urban driving or reduced cargo load, the ICE operation of conventional trucks shifts toward lower torque regions associated with lower BTE. Applying the proposed technologies and modifications to the baseline HPDI NG engine improves BTE in the high-torque region ($T > 0.5 T_{\max}$), particularly within the active speed range of 1000–1400 rpm. However, in the lower torque range, the corresponding BTE gains are limited or may even become negative. As a result, the higher proportion of high-torque operation in highway cycles enables the technology-enhanced engines to achieve higher average BTE, thereby reducing fuel consumption relative to the conventional NG baseline model.

The impact of drive cycle and cargo load variations on the engine operating points of the conventional HPDI NG baseline model is depicted in Figure 6.10. Shifting from highway drive cycles to urban cycles, or reducing the cargo load, causes the engine to operate less frequently within the high-torque region ($T > 0.5 T_{\max}$), associated with the BSFC values below 200 g/kWh. Consequently, the benefits of the technology-

Table 6.7: Engine average brake thermal efficiency of the designed Class 8 conventional truck configurations equipped with various engines and technology packages, across the selected drive cycles under different cargo load conditions.

Designed Conventional Truck Configurations with Different Engines and Technologies	Engine Average Brake Thermal Efficiency (%)								
	Highway-2			HHDDT-65			CSHVR		
	35% Cargo	70% Cargo	100% Cargo	35% Cargo	70% Cargo	100% Cargo	35% Cargo	70% Cargo	100% Cargo
Conv. Diesel Baseline	42.3	42.8	43.1	41.4	41.9	42.3	36.6	38.1	38.8
Conv. HPDING Baseline	41.5	41.8	42	41.1	41.2	41.3	37.4	38.2	39.0
Conv. HPDING + CR	41.6	42	42.3	41.2	41.4	41.5	37.1	38	38.9
Conv. HPDING + SOI	41.5	41.9	42.3	41.2	41.3	41.6	37.2	38.2	39.1
Conv. HPDING + TC	41.7	42.3	42.7	40.9	41.5	41.9	36.5	37.7	38.6
Conv. HPDING + TC + CR	41.9	42.7	43.2	41.0	41.7	42.3	36.4	37.6	38.5

enhanced HPDING engines become less pronounced—or may even reverse—resulting in higher fuel consumption for the corresponding conventional models, as presented earlier in Table 6.6. Among the selected drive cycles, Highway-2 imposes the highest torque demand on the engine due to the inclusion of a varying road grade profile and a higher average speed. However, CSHVR exhibits greater speed fluctuations characteristic of urban driving conditions (Figure 6.10).

6.5.2 Fuel Conversion Efficiency and Engine-Out Emissions of the Hybrid Electric truck Models with technology-Enhanced Natural Gas Engines

The impact of advanced engine technologies and modifications on fuel consumption of the Class 8 conventional truck models was evaluated in Section 6.5.1. Although some improvements were observed, particularly under highway drive cycles and high cargo load conditions, the benefits were limited or even diminished under lower cargo loads and urban driving conditions. This section extends the analysis by incorporating the technology-enhanced HPDING engines into pre-transmission parallel (P₂), power-split, and series plug-in hybrid electric Class 8 truck configurations. In addition to fuel consumption, the evaluation also considers engine-out CO₂, NO_x, and CH₄ emissions of the hybridized truck models relative to the conventional baseline configurations.

The designed hybrid electric truck configurations equipped with different variants of NG-CI engines exhibit lower diesel-equivalent fuel consumption than their conventional counterparts across all selected drive cycles and cargo load conditions, as shown in Table 6.8. Integrating engines with higher BTE in the high-torque region

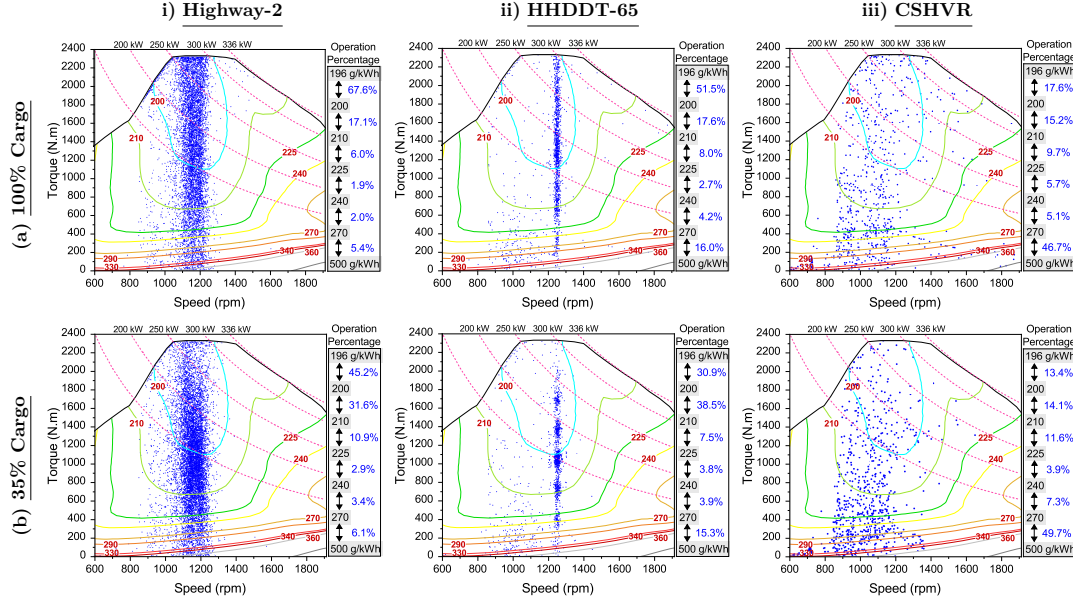


Figure 6.10: Weighted engine operating points on BSFC (diesel-equivalent) contour plots for the conventional HPDI NG baseline model across the three drive cycles: i) Highway-2, ii) HHDDT-65, and iii) CSHVR, under (a) 100% and (b) 35% cargo load conditions. Each point represents the ICE operating state during engine-on periods over a one-second interval. The percentage tables shown beside each sub-figure indicate the fraction of engine operating time spent within the corresponding BSFC ranges.

($T > 0.5 T_{\max}$) magnifies the fuel-saving potential of the hybridized configurations, as they enable more frequent engine operation within the high-efficiency zone. Moreover, hybrid electric architectures with higher net powertrain energy conversion efficiency, such as power-split and plug-in configurations, provide additional gains by enabling more flexible engine operation and more effective utilization of electrical energy pathway.

Among the designed parallel hybrid configurations, the one equipped with the 13 L HPDI NG + TC + CR engine achieves the greatest reductions in fuel consumption (Table 6.8). The reductions relative to the conventional NG baseline model reach up to 8.7% over the highway cycles (Highway-2 and HHDDT-65) and are more pronounced, reaching up to 37% over the urban cycle (CSHVR). The greater amount of regenerative braking energy recovered over the urban cycle is the primary reason for the significantly higher fuel consumption reductions. Among the proposed power-split hybrid configurations, the most fuel-efficient design is achieved with the same engine configuration, resulting in reductions of up to 10.2% and 38.3% over the highway and urban cycles, respectively. The plug-in hybrid configurations equipped with the micro-pilot NG engine exhibit substantially higher fuel consumption reductions

Table 6.8: Diesel-equivalent fuel consumption and engine-out CO₂ emissions of the designed Class 8 conventional and hybrid electric truck configurations equipped with various engines and technology packages, across the selected drive cycles under different cargo load conditions. Absolute fuel consumption and emission values are reported for the conventional models, while relative values are reported for the hybridized models, referenced to the conventional NG baseline model.

Designed Class 8 Truck Configurations with Different Engines and Technologies	Highway-2		HHDDT-65		CSHVR	
	35% Cargo	100% Cargo	35% Cargo	100% Cargo	35% Cargo	100% Cargo
Fuel Consumption of the Conventional Baseline Models (L/tonne_{cargo}.100 km)						
Conv. Diesel Baseline (Absolute value)	4.11	1.86	4.11	1.78	5.50	2.84
Conv. HPDI NG Baseline (Absolute value)	4.22	1.93	4.19	1.83	5.41	2.83
Engine-out CO₂ Emissions of the Conventional Baseline Models (g/tonne_{cargo}.km)						
Conv. Diesel Baseline (Absolute value)	109.25	49.51	109.18	47.17	144.66	75.30
Conv. HPDI NG Baseline (Absolute value)	86.25	39.45	85.75	37.43	110.67	57.90
Reduction in Fuel Consumption¹ Relative to the Conv. HPDI NG Baseline Model $\left(\frac{\text{Conv. HPDI NG Baseline} - \text{Hybrid Design}}{\text{Conv. HPDI NG Baseline}} \times 100 \right)$						
P ₂ HPDING + TC + CR (9L)	3.4	4.0	3.9	4.2	33.9	32.3
P ₂ Micro-Pilot NG (6.7 L)	4.5	5.4	4.7	5.6	34.5	33.3
P ₂ HPDING (13L)	3.6	4.2	4.1	4.4	34.2	32.5
P ₂ HPDING + CR (13L)	4.3	5.2	4.8	5.5	34.2	33.2
P ₂ HPDING + SOI (13L)	4.3	5.2	4.5	5.5	34.4	33.2
P ₂ HPDING + TC (13L)	5.9	6.7	6.4	7.1	35.3	33.9
P ₂ HPDING + TC + CR (13L)	7.4	8.3	7.9	8.7	37.0	35.0
Power-Split HPDING (13L)	5.8	5.2	6.6	5.8	36.5	34.6
Power-Split HPDING + TC + CR (13L)	9.5	9.2	10.2	9.9	38.3	36.1
Plug-in HPDING + TC + CR (9L)	33.7	24.9	34.4	26.8	68.6	53.0
Plug-in Micro-Pilot NG (6.7 L)	35.7	27.1	36.2	28.8	69.4	54.0
Reduction in Engine-Out CO₂ Emissions² of the Plug-in Hybrid Models Relative to the Conv. HPDI NG Baseline Model						
Plug-in HPDING + TC + CR (9L)	61.6	46.1	62.5	49.4	90.4	67.6
Plug-in Micro-Pilot NG (6.7 L)	63.4	48.1	64.1	51.2	91.0	68.4

¹ For non-plug-in hybrid models, the reduction in engine-out CO₂ emissions is equal to the reduction in diesel-equivalent fuel consumption.

² In the plug-in hybrid, tailpipe CO₂ reductions exceed fuel savings because the electrical energy contribution is assumed to produce zero tailpipe emissions. However, electrical energy consumption is included in the total diesel-equivalent fuel consumption on an energy-equivalent basis.

compared to the non-plug-in counterparts, reaching up to 36.2% over the highway cycles. This improvement is attributed to increased reliance on the electrical driveline, including an e-motor and a large battery that can be charged externally from the grid. Consequently, engine-on operation is significantly reduced, and when active, the engine operates predominantly within the high-efficiency region.

The reductions in engine-out (tailpipe) CO₂ emissions of the designed hybrid electric configurations result from both the lower carbon content of NG fuel and the reduced fuel consumption enabled by hybridization. As presented in Table 6.8, the CO₂ emissions of the conventional NG model over the selected drive cycles and cargo load conditions are reduced by 18.3% to 23.5% relative to the conventional diesel baseline model. The CO₂ emission reductions of the non-plug-in hybrid configurations correlate directly with their respective fuel consumption reductions relative to the conventional NG model. In contrast, tailpipe CO₂ reductions in the plug-in hybrid configurations exceed the associated fuel savings (Table 6.8), as the electrical energy used to externally charge the battery does not contribute to tailpipe emissions (upstream emissions from electricity generation are not considered).

One of the primary benefits of hybridization is increasing the amount of time that engine operates near its high-efficiency region, corresponding to lower BSFC values typically achieved at high torque levels (i.e., $T > 0.5 T_{\max}$). Under low torque demand conditions, such as urban driving, as well as during acceleration events, the electrical driveline supplies all or part of the required wheel power, thereby preventing inefficient engine operation. As shown in Table 6.9, across all the selected drive cycles and cargo load conditions, the hybrid electric configurations equipped with different variants of NG-CI engines exhibit a higher average engine BTE compared to the conventional NG baseline model. The improvements achieved by the proposed P₂ hybrid configurations range from 0.3% to 5.8%, with the highest values observed over the urban cycle (CSHVR) at lower cargo load conditions. Slightly higher improvements are achieved by the power-split hybrid configurations, ranging from 1.1% to 6.0%. For the plug-in hybrid configurations, the improvements range from 0.7% to 5.8%. The use of NG engine variants with higher overall efficiency in the mid-to-high torque regions further amplifies these improvements when combined with powertrain hybridization.

Following the integration of various engine configurations into the parallel and power-split hybrid architectures, the EMS-related optimization parameters were recalibrated to ensure engine operation remains within the high-efficiency region. To this end, the engine efficiency threshold (δ_e), which defines the torque–speed operating boundaries within the rule-based control strategies, was optimized as part of

Table 6.9: Engine average brake thermal efficiency for the designed Class 8 conventional HPDI NG baseline and hybrid electric configurations equipped with different variants of low-carbon NG-CI engines, across the selected drive cycles under different cargo load conditions. The average efficiency is calculated during engine-on operation.

Designed Class 8 Truck Configurations with Different Engines and Technologies	Engine Average Brake Thermal Efficiency (%)					
	Highway-2		HHDDT-65		CSHVR	
	35% Cargo	100% Cargo	35% Cargo	100% Cargo	35% Cargo	100% Cargo
Conv. HPDI NG Baseline	41.5	42.0	41.1	41.3	37.4	39.0
P ₂ HPDI NG + TC + CR (9L)	42.0	42.3	41.9	42.0	39.9	41.1
P ₂ Micro-Pilot NG (6.7 L)	42.9	43.1	42.8	43.0	41.8	42.0
P ₂ HPDI NG (13 L)	42.4	42.5	42.2	42.4	41.1	41.3
P ₂ HPDI NG + CR (13 L)	42.8	42.9	42.6	42.8	41.5	41.7
P ₂ HPDI NG + SOI (13 L)	42.7	42.9	42.5	42.8	41.6	41.8
P ₂ HPDI NG + TC (13 L)	43.5	43.7	43.3	43.6	42.2	42.4
P ₂ HPDI NG + TC + CR (13 L)	44.2	44.4	44.0	44.3	43.2	43.4
Power-Split HPDI NG (13 L)	42.6	42.8	42.5	42.7	41.7	41.9
Power-Split HPDI NG + TC + CR (13 L)	44.4	44.7	44.2	44.5	43.4	43.6
Plug-in HPDI NG + TC + CR (9L)	42.8	42.7	42.8	42.7	42.7	42.5
Plug-in Micro-Pilot NG (6.7 L)	43.3	43.2	43.4	43.3	43.2	43.1

the component sizing framework, as previously described in Section 4.3 of Chapter 4. The calibrated operating boundaries for each engine variant in the P₂ and power-split hybrid configurations are provided in Appendix C (Figure C.1). For the series plug-in hybrid configuration, the PMP optimization was re-executed using the revised engine efficiency map and constraints, yielding updated optimal engine speed–torque trajectories under the blended charge-depletion strategy.

The designed hybrid Class 8 electric truck configurations significantly reduce engine-out NO_x and CH₄ emissions relative to the conventional NG baseline model, as shown in Table 6.10. As discussed earlier in Sections 6.2 and 6.3, the emission maps of different HPDI NG engine variants indicate that the brake-specific NO_x and CH₄ values decrease with increasing engine torque. Therefore, hybridization leads to more pronounced emission reductions by promoting engine operation at higher torque levels while reducing the overall engine-on duration over the drive cycle compared to conventional counterparts. It should be noted that the reported emission reductions in Table 6.10 reflect only engine-out emissions and do not include emissions from electricity generation used to externally charge the battery in the plug-in hybrid configuration.

The engine-out NO_x emission reductions for the P₂ hybrid configurations, relative

to the conventional NG baseline model, range from 2% to 17.9% over the highway cycles and from 22.5% to 49.7% over the urban cycle (Table 6.10). The worst relative reduction in engine-out NO_x emissions among the proposed P_2 hybrid configurations is observed for the parallel HPDI NG + SOI design, primarily due to the increase in NO_x emissions associated with advanced SOI. The power-split hybrid configurations achieve more pronounced reductions, ranging from 5.8% to 19.8% over the highway cycles and from 34.7% to 52.0% over the urban cycle. The most significant reductions are achieved by the plug-in hybrid configurations, ranging from 70.8% to 83.6% over the highway cycles and from 76.0% to 87.7% over the urban cycle.

The reduction of engine-out CH_4 emissions in HPDI NG engines is particularly important, as methane oxidation in aftertreatment systems presents significant challenges. In this context, the designed hybrid configurations demonstrate significant potential by eliminating methane emissions associated with low-load engine operation. (Table 6.10). For the P_2 hybrid configurations, the CH_4 emission reductions relative to the conventional NG baseline model range from 30.2% to 62.3% over the highway cycles and from 85.9% to 92.2% over the urban cycle. The power-split hybrid configurations achieve slightly greater reductions, ranging from 34.1% to 63.2% over the highway cycles and from 87.1% to 92.9% over the urban cycle. The plug-in hybrid configuration exhibits near-zero engine-out CH_4 emissions, primarily due to the substantially reduced engine-on duration and operation predominantly within high-torque regions, where methane emissions remain below 0.1 g/kWh.

Overall, the 13 L HPDI NG engine equipped with both a TC system and a high CR is identified as the most advantageous among the studied NG engine options for integration with non-plug-in hybrid configurations. It achieves the lowest fuel consumption and NO_x and CH_4 emissions across both highway and urban driving conditions. However, for plug-in hybrid configurations, where engine downsizing is required, the 6.7 L micro-pilot NG engine exhibits slightly greater fuel consumption and CO_2 emission reductions compared to the downsized 9 L HPDI NG engine with TC and high CR. This is attributed to the 1-2% higher overall BTE of the micro-pilot engine relative to the HPDI counterpart in the high-torque region.

6.5.3 Hybridization Effects on Engine Operating Points

In this section, the ICE operating points and their locations on the engine BSFC maps are discussed among the designed hybrid electric Class 8 truck configurations. This analysis underscores the capability of the P_2 , power-split, and series plug-in hybrid configurations to enhance engine average BTE compared to their conventional

counterparts, irrespective of cargo load and drive cycle variations.

One of the main benefits of hybridization is the ability to decouple engine torque and/or speed from the wheels, allowing the engine to operate more frequently within its high-efficiency region at higher torque levels. Figure 6.11 shows the ICE operating points overlaid on the engine BSFC contour maps for four of the designed hybrid electric configurations across the selected drive cycles under 100% cargo load, along with the corresponding percentages of engine operation within different BSFC zones. Over the highway drive cycles (Highway-2 and HHDDT-65), the share of engine operation within the high-efficiency region, defined by BSFC values below 200 g/kWh, remains within the range of 88.8% to 93.2% for the selected hybrid configurations (Figure 6.11). In contrast, this share is significantly lower for the conventional NG baseline model, ranging from 51.5% to 67.6%, as previously shown in Figure 6.10. Over the urban drive cycle (CSHVR), this share ranges from 80.7% to 88.8% for the selected hybrid configurations, whereas it drops to only 17.8% for the conventional NG model.

The use of technology-enhanced HPDI NG engines, which improve BTE in high-torque regions ($T > 0.5 T_{\max}$), further amplifies the gains in cycle-average engine BTE for the hybrid configurations compared to the conventional NG baseline model. As shown in Figure 6.11, the P₂ HPDI NG + TC + CR model spends 80.6% to 90.4% of its engine operation within the high-efficiency region corresponding to BSFC values between 187 and 195 g/kWh across the selected drive cycles. As a result, this configuration achieves a higher average engine BTE than the other hybrid configurations shown in Figure 6.11, whose highest-efficiency regions correspond to BSFC values between 195 and 200 g/kWh.

For the conventional models, reducing the cargo load from 100% to 35% and transitioning to a lower drive cycle power demand result in a substantial reduction in the share of engine operation within high-torque regions associated with BSFC values below 200 g/kWh. For the conventional NG model over the Highway cycles, this reduction ranges from 20.6% to 22.4%, as previously shown in Figure 6.10. In contrast, for the hybrid configurations, the reduction is negligible or substantially smaller, reaching up to 8.2% (Figure 6.12). Hence, hybridization leads to greater improvements in average engine BTE relative to the conventional configurations under lower power-demand conditions, such as reduced cargo loads and urban driving cycles.

The power-split hybrid configuration, which decouples both engine torque and speed from the wheels, offers greater flexibility to maintain the ICE operation within

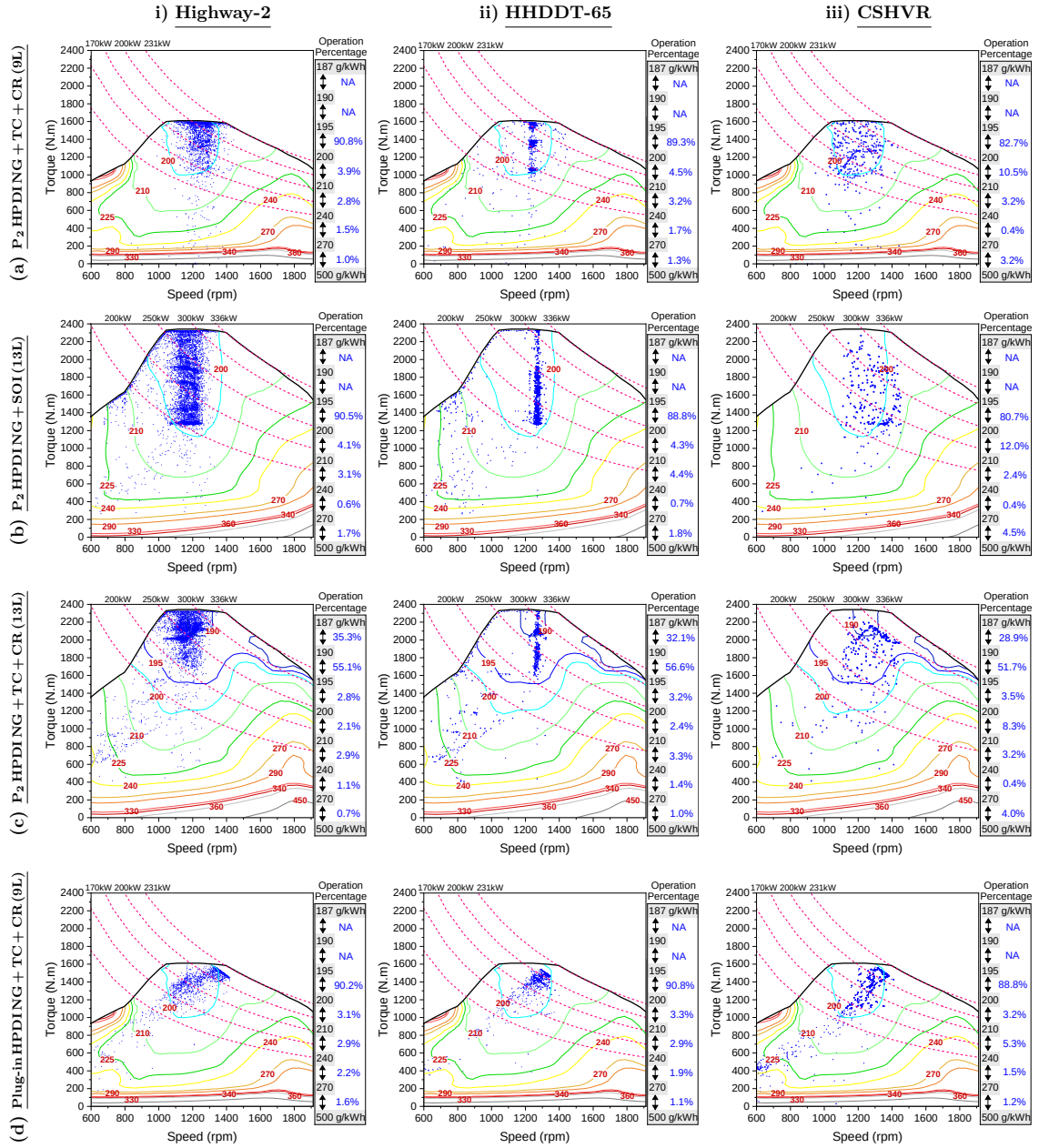


Figure 6.11: Weighted engine operating points on BSFC contour plots over the three drive cycles: (i) Highway-2, (ii) HHDDT-65, and (iii) CSHVR, evaluated for the designed hybrid electric Class 8 truck configurations at 100% cargo load. Sub-figures correspond to: (a) P_2 HPDI NG + TC + CR (9L), (b) P_2 HPDI NG + SOI (13L), (c) P_2 HPDI NG + TC + CR (13L), and (d) series plug-in HPDI NG + TC + CR (9L). Each point represents the ICE operating state during engine-on periods over a one-second interval. The percentage tables shown besides each sub-figure indicate the fraction of engine operating time spent within the corresponding BSFC ranges.

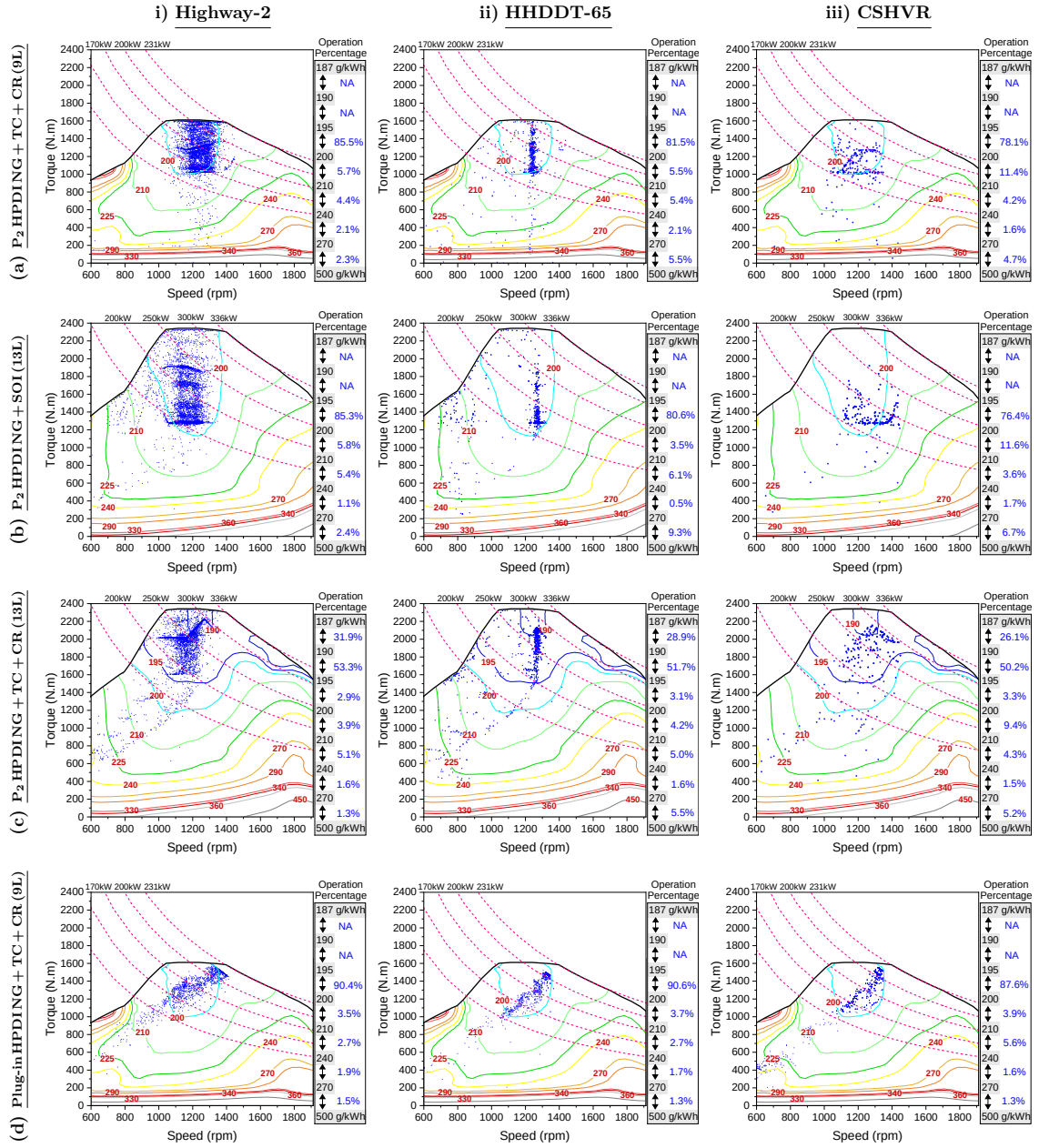


Figure 6.12: Weighted engine operating points on BSFC contour plots over the three drive cycles: (i) Highway-2, (ii) HHDDT-65, and (iii) CSHVR, evaluated for the designed hybrid electric Class 8 truck configurations at 35% cargo load. Sub-figures correspond to: (a) P₂ HPDI NG + TC + CR (9L), (b) P₂ HPDI NG + SOI (13L), (c) P₂ HPDI NG + TC + CR (13L), and (d) series plug-in HPDI NG + TC + CR (9L). Each point represents the ICE operating state during engine-on periods over a one-second interval. The percentage tables shown besides each sub-figure indicate the fraction of engine operating time spent within the corresponding BSFC ranges.

its high-efficiency region compared to the parallel hybrid designs, which only decouple torque. Figure 6.13 shows that the power-split hybrid design equipped with the baseline HPDI NG engine achieves a 1.8% to 2.2% higher percentage of engine operation within the high-efficiency region (BSFC < 200 g/kWh) over the highway cycles than the parallel counterpart with the same engine. This difference becomes more pronounced over the urban drive cycle, reaching up to 3.7% under the full cargo load condition. As a result, the power-split hybrid configuration achieves a slightly higher average engine BTE than its parallel counterpart under the same drive cycle and cargo load condition.

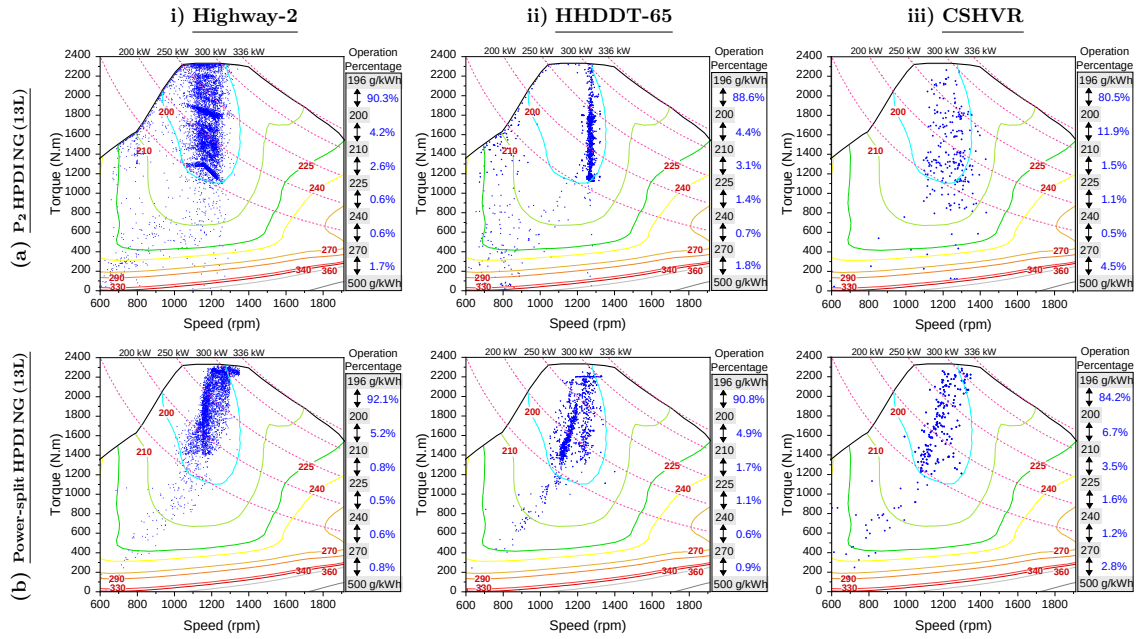


Figure 6.13: Weighted engine operating points on BSFC contour plots over the three drive cycles: (i) Highway-2, (ii) HHDDT-65, and (iii) CSHVR, evaluated for the designed hybrid electric Class 8 truck configurations at 100% cargo load. Sub-figures correspond to: (a) P₂ HPDI NG (13L), and (b) power-split HPDI NG (13L). Each point represents the ICE operating state during engine-on periods over a one-second interval. The percentage tables shown besides each sub-figure indicate the fraction of engine operating time spent within the corresponding BSFC ranges.

Hybridization allows the engine to remain shut down when the battery has sufficient charge and the e-motor(s) can independently meet the road power demand. As a result, the engine is activated less frequently and operates predominantly within its high-efficiency region when activated. Hybridized configurations exhibit greater reductions in engine-on duration as the drive cycle becomes less power-demanding and the available regenerative braking energy increases, as observed over the urban cycle

(CSHVR). As indicated in Table 6.11, the plug-in hybrid configuration maintains a substantially shorter engine-on duration than the non-plug-in hybrid configurations across all selected drive cycles and cargo load conditions. The presence of a large battery in the plug-in hybrid design, which can be fully depleted (from 90% to 20%) over a known driving distance, further enables extended engine-off operation. The engine operates solely to charge the battery in accordance with the predefined blended charge-depletion trajectory.

Table 6.11: Time-averaged percentage of engine-on duration for the designed Class 8 conventional and hybrid electric truck configurations across the selected drive cycles and cargo load conditions.

Designed Class 8 Truck Configurations with Different Engines and Technologies	Time-Averaged Percentage of Engine-On Duration (%)					
	Highway-2		HHDDT-65		CSHVR	
	35% Cargo	100% Cargo	35% Cargo	100% Cargo	35% Cargo	100% Cargo
Conv. HPDING Baseline	100	100	100	100	100	100
P ₂ HPDING + TC + CR (9L)	86.0	94.9	82.1	86.6	27.7	35.4
P ₂ HPDING (13L)	78.9	83.4	73.3	76.7	23.3	30.3
P ₂ HPDING + SOI (13L)	78.2	82.9	72.8	76.2	21.7	29.6
P ₂ HPDING + TC + CR (13L)	75.1	79.9	69.2	73.9	20.8	27.4
Power-Split HPDING (13L)	74.5	68.1	72.2	75.0	20.3	27.1
Plug-in HPDING + TC + CR (9L)	32.9	49.2	24.9	38.4	8.7	18.8

Among the designed non-plug-in hybrid configurations, the power-split design maintains a slightly shorter engine-on duration than its parallel counterparts across the selected drive cycles and cargo load conditions, as shown in Table 6.11. This is due to the presence of two e-motors in the power-split hybrid configuration, capable of delivering propulsion simultaneously, thereby increasing all-electric drive capability. In contrast, the parallel hybrid configuration equipped with the downsized NG engine exhibits the highest engine-on duration under the selected conditions (Table 6.11). This is because the downsized engine must operate more frequently to sustain the battery SOC at the same level at the beginning and end of the trip.

6.5.4 Powertrain Energy Balance and Component Efficiencies

This section analyzes the energy flows between the primary components of the designed conventional and hybrid electric truck configurations equipped with different variants of the HPDI NG engine. It provides insights into the component-level energy efficiencies of the designed configurations.

Hybridization offers the ability to shift engine operation into higher efficiency zone (discussed in Section 6.5.3) and recover energy during deceleration. Each energy conversion results in an efficiency penalty (energy loss), which reduces the net vehicle fuel consumption and CO₂ emissions benefits of the hybrid powertrains. The relative importance of these factors can be assessed using energy flow diagrams, as shown over the Highway-2 drive cycle at 100% cargo load condition in Figure 6.14. In these diagrams, the ‘Fuel’ block represents the total chemical energy supplied by fuel to propel the vehicle. The ‘Driving Resistance’ block accounts for all energy losses that occur downstream of the gearbox to the driving wheels, including aerodynamic drag, rolling, braking, grade and inertia resistances, along with losses from final drives. The hybrid electric designs show an increase of up to 5.3% in the power delivered to the driving wheels, to overcome their increased weight, as previously detailed in Section 6.4. It should be noted that the additional energy used to accelerate the heavier hybridized vehicles can be partially recovered during deceleration through regenerative braking.

The equivalent fuel consumption (L/100 km) of the non-plug-in hybrid powertrains is directly proportional to the energy supplied by the fuel, as shown in Figure 6.14, since these configurations rely solely on the engine to maintain the battery state of charge. In contrast, for the plug-in hybrid configuration, the total equivalent energy consumption includes contributions from both the fuel and the electrical energy drawn from the battery, which can be externally charged.

The presence of electrical driveline components and higher auxiliary loads in hybridized powertrains introduces additional energy losses compared to the conventional configuration. However, these losses are offset by improvements in engine BTE, regenerative braking, and the use of more efficient downsized automatic gearboxes, resulting in a lower fuel energy demand than the conventional counterpart.

The lower fuel energy demand of the power-split hybrid configuration, compared to the parallel counterpart with the same engine, is primarily attributed to the higher operational efficiencies of the engine, e-motors, and transmission system. The power-split design allows for more flexible hybrid operational modes, enabled by a planetary gearset, letting the engine and e-motors operate more often in their high-efficiency regions. This results in reduced energy losses from the engine and e-motors compared to the P₂ hybrid design under the same driving conditions (Figure 6.14). Furthermore, total energy losses from the planetary gearset and the 2-speed gearbox are lower than those from the 6-speed gearbox in the P₂ hybrid design. Moreover, the presence of two e-motors allows a slightly higher recovery of the regenerative braking energy

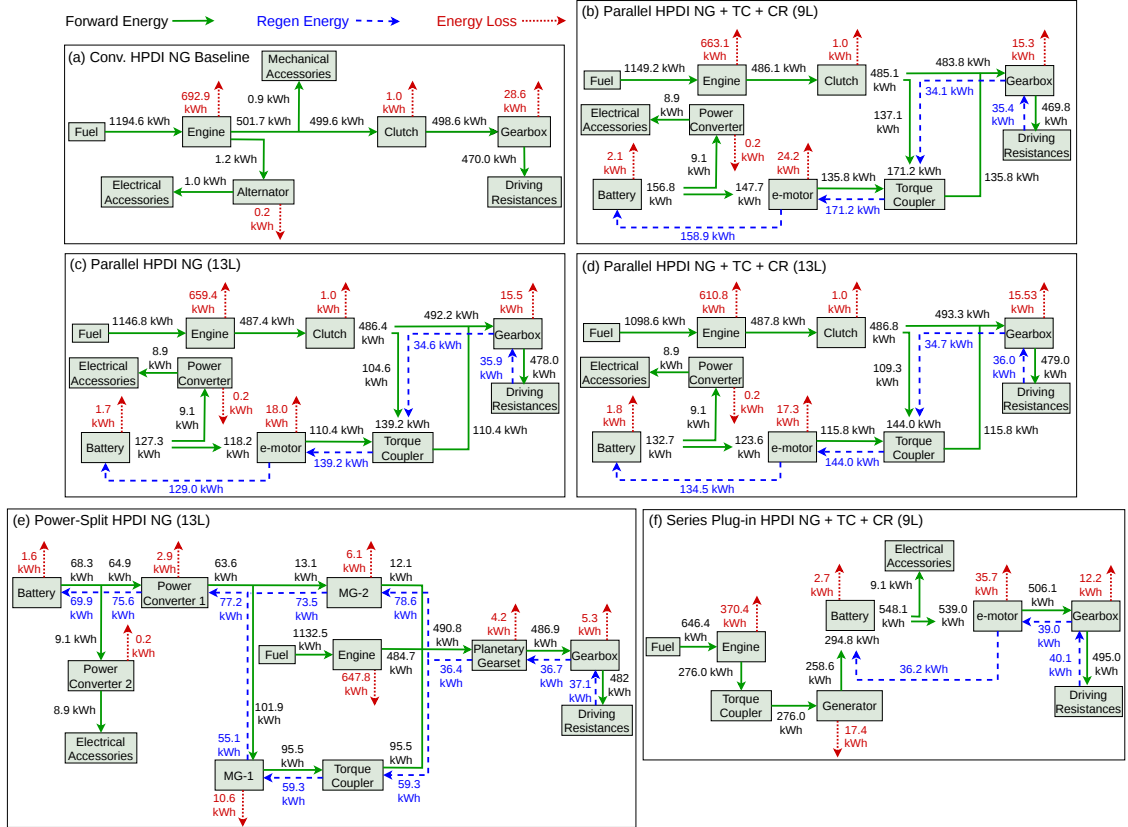


Figure 6.14: Energy flow diagrams for the designed class 8 regional-haul truck configurations over one repetition of the Highway-2 drive cycle (265 km) at 100% cargo load. Sub-figures correspond to: (a) conventional HPDI NG baseline, (b) P_2 HPDI NG + TC + CR (9L), (c) P_2 HPDI NG (13L), (d) P_2 HPDI NG + TC + CR (13L), (e) power-split HPDI NG (13L), and (f) series plug-in HPDI NG + TC + CR (9L) powertrains. The term ‘Driving Resistances’ includes aerodynamics, rolling, braking, acceleration, and grade resistive losses, along with losses from the final drives.

at wheels compared to the P₂ hybrid design, which incorporates a single e-motor. The power-split design can enable one of the e-motors to operate in generation mode and directly transfer power to the other e-motor operating in traction mode, without routing the generated energy through the battery and power converters. This helps minimize the losses associated with the battery and power converters. However, the P₂ hybrid design must send all the regenerative energy to the battery before being reused by the e-motor.

The primary reason for the difference in fuel energy demand between the parallel hybrid configurations equipped with full-size (13 L) NG engines is the variation in average engine BTE, influenced by the integrated technology packages and modifications, as shown in Figure 6.14(c,d). In the P₂ hybrid configuration with the downsized (9 L) NG engine, as shown in Figure 6.14(b), higher energy losses in both the e-motor and the engine lead to a higher fuel energy demand compared to the P₂ configurations with full-size engines. The increased reliance on the e-motor for propulsion during transient driving conditions reduces e-motor propulsion efficiency and requires the downsized engine to consume more energy for sustaining the battery charge over the trip.

The powertrain component sizes of the designed hybrid electric Class 8 truck configurations were optimized to minimize fuel consumption over the Highway-2 drive cycle at 100% cargo load conditions. However, variations in cargo load and drive cycles can affect the fuel consumption reduction of the hybridized configurations relative to the conventional NG baseline model. This is due to changes in the operating efficiencies of hybrid powertrain components and the amount of regenerative braking energy recoverable from the wheels. The cycle-average efficiencies of the e-motor(s), transmission system, and the available regenerative braking energy for each designed hybrid configuration across the selected drive cycles at both 100% and 35% cargo loads are provided in Appendix C (Table C.1).

6.6 Summary of Chapter

This chapter optimized full hybrid electric Class 8 truck configurations equipped with different variants of low-carbon natural gas (NG)-CI engines to reduce fuel consumption and engine-out emissions (CO₂, CH₄, and NO_x) relative to the conventional diesel baseline model. The fuel consumption and emission maps of the considered diesel-pilot premixed NG (micro-pilot NG) and technology-enhanced diesel-pilot HPDI NG engines were presented. The technologies and modifications applied to the HPDI NG

engine to improve its efficiency, particularly for hybrid applications at mid-to-high torque regions, include increasing the compression ratio (CR), advancing the start of injection (SOI), and integrating a turbo-compound (TC) system. The considered hybrid electric truck configurations include a pre-transmission parallel (P_2), a power-split, and a series plug-in hybrid design. Comparisons of fuel consumption and emissions were performed between the hybridized configurations equipped with different NG-CI engine variants and the conventional models equipped with the Cummins X15 diesel or the baseline 13 L HPDI NG engine, over both highway and urban drive cycles. Fuel consumption was expressed in diesel-equivalent units of $\text{L/tonne}_{\text{cargo}} \cdot 100 \text{ km}$, and emissions were reported in $\text{g/tonne}_{\text{cargo}} \cdot \text{km}$. The key findings of this chapter are as follows:

Technology-enhanced HPDI NG engines:

Incorporating the aforementioned modifications and technologies into the baseline HPDI NG engine leads to a relative fuel consumption reduction (measured in g/s) of 1% to 4.2%, particularly at mid-to-high torque regions ($T > 0.5 T_{\text{max}}$). However, these modifications result in relatively higher fuel consumption at lower-than-mid torque levels, in the range of 1% to 8%, within the speed range of 1100–1400 rpm. The most promising technology package is the combination of a high CR with a TC system, which achieves the greatest reduction at high torque regions without adversely affecting engine-out CH_4 and NO_x emissions.

Downsizing the baseline HPDI NG engine from 13 L (336 kW) to a 9 L (231 kW) version, while incorporating the same technologies, leads to a 1% to 2% increase in fuel consumption at mid-to-high torque levels. This is attributed to greater thermal and exhaust enthalpy losses in the downsized four-cylinder engine compared to the six-cylinder version. This performance penalty makes the downsized engine less efficient for hybrid applications, where the engine is operated more frequently within mid-to-high torque regions.

Conventional NG vs. Conventional diesel:

The conventional model equipped with the baseline 13 L HPDI NG engine exhibits fuel consumption comparable to the diesel counterpart, with up to 3.6% higher consumption on highway cycles and 1.7% lower consumption on urban cycles. These differences are attributed to variations in overall engine brake thermal efficiency (BTE) between the diesel and HPDI NG engines across different torque regions.

Employing the 13 L HPDI NG engine with the proposed technology packages leads

to up to a 2.6% reduction in fuel consumption under high power demanding conditions, such as highway driving at full cargo load, relative to the conventional diesel counterpart. However, this benefit becomes marginal or can even reverse, resulting in up to a 6.3% increase in fuel consumption under lower power demanding conditions such as partial cargo loads or urban driving. Therefore, the use of the technology-enhanced HPDI NG engine is not recommended for conventional trucks, as these technologies are primarily optimized to improve performance in high torque regions, which are more effectively utilized in hybridized configurations.

Hybridization effects on engine operation:

Hybridization enables the engine to either remain off or operate for more than 85% of the time within high-torque regions ($T > 0.5 T_{\max}$) associated with higher BTE, regardless of variations in drive cycles (highway or urban) or cargo load conditions. In contrast, this percentage varies significantly for conventional models, typically ranging from 13% to 75%, with lower values observed under less power-demanding conditions such as low cargo loads or urban driving. This results in a higher cycle-average BTE for hybridized models, thereby improving net powertrain energy efficiency.

Therefore, incorporating technology-enhanced engines that achieve higher BTE at high torque regions further amplifies the cycle-average BTE of hybridized configurations compared to the conventional counterparts. This synergy between hybridization and engine-level efficiency improvements leads to additional gains in overall powertrain energy efficiency.

Hybridization effects on fuel consumption:

The designed hybridized configurations achieve lower equivalent fuel consumption than their conventional counterparts due to higher average engine BTE and the recovery of regenerative braking energy over both highway and urban drive cycles. The greatest reduction is observed for the series plug-in hybrid configuration equipped with the downsized (6.7 L, 231 kW) micro-pilot NG engine, reaching up to 36.2% over highway cycles and up to 69.4% over urban cycles, relative to the conventional NG baseline model. This improvement is attributed to the increased reliance on the electrical driveline, including the e-motor and a large battery that can be charged externally from the grid. Consequently, engine-on duration is significantly reduced, and when active, the engine operates predominantly within the high-efficiency region.

Among the designed non-plug-in hybrid configurations, the greatest reduction is observed for the power-split hybrid design equipped with the 13 L, 336 kW HPDI NG

engine with both a TC system and a high CR, reaching up to 10.2% over highway cycles and up to 38.3% over urban cycles, relative to the conventional NG baseline model. For the P₂ hybrid design equipped with the same engine, the reductions are slightly less pronounced, reaching up to 8.7% over highway cycles and up to 37% over urban cycles. The greater reductions of the power-split configuration are attributed to its higher operational flexibility, enabled by the planetary gaset coupled with a higher-efficiency two-speed gearbox, which allows both the engine and the e-motors to operate closer to their peak efficiencies.

Hybridization effects on NO_x emissions:

Co-optimization of the engine and hybrid powertrain significantly reduces engine-out NO_x emissions primarily by concentrating engine operation in high-efficiency regions with minimum emissions, and avoiding low-load, inefficient regions. The greatest reductions are observed for the plug-in hybrid configurations equipped with a downsized NG-CI engine, ranging from 70.8% to 83.6% over highway cycles and from 76% to 87.7% over urban cycles, relative to the conventional NG baseline model.

These reductions for the P₂ hybrid configurations equipped with different variants of NG-CI engines range from 2% to 17.9% over highway cycles and from 22.5% to 49.7% over urban cycles. The power-split hybrid configurations achieve more pronounced reductions, ranging from 5.8% to 19.8% over highway cycles and from 34.7% to 52% over urban cycles. Notably, the hybrid configurations equipped with the 13 L HPDI NG engine with both a TC system and a high CR exhibit the highest NO_x reductions among the considered NG engine variants and technology packages.

Hybridization effects on GHG (CO₂ and CH₄) emissions:

The reductions in tailpipe GHG emissions (CO₂-equivalent), including CH₄, for the designed Class 8 conventional and hybridized configurations equipped with different variants of NG-CI engines, relative to the conventional diesel baseline model, are shown in Figure 6.15. These reductions are evaluated under the full cargo load condition over both highway (Highway-2) and urban (CSHVR) drive cycles. Replacing the diesel engine with the low-carbon HPDI NG engine in the conventional truck model leads to tailpipe GHG emission reductions of 19.7% over the highway cycle and 20.7% over the urban cycle. By integrating powertrain hybridization with the technology-advanced HPDI NG engines, these reductions become more pronounced, reaching up to 56.9% over the highway cycle and 75.1% over the urban cycle for the plug-in hybrid configuration.

Among the parallel and power-split hybrid configurations, the greatest equivalent GHG reductions are achieved by those equipped with the 13 L HPDI NG engine incorporating both a TC system and a high CR, as depicted in Figure 6.15. The capability of hybridization to either keep the engine off or operate it predominantly within mid-to-high torque regions ($T > 0.5 T_{\max}$), which are associated with lower CO_2 and CH_4 emissions, is the primary reason for these reductions. Moreover, incorporating the TC system and high CR into the HPDI NG engine improves the overall BTE within mid-to-high torque regions, leading to lower fuel consumption and consequently reduced CO_2 emissions compared to the other proposed technology packages.

To further explain the large tailpipe GHG emission reductions of the plug-in hybrid configuration in Figure 6.15, Figure 6.16 shows the contribution of on-board fuel energy and grid-supplied electrical energy for the plug-in hybrid equipped with the downsized HPDI NG engine with TC and high CR. Since this configuration uses a DP-based blended charge-depletion strategy, the battery is fully depleted over the 265 km trip, resulting in a constant grid electrical energy contribution of 256 kWh for all cases. The remaining energy demand is supplied by the engine, which varies with drive cycle and cargo load. At 35% cargo load, grid electricity provides 42–43% of the total supplied energy over the highway cycles and 69% over the urban CSHVR cycle. At 100% cargo load, this share decreases to 28–31% because of the higher fuel energy demand. Therefore, the large GHG reduction of the plug-in hybrid configuration is partly due to shifting a considerable portion of the vehicle energy demand from on-board fuel to grid electricity. Since Figure 6.15 reports tank-to-wheel GHG emissions, emissions associated with grid electricity generation are not included and should be addressed in future well-to-wheel analysis.

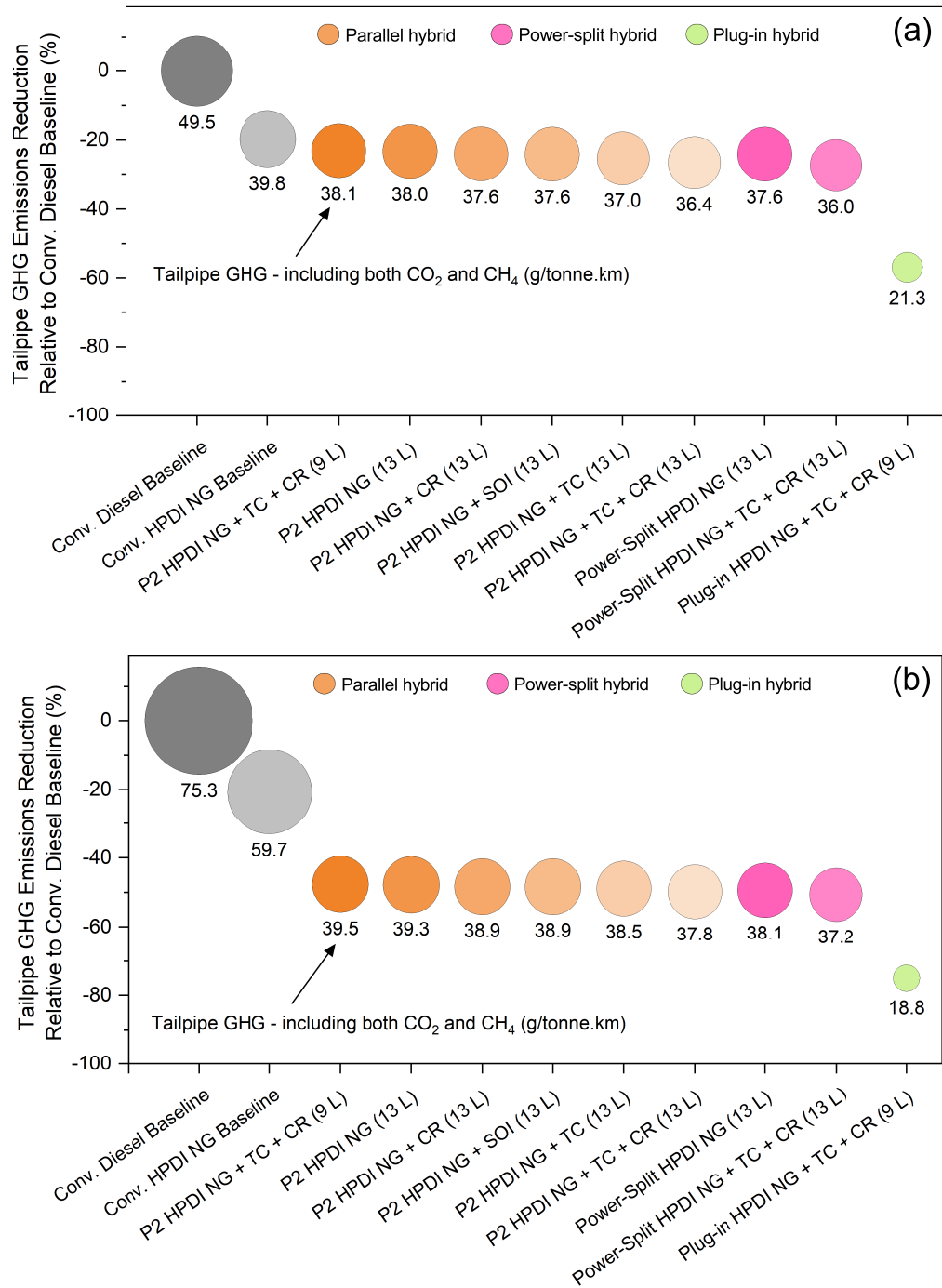


Figure 6.15: Comparison of tailpipe GHG emissions (CO₂-equivalent), including CH₄, between the conventional diesel and conventional HPDI NG baseline models, and the designed hybrid electric Class 8 truck configurations equipped with different variants of the HPDI NG engine over: (a) the Highway-2 drive cycle, and (b) the CSHVR (urban) drive cycle under 100% cargo load conditions. The CH₄ emissions are converted to CO₂-equivalent emissions using a 100-year global warming potential (GWP) factor of 29.8 [79].

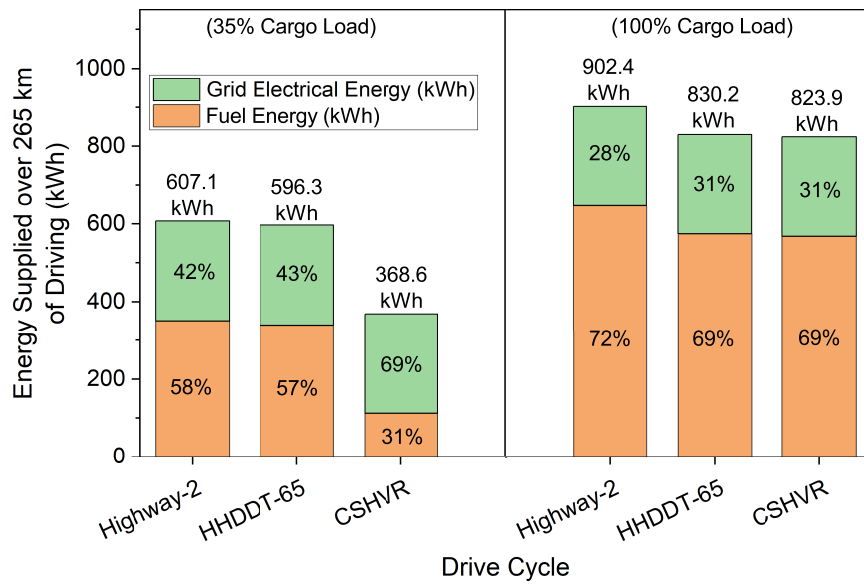


Figure 6.16: Fuel and grid-supplied electrical energy contributions to the total supplied energy of the designed plug-in hybrid Class 8 truck configuration equipped with the down-sized HPDI NG engine with turbo-compound (TC) and high compression ratio (CR), across the selected drive cycles under 35% and 100% cargo load conditions. All cases are evaluated over a 265 km driving distance, corresponding to one Highway-2 cycle length.

Chapter 7

Energy Saving and Near-Zero Greenhouse Gas Emissions in Hybrid Electric Regional-Haul Trucks Equipped with a High-Pressure Direct Injection Hydrogen Engine¹

The effects of integrating full powertrain hybridization with different variants of high-efficiency natural gas (NG)-CI engines on reducing fuel consumption and emissions in Class 8 HD trucks were evaluated in the previous chapter. This evaluation is extended in the present chapter by employing a commercially available diesel-pilot HPDI H₂ CI engine with a maximum brake thermal efficiency (BTE) of approximately 52%. This engines use a small amount of diesel fuel (1–5% of total fuel energy) as a pilot for ignition and offer 2–8% higher overall BTE than conventional diesel engines. This chapter creates new models to simulate the H₂-CI engine and the auxiliary load imposed on the engine by the on-board hydrogen compression system to maintain a 300 bar hydrogen injection pressure as on-board storage tanks are depleted. A hybrid electric powertrain is designed as a pre-transmission parallel (P₂) hybrid configuration for regional-haul HD truck applications. The individual diesel and hydrogen fuel consumptions, as well as the diesel-equivalent total fuel consumption, are compared between the conventional and P₂ hybrid models equipped with the HPDI H₂ engine. The comparison is performed over both regional-haul and urban drive cycles under different cargo load conditions. Additionally, the effect of

¹This chapter is fully based on the journal paper currently under review in [6].

powertrain hybridization on increasing the driving range and reducing the parasitic load of the on-board hydrogen compression system on the engine is quantified relative to the conventional H₂ counterpart. Finally, the reduction in tailpipe CO₂ emissions relative to the conventional diesel baseline is calculated to ensure that the hybridized configuration meets the European Union (EU) near-zero CO₂ emission threshold.

7.1 Introduction and Background

Among the potential ZEV technologies, hydrogen ICEs have gained increasing attention as a near-term solution for HD trucks [242]. Although battery electric and fuel-cell electric systems offer significant potential, challenges remain, including system cost, battery weight, performance under adverse conditions, and long-term durability [243, 244]. Given that the majority of the global Class 8 fleet still operates on conventional diesel engines, hydrogen ICEs offer a major advantage: they are compatible with existing engine platforms [48]. This enables their integration into existing engine platforms and vehicle systems with minimal modifications, rather than the development of entirely new powertrains. The availability of hydrogen refueling infrastructure, and the price of hydrogen remain key barriers to the widespread adoption of H₂-based truck technologies [245]. By improving the operational efficiency of H₂ trucks, particularly through powertrain hybridization, hydrogen consumption, operating costs, and refueling frequency are reduced.

Hydrogen has great potential to reduce GHG emissions in freight transportation, particularly for Class 8 HD trucks. Different combustion strategies for hydrogen-fueled ICEs were previously reviewed in Chapter 1 (Section 1.2.3). The analysis identified several drawbacks associated with early PFI systems, including intake charge displacement, reduced power density, susceptibility to knocking, and backfiring [95, 97]. The development of hydrogen direct injection (H₂-DI) has addressed many of these challenges, enabling improved combustion control and enhanced efficiency and emissions performance. In particular, the HPDI concept, utilizing a small amount of diesel as a pilot for ignition, has demonstrated substantial improvements in BTE ($\sim 52\%$ [98, 99]), along with high brake mean effective pressure (up to 30 bar) and specific power levels around 30 kW/L [39].

The EU proposed a ZEV threshold of less than 3 g CO₂/tonne_{cargo}.km (tailpipe) [39] for HD trucks, encompassing technologies such as battery electric, fuel-cell electric, and ICEs operating on zero-carbon fuels such as hydrogen. This threshold supports the deployment of H₂-based truck technologies. Recent studies have examined the

allowable share of carbon-based pilot fuel, expressed as the pilot energy ratio (PER), required for H_2 -CI engines to satisfy the ZEV criterion [43, 99]. Findings suggest that maintaining a relatively low PER, typically in the range of 4–6%, enables H_2 HPDI engines to meet the EU ZEV requirement for a considerable portion of HD truck applications, particularly in regional- or long-haul operations [98].

In H_2 -fueled HD trucks, hydrogen is commonly stored on-board in gaseous form within high-pressure cylinders, typically at standard filling pressures of 350 and 700 bar [246]. For tank pressures above the hydrogen injection target pressure (i.e., 300 bar), on-board compression is unnecessary. Under this condition, approximately 50% of the stored hydrogen in a 700 bar tank can be delivered directly and pressure-regulated to the target rail pressure for in-cylinder injection [39]. For tank pressures below the target hydrogen injection pressure an on-board hydrogen compression system is required to utilize more available fuel to extend the vehicle range. This compressor boosts the hydrogen from the low-pressure storage tanks to a high-pressure reservoir to maintain the hydrogen injection pressures at a stable level throughout the driving cycle [86]. Development of on-board hydrogen compression systems is still in progress in the industry, focusing on minimizing the compressor’s parasitic load on the HPDI H_2 engines while maximizing fuel utilization. Industry reports indicate that incorporating a hydrogen compressor enables utilization of approximately 75% of the stored on-board hydrogen with an associated fuel consumption penalty of less than 2% [98, 247].

As noted previously in Chapter 1 (Section 1.3.2), there is a lack of published studies on integrating powertrain hybridization with high-efficiency H_2 -CI engines to reduce fuel consumption in Class 8 HD trucks while achieving near-zero tailpipe CO_2 emissions. Only a limited number of studies have considered H_2 -SI engines (PFI or DI), which typically exhibit 2–6% lower overall BTE compared to their CI counterparts [90, 146, 173]. This chapter addresses this research gap by designing both conventional and P_2 hybrid Class 8 truck configurations equipped with an experimentally validated 13 L diesel-pilot HPDI H_2 engine. This includes developing a model to quantify the parasitic load of an on-board compression system that maintains a 300 bar hydrogen injection pressure when the storage tank pressure falls below this level. Afterward, the diesel-equivalent fuel consumption and tailpipe CO_2 emissions of the developed H_2 truck models are compared with those of the conventional diesel baseline model used in this thesis.

7.2 Conventional Hydrogen Truck Model

The Class 8 conventional diesel truck configuration used as the baseline model in this study was previously developed and validated in Chapter 2, Section 2.2. To develop the conventional hydrogen truck model, the Cummins diesel engine in the baseline configuration is replaced with a 13 L HPDI H₂ CI engine model. In addition, a dedicated real-time model is developed and integrated to estimate the parasitic load of the on-board hydrogen compression system as a function of the gaseous hydrogen pressure in the storage tanks.

7.2.1 Diesel-Pilot HPDI Hydrogen Engine Specifications

The HPDI H₂ engine considered in this study is based on the 13 L CBE1 (DC13 173 560 Euro 6) HD diesel platform developed by Scania CV AB and equipped with Cespira's HPDI fuel system. This work uses the publicly presented data [98, 99] for this engine, including BTE across different torque-speed operating points and the diesel mass flow rate used as pilot fuel for ignition over the corresponding operating range. The engine specifications are shown in Table 7.1.

Table 7.1: Key specifications of the 13 L Scania CBE1 (DC13 173 560 Euro 6) HPDI H₂ engine in this study [248].

Parameter	Description
Engine configuration	Inline six-cylinder
Max. power, kW	410 @1800 rpm
Max. brake mean effective pressure (BMEP), bar	27
Max. torque, N.m	2,800 @900-1400 rpm
Max. speed, rpm	2,400
Idle speed, rpm	600
Fuel injection system	Direct injection, dual concentric needle HPDI (300 bar H ₂ injection pressure)
Displacement per cylinder, L	2.1
Compression ratio	23:1
Bore $\times$ stroke, mm $\times$ mm	130 $\times$ 160
Aspiration	Turbocharged, Variable Geometry Turbo (VGT)
Aftertreatment system	Twin SCR (dual AdBlue dosing); DPF; ASC ¹
Dry weight, kg	1,150

¹ Ammonia slip catalyst, paired with SCR

The total fuel mass input required to deliver a given engine brake power was calculated, as shown in Eq. (7.1), using the reported BTE values and the weighted lower heating values (LHVs). The total fuel mass flow rate was partitioned between hydrogen and pilot diesel, as shown in Eq. (7.2), using PER, defined as the fraction of total

fuel energy supplied by the diesel pilot. Reported HPDI H₂ engine studies indicate that PER varies with operating condition, typically ranging between 1–5% [39, 95], with higher values observed at low engine loads. Accordingly, PER was treated as a function of engine speed and torque based on literature data, rather than assuming a constant value.

$$\dot{m}_{f,H_2} + \dot{m}_{f,diesel} = \frac{P_{brake}}{BTE \times [(1 - PER) LHV_{H_2} + PER LHV_{diesel}]} \quad (7.1)$$

$$\frac{\dot{m}_{f,diesel} \cdot LHV_{diesel}}{\dot{m}_{f,H_2} \cdot LHV_{H_2}} = \frac{PER}{1 - PER} \quad (7.2)$$

Where $\dot{m}_{f,H_2}$ and $\dot{m}_{f,diesel}$ denote the hydrogen and pilot diesel mass flow rates, respectively. LHV of hydrogen and diesel, taken as 120 MJ/kg and 42.6 MJ/kg, respectively. P_{brake} represents the engine brake power, and BTE denotes the brake thermal efficiency.

For vehicle-level modeling, a diesel-equivalent BSFC map was derived from the individual hydrogen and pilot diesel mass flow rates obtained for the HPDI H₂ engine. The hydrogen fuel contribution was converted to a diesel-equivalent basis using the ratio of fuel LHVs, as described in Eq. (7.3), and the resulting equivalent BSFC map is shown in Figure 7.1.

$$BSFC_{e,de} = \frac{\dot{m}_{f,diesel} + \dot{m}_{f,H_2} \cdot \frac{LHV_{H_2}}{LHV_{diesel}}}{P_{brake}} \quad (7.3)$$

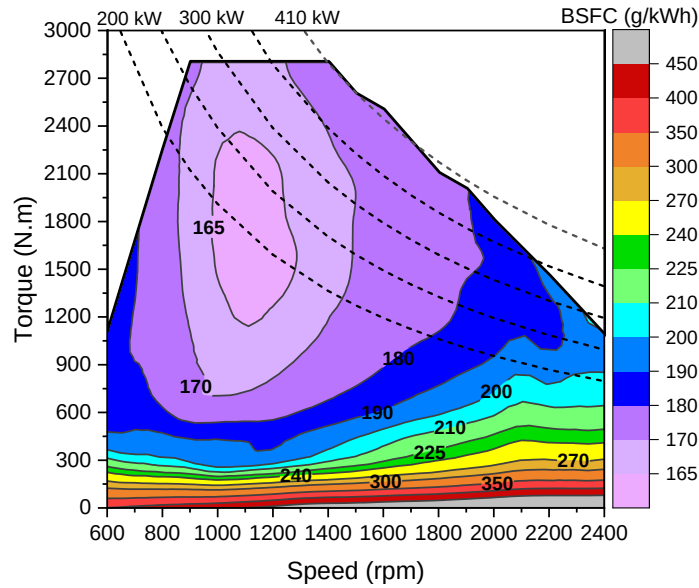


Figure 7.1: Diesel-equivalent BSFC map of the 410 kW, 13 L HPDI H₂ engine.

In addition to the BSFC map, diesel pilot fuel mass flow and corresponding CO₂ emissions maps for the HPDI H₂ engine are shown in Figure 7.2(a) and Figure 7.2(b), respectively. Since tailpipe CO₂ emissions originate exclusively from the pilot diesel injection, the CO₂ map was constructed by converting the pilot diesel mass flow rate using a carbon intensity of 3.19 kg CO₂ per kg of diesel [249].

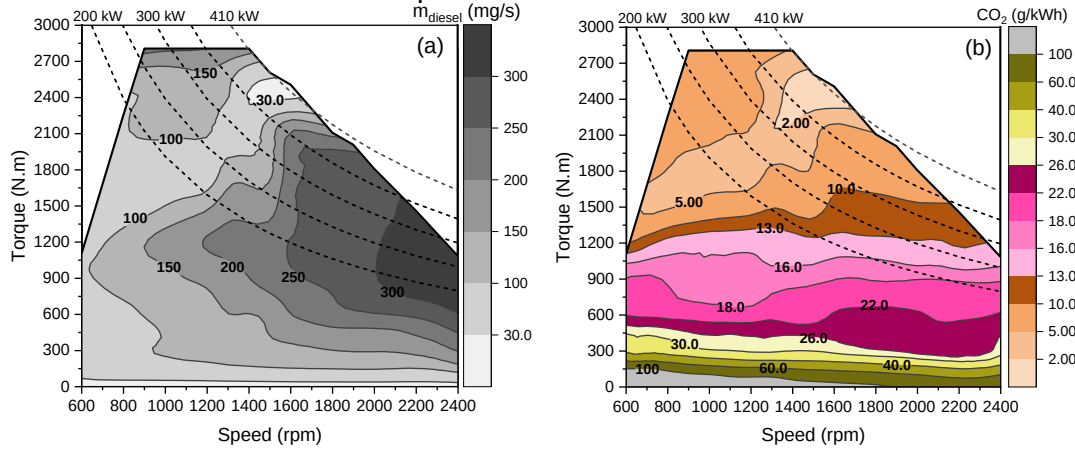


Figure 7.2: Emission characteristics of the 13 L diesel-pilot HPDI hydrogen engine in this study: (a) diesel-pilot fuel mass flow rate map, and (b) engine-out brake-specific CO₂ emissions map.

7.2.2 On-Board Hydrogen Fuel System

This section represents a real-time model to estimate the instantaneous in-tank hydrogen pressure and the corresponding power demand of the compression system required to maintain the 300 bar injection pressure of the HPDI H₂ engine across a drive cycle. The hydrogen fuel is stored in gaseous form at high pressure within Type-IV composite cylinders, with a nominal storage pressure of 700 bar. The compression system is mechanically driven by the engine crankshaft, thereby introducing an additional parasitic load on the powertrain. The resulting powertrain penalty weight of the conventional HPDI H₂ configuration is then quantified relative to the baseline diesel counterpart for an equivalent driving range of 530 km over the Highway-2 drive cycle.

The model of the on-board hydrogen compression system for the conventional hydrogen truck model in this study was previously developed by the collaborative research team at Simon Fraser University as part of a joint project. Detailed design and validation of the model can be found in [86]. A schematic of the on-board hydrogen compression system coupled to the engine crankshaft is illustrated in Figure 7.3. For tank pressures above the target hydrogen injection pressure (300 bar), hydrogen

is regulated and supplied to the engine fuel rail via a high-pressure line that bypasses the compressor. For tank pressures below 300 bar, the remaining hydrogen on-board is boosted to 300 bar using two hydraulically-driven single-stage gas boosters with free-floating pistons, working in parallel. Under this operating mode, the control valve in the high-pressure bypass line is closed, while the valves upstream of the compressor and downstream of the high-pressure gas reservoir are opened, enabling hydrogen compression and delivery to the engine via the compressor flow path. The compressor is actuated by a variable-displacement hydraulic pump mechanically coupled to the engine crankshaft. The pump displacement is modulated over the driving cycle to regulate compressor cycle rate, thereby matching the engine's hydrogen demand at each operating point as a function of engine speed and the inlet gas density. However, a high-pressure buffer gas reservoir, upstream of the engine, is employed to attenuate fuel-line pressure pulsations and support short-term transient demands.

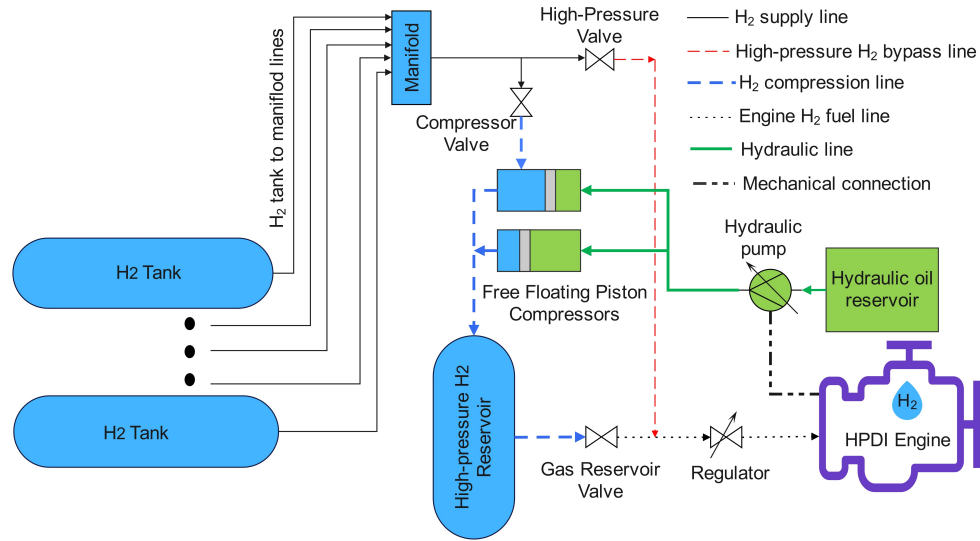


Figure 7.3: Schematic of on-board hydrogen storage and compression systems connected to the 13 L HPDI H₂ engine.

For vehicle modeling purposes, a compressor-specific work input map, parameterized by the storage tank pressure and summarized in Table 7.2, is employed to estimate the compressor power input. This map is based on testing and simulation of a hydraulically driven gas compressor [86], and the corresponding power demand is calculated as follows:

$$P_{\text{pump}} = \dot{m}_{\text{f,H}_2} e_{\text{p,input}} \quad (7.4)$$

where P_{pump} and $e_{\text{p,input}}$ denote the hydraulic pump parasitic power input, and

Table 7.2: Compressor specific work input as a function of tank pressure. This map was provided by the collaborative research team at the Simon Fraser University and further information can be found in [86]

Tank Pressure (bar)	Pressure ratio	Compressor specific work input (MJ/kg of H ₂)
> 300	—	0
300	1:1	1.8
150	2:1	3.0
100	3:1	4.0
75	4:1	4.8
50	6:1	6.3
30	10:1	8.6

the specific work input per unit mass of compressed hydrogen at each time step over the drive cycle. $\dot{m}_{f,H_2}$ corresponds to the hydrogen fuel mass flow rate injected to the HPDI H₂ engine at each time step. For tank pressures above 300 bar, the compressor is bypassed and the compression specific work is taken as zero (Table 7.2). As the tank pressure decreases, the required pressure ratio increases and the specific work input rises accordingly. A non-zero specific work is also considered at an effective pressure ratio of 1:1, attributable to losses in the hydraulic drive even when no net gas pressure rise is produced. The calculated P_{pump} in each time step of the drive cycle is incorporated into the total mechanical accessory power to determine the overall auxiliary energy consumption of the conventional hydrogen configuration. The model used to calculate the total auxiliary load is identical to that previously presented for the conventional diesel baseline model in Eq. (2.7) in Chapter 2 of this thesis.

To obtain the specific work input of the compressor from Table 7.2, the instantaneous pressure of the on-board hydrogen tank is calculated at each time step as follows:

$$P_{\text{tank},i} = \frac{m_{\text{tank},i} Z(P_{\text{tank},i}, \Theta_{\text{tank},i}) R_g \Theta_{\text{tank},i}}{V_{\text{tank}}} \quad (7.5)$$

where $m_{\text{tank},i}$ is the remaining hydrogen mass in the tank at each time step, V_{tank} is the tank internal volume, R_g is the gas constant, and Θ_{tank} is the in-tank hydrogen temperature, assumed to be equal to the ambient temperature of 24 °C. The compressibility factor Z of hydrogen depends on the in-tank gas pressure and temperature at each time step, and is obtained from the Redlich–Kwong (RK) equation of state to account for real-gas effects [250].

In the conventional diesel baseline model, as previously detailed in Table 2.2 of Chapter 2, the combined weight of the engine and fuel system—including the diesel

fuel and storage tanks—is 2198 kg. The capacities and weights of the diesel and hydrogen tanks in the conventional hydrogen configuration are determined to achieve the driving range of 530 km over the Highway-2 cycle while maintaining the same maximum cargo capacity as the baseline diesel configuration. Since diesel contributes no more than 5% of the total fuel energy in the HPDI H₂ engine across a wide range of operating conditions, the diesel fuel capacity is set to 5% of that in the conventional diesel baseline model, resulting in 40 L. Under this assumption, a total of 72.2 kg of compressed hydrogen is required, with the additional constraint that the in-tank hydrogen pressure does not fall below 75 bar at the end of the specified driving range. This minimum tank pressure limits the maximum pressure ratio to 4:1 in the on-board compressor to meet the design constraints.

Incorporating the HPDI H₂ engine into the Class 8 conventional truck model, and the addition of the hydrogen fuel supply system—including pressurized storage tanks and the on-board compression system—adds 732 kg of vehicle mass relative to the baseline diesel configuration (Table 7.3). Commercially available cylindrical Type-IV hydrogen storage tanks (700 bar) typically store 9–10 kg of hydrogen with a gravimetric capacity (hydrogen mass to cylinder mass ratio) of ~5% [246]. Based on this specification, the required 72.2 kg of on-board hydrogen storage is distributed across eight tanks, resulting in an estimated empty mass of 180.5 kg per tank. The total mass of the hydrogen compression system is assumed to be 210 kg [99], consisting of 135 kg for the compressors and 75 kg for the gas reservoir, hydraulic pump, valves, sensors, and piping.

Table 7.3: Weight specifications of the engine, fuel, fuel storage tanks, and on-board fuel compression system for the Class 8 conventional diesel and HPDI H₂ truck configurations in this study, resulting difference in fully loaded vehicle mass.

Conventional Truck model	Engine Dry Weight	Fuel Weight	Fuel Tank Weight	H ₂ Compression System Weight	Fully Loaded Vehicle Mass
Conv. Diesel	1,430 kg	Diesel: 668 kg	2 diesel tanks each 50 kg	-	39,500 kg
Conv. HPDI H ₂ *	1,150 kg	Diesel: 33.4 kg H ₂ : 72.2 kg	Diesel: 20 kg (one tank) H ₂ : 1,444 kg (8 tanks) **	210 kg ***	40,232 kg

* The hydrogen engine is based on the 13 L Scania platform and is not derived from the Cummins X15 diesel engine.

** Cylindrical Type-IV carbon-fiber hydrogen storage tanks, each with a water-equivalent volume of 226.9 L.

*** Includes the weight of the free-floating piston compressors, gas reservoir, hydraulic pump, valves, sensors, and piping.

The disengagement strategy of the crankshaft-driven hydrogen compression system in the conventional HPDI H₂ model is defined based on engine operating conditions

to ensure stable and efficient powertrain performance, as detailed in Table 7.4. Compressor operation is restricted outside the allowed engine speed range to prevent idle instability and excessive high-speed loading. It is also deactivated during in-gear deceleration when the driver torque request is zero, avoiding unnecessary parasitic losses under fuel cut-off conditions. Additionally, the compressor is disengaged during gear shifts to minimize torque disturbances and preserve tractive performance.

Table 7.4: Summary of disengagement logic for the crankshaft-driven hydrogen compression system in the Class 8 conventional HPDI H₂ regional-haul truck model.

ID	Operating Conditions	Description
1	$\omega_e \leq 650 \text{ rpm} \vee \omega_e \geq 2250 \text{ rpm}$	Allowable operating speed limits for the hydraulically driven pump
2	$\omega_e > 650 \text{ rpm} \wedge T_{\text{drv,req}} = 0$	Fuel cut-off during in-gear deceleration (driver torque request equals zero)
3	$\omega_e > 650 \text{ rpm} \wedge u_c = 0$	Gear-shifting events when the clutch is disengaged

7.3 Hybridization of the HPDI Hydrogen Engine for Class 8 Regional-Haul Truck Applications

The Class 8 conventional HPDI H₂ powertrain is hybridized in this section using a P₂ hybrid electric configuration. This offers additional reductions in fuel consumption and tailpipe GHG emissions by jointly leveraging powertrain electrification and a near-zero-carbon H₂-ICE. The P₂ hybrid configuration is selected as it offers a favorable balance between cost, complexity, and performance compared to other potential hybrid architectures for HD trucks.

The P₂ hybrid electric configuration is designed using the same engine as the conventional hydrogen model. To this end, the diesel-equivalent fuel consumption and CO₂ emissions maps of the 13 L HPDI H₂ engine are incorporated into the engine model of the P₂ hybrid configuration previously developed in Chapter 3, Section 3.2. Additionally, the same on-board hydrogen fuel system model used in the conventional hydrogen configuration is implemented in the hybridized configuration. The total energy consumption of the auxiliary systems in the P₂ hybrid electric design is calculated following the same approach as that used for the conventional hydrogen model, with the following modifications. The hydraulic pump of the on-board hydrogen compression system is the only mechanically driven accessory load, and its power demand is determined based on Eq. (7.4). All other mechanical accessory loads present in the conventional model are replaced with a constant 3 kW electrical auxiliary load, as previously discussed in Chapter 3, Section 3.2.1. The powertrain

configuration of the P_2 hybrid electric design equipped with the HPDI H_2 engine is shown in Figure 7.4. Except for the aforementioned modifications to the engine and on-board fuel system, all other component specifications and models, as well as the EMS control strategy, are identical to those previously developed for a generic P_2 hybrid configuration in Chapter 3.

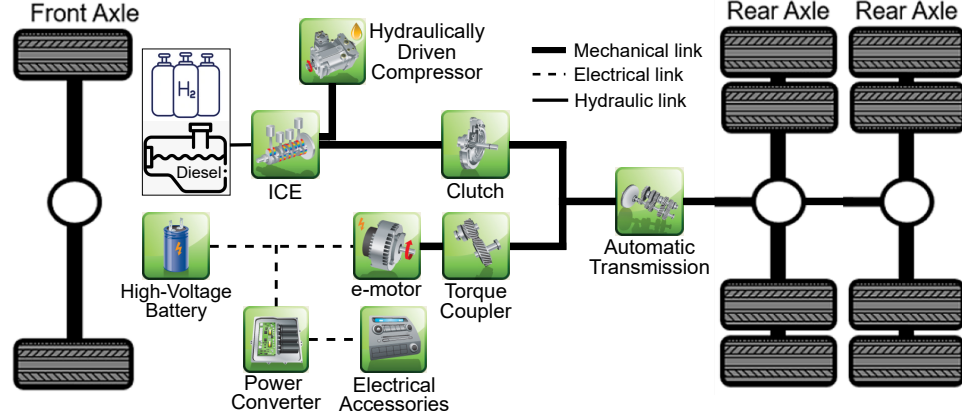


Figure 7.4: Powertrain configuration of the designed pre-transmission parallel (P_2) hybrid Class 8 regional-haul truck equipped with the 13 L HPDI H_2 engine.

To determine the optimal sizing of the powertrain components for the designed P_2 hybrid electric configuration, the multi-stage optimization framework detailed in Chapter 4 (Section 4.3) is employed. The primary optimization objective is to minimize the fuel consumption of the hybrid configuration over the Highway-2 drive cycle while maintaining propulsion system performance comparable to that of the conventional diesel baseline model. The capability of the designed conventional diesel and HPDI H_2 truck models to satisfy the proposed performance criteria, as specified by the SAE J2807 standard [165], is summarized in Table 7.5. To account for the parasitic load of the on-board hydrogen compression system in the conventional HPDI H_2 configuration when evaluating compliance with the performance criteria, the initial total on-board hydrogen mass is set to 35.8 kg across the eight storage tanks. This initial condition corresponds to the pressure of ~ 300 bar in each hydrogen tank at the temperature of 24 °C.

The variation ranges of the critical DOE variables, as part of the optimal component sizing process for the P_2 hybrid configuration, are summarized in Table 7.6. These ranges are determined based on the preliminary component sizing procedure previously detailed in Chapter 4, Section 4.3.1. The optimal sizes of the primary powertrain components for the P_2 hybrid electric truck configurations equipped with the 13 L HPDI H_2 engine are determined and summarized in Table 7.7.

Table 7.5: Performance capability of the Class 8 conventional diesel and conventional HPDI H₂ truck configurations designed in this study aligned with the SAE J2807 standard [165].

Class 8 Truck Configuration	Purpose	Driving Range per Refueling (km)	0 → 48 km/h Acceleration Time (s)	0 → 96 km/h Acceleration Time (s)	Flat Road Top Speed (km/h)	Gradeability: Max. Carryable Cargo Load along Davis Dam Road (kg)
Conv. Diesel	Regional-haul	530 (2×Highway-2)	16	64	122	8,044 (34.0% of total cargo capacity)
Conv. HPDI H ₂	Regional-haul	530 (2×Highway-2)	14	62	126	8,870 (37.5% of total cargo capacity)

* Driving range, level road acceleration, and top speed criteria must be satisfied when the truck is fully loaded.

Table 7.6: Variation range of the critical DOE variables within the optimal component sizing framework for the designed Class 8 pre-transmission parallel (P₂) hybrid electric truck configuration equipped with the 13 L HPDI H₂ engine.

Variables (Unit)	Index	Variation Range	Increment Change
Final drive ratio (-)	n_{fd}	2-6	0.1
e-motor rated power (kW)	P_m	115-304	1
Torque coupler ratio (-)	n_{tc}	1-4	0.1
Number of parallel battery modules (-)	$N_{batt,p}$	5-65	1
Engine rated power (kW)	P_e	Constant 410	-
Engine efficiency threshold for EMS (%)	δ_e	0-0.03	0.001
Low battery SOC threshold for EMS (%)	SOC _L	0.2-0.6	0.01
High battery SOC threshold for EMS (%)	SOC _H	0.5-0.9	0.01

The powertrain penalty weight of the designed P₂ hybrid configuration, relative to the conventional diesel baseline model, is estimated to be 877 kg following the procedure described in Chapter 4, Section 4.3.2. The combined weight of the engine and the on-board hydrogen fuel system is assumed to be the same as that of the conventional hydrogen model, as previously described in Table 7.3. To improve the ability of the P₂ hybrid electric design equipped with a six-speed gearbox to maintain the engine operation within the efficient speed range of 1100–1450 rpm, two distinct final drive ratios are selected for highway and urban driving. One ratio is optimized to minimize fuel consumption over the Highway-2 cycle, while the other is optimized to achieve the same objective over the CSHVR cycle.

In the designed Class 8 truck configurations equipped with the HPDI H₂ engine, both fuel consumption and GHG emissions are influenced by the driving range per refueling. Longer driving ranges lead to greater hydrogen pressure depletion in the storage tanks, which in turn increases the auxiliary power demand of the on-board hydrogen compression system. Accordingly, the proposed conventional and hybridized truck models are simulated over two consecutive repetitions of the Highway-2 cycle, corresponding to a total driving range of 530 km. For each design, an equivalent driving range is obtained over the HHDDT-65 and CSHVR cycles by applying a

Table 7.7: Component specifications of the conventional diesel, conventional hydrogen, and pre-transmission parallel (P₂) hybrid Class 8 truck configurations in this study. The conventional diesel model uses the Cummins X15 engine, while the other models use the 13 L HPDI H₂ engine.

Component/Module		Conventional Diesel Baseline	Conventional HPDI H ₂	Parallel HPDI H ₂
Engine	Type	Cummins X15 diesel	13 L HPDI H ₂	13 L HPDI H ₂
	Max. power (kW)	336	410	410
	Max. torque (N.m)	2330	2800	2800
	Max. speed (rpm)	1900	2400	2400
Fuel system	Type	Single fuel diesel	Dual-fuel H ₂ -diesel	Dual-fuel H ₂ -diesel
	Per tank capacity (kg)	334 (diesel)	diesel: 33.4 & H ₂ : 9.03	diesel: 33.4 & H ₂ : 9.03
	Number of tanks	2	diesel: 2 & H ₂ : 8	diesel: 2 & H ₂ : 8
Traction motor	Max. power (kW)	-	-	162 (@1100 rpm)
	Max. torque (N.m)	-	-	1290
	Max. speed (rpm)	-	-	5550
High-voltage battery	Type	-	-	Li-Ion
	Number of parallel modules	-	-	18
	Number of series cells	-	-	98
	Single cell capacity (Wh)	-	-	15.8
	Total capacity * (kWh)	-	-	27.9
Multi-speed	Type	10-Speed manual	10-Speed manual	6-Speed automatic
transmission	Gear ratios	Table 2.2	Table 2.2	$[n_1, n_2, n_3, n_4, n_5, n_6] = [6.84, 3.92, 2.41, 1.53, 1, 0.85]$
Power	Voltage range (V)	-	-	12 const.
converter	Power range (kW)	-	-	0 to 3.1
Torque coupler	Single-gear ratio	-	-	2.1
Final drive ratio	-	2.85	2.80	Highway: 2.6 Urban: 3.0
Constrained on/off EMS	Engine efficiency threshold (δ)	-	-	0.012
	Low battery SOC threshold (SOC _L)	-	-	0.31
	High battery SOC threshold (SOC _H)	-	-	0.56
Powertrain penalty weight				
compared to the	-	-	+732 kg	+877 kg
Conv. diesel baseline				

* The total battery capacity spans an SOC range of 0% to 100%, while the usable capacity is limited to the range of 20% to 90% SOC.

sufficient number of cycle concatenations to match the total fuel energy consumption of the Highway-2 cycle under identical cargo load conditions.

7.4 Results and Discussions

The Class 8 P₂ hybrid electric configuration equipped with the HPDI H₂ engine was designed and optimized to minimize fuel consumption while achieving a 530 km driving range over the Highway-2 drive cycle. This section presents the reductions in fuel consumption and tailpipe CO₂ emissions achieved by the hybridized configuration relative to the conventional diesel and HPDI H₂ configurations across different drive cycles and cargo load conditions. It also examines the impact of powertrain hybridization on the auxiliary power demand of the on-board hydrogen compression system associated with the HPDI H₂ engine. In addition, the engine operating conditions and the distribution of energy flows among the primary powertrain components are analyzed for the designed conventional and hybridized truck configurations.

7.4.1 Fuel Conversion Efficiency and Tailpipe GHG Emissions

The diesel-equivalent fuel consumption results, expressed in L/tonne_{cargo}·100 km, for the designed conventional and hybrid electric truck models across the selected drive cycles and three cargo load conditions are shown in Figure 7.5. For the conventional models, the HPDI H₂ engine exhibits 10.2% to 15% reductions in equivalent fuel consumption relative to the diesel engine during highway driving (Highway-2 and HHDDT-65), and 21.1% to 21.6% reductions over urban driving (CSHVR), under different cargo loads. These substantial reductions result from a higher brake thermal efficiency (BTE) of the Scania HPDI H₂ engine compared to the Cummins X15 diesel engine over a wide torque range and within the active speed region of 1000–1400 rpm. The higher BTE could be due to a higher compression ratio (CR) and combustion characteristics enabled by late-cycle hydrogen injection near top dead center, which introduces a molar expansion effect and recovers part of the compression work during the expansion stroke. In addition, hydrogen’s high flame speed promotes rapid and well-phased heat release, effectively increasing the expansion ratio and enhancing thermal efficiency across different torque ranges.

The pre-transmission parallel (P₂) hybrid electric configuration, equipped with the 13 L HPDI H₂ engine, exhibits lower equivalent fuel consumption than its conventional counterparts across all selected drive cycles and cargo load conditions (Figure 7.5).

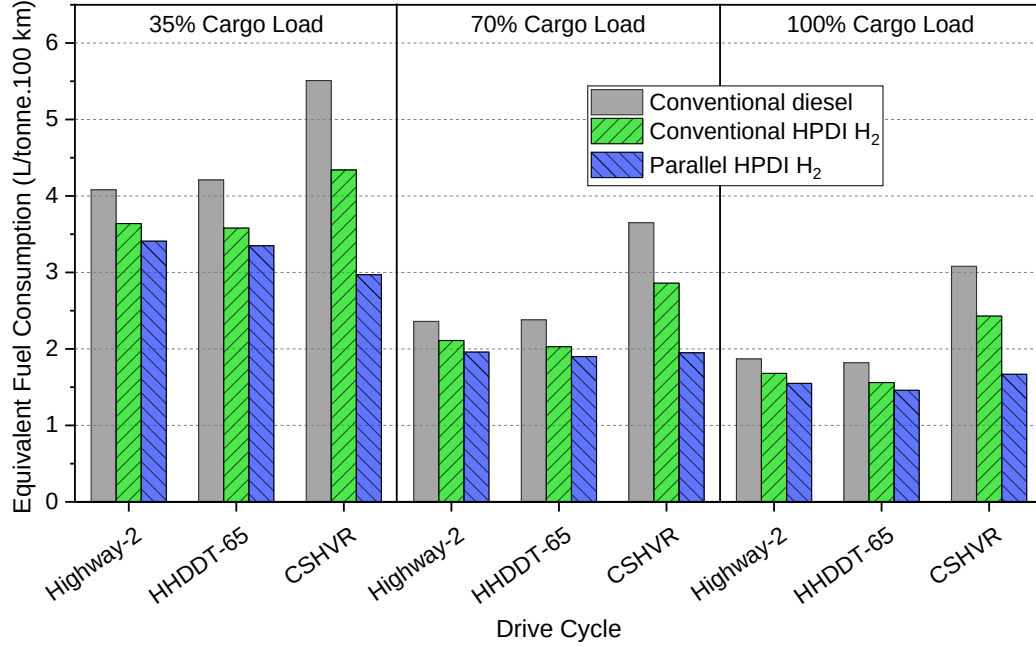


Figure 7.5: Diesel-equivalent fuel consumption of the conventional diesel, conventional HPDI H₂, and pre-transmission parallel (P₂) hybrid Class 8 regional-haul truck models over three HD truck drive cycles under different cargo loads. Note that the conventional diesel model uses the Cummins X15 engine, while the other models use the 13 liter HPDI hydrogen engine.

Relative to the conventional HPDI H₂ model, fuel consumption reductions range from 6.3% to 7.7% over the highway cycles (Highway-2 and HHDDT-65) as the cargo load increases from 35% to 100%. Over the urban cycle (CSHVR), the reductions are more pronounced, reaching up to 31.8%. In the parallel hybrid design, a portion of the braking energy dissipated at the wheels is recovered and stored in the battery and, together with energy supplied by the engine, is used to sustain the battery state of charge (SOC) over the trip. During acceleration, peak power demand, and low-speed operations, the battery energy is used by the e-motor to either assist the engine or propel the vehicle independently, enabling more frequent operation of the internal combustion engine (ICE) within its high-efficiency torque-speed region (i.e., BSFC $\leq$ 165 g/kWh). These operating features lead to reduced fuel consumption in the hybrid electric design compared to the conventional counterpart using the same engine, with more pronounced benefits over urban cycles including greater regenerative braking availability.

The time-averaged engine BTEs, evaluated during engine-on operation, are shown in Figure 7.6 for the designed conventional and hybridized truck models across the considered drive cycles and cargo load conditions. Across all cases, hybridization im-

proves the average engine BTE compared to conventional models. Over the highway cycles (Highway-2 and HHDDT-65), the parallel hybrid configuration achieves 1.5% to 2.4% higher average engine BTE than the conventional HPDI H₂ model as the cargo load decreases from 100% to 35%. The improvement is more substantial as 1.9% to 4.1% over the urban cycle (CSHVR). As the tractive demand at the wheels decreases, due to less power-demanding urban driving or reduced cargo load—the ICE operation of conventional trucks shifts toward lower torque regions associated with lower BTE. In contrast, in the parallel hybrid configuration the ICE can either operate within its narrow optimal efficiency region (i.e., BSFC $\leq$ 165 g/kWh) or be turned off, with the electrical driveline (battery and e-motor) providing the required traction during low-efficiency operating conditions. Hence, the average engine BTE improvements of the hybridized configuration compared to the conventional counterparts become more pronounced under low cargo load conditions and over urban cycles.

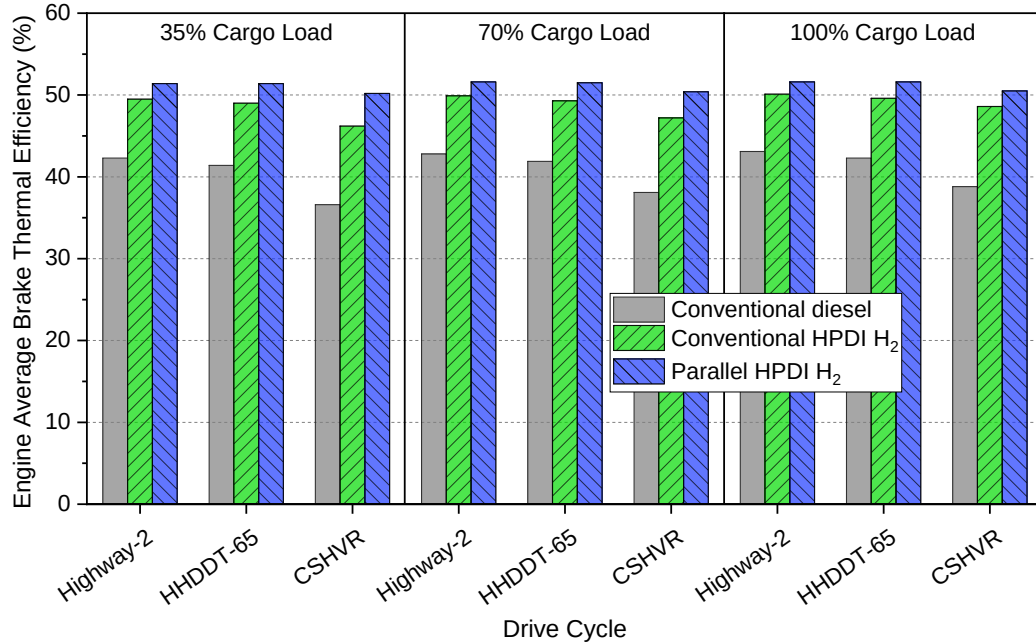


Figure 7.6: Engine average brake thermal efficiency of the conventional diesel, conventional HPDI H₂, and pre-transmission parallel (P₂) hybrid Class 8 regional-haul truck models over three HD truck drive cycles under different cargo loads. Note that the conventional diesel model uses the Cummins X15 engine, while the other models use the 13 liter HPDI hydrogen engine.

According to the European regulatory definition for zero-emission HD trucks, tailpipe (tank-to-wheel) CO₂ emissions must be below 3 g/tonne_{cargo}.km [39]. Although the conventional HPDI H₂ model achieves a 96.3% to 98.5% reduction in

tailpipe CO₂ emissions relative to the conventional diesel counterpart, its absolute emissions still exceed the 3 g/tonne_{cargo}.km threshold under low cargo load conditions (Table 7.8). In contrast, the parallel HPDI H₂ model achieves a larger reduction of 98.4% to 99.1% and remains below the regulatory threshold across all the evaluated conditions. Tailpipe CO₂ emissions from the HPDI H₂ engine correlate directly with the amount of diesel fuel consumed, as hydrogen combustion produces no carbon-based tailpipe emissions. As driving conditions become less power-demanding, due to reduced cargo load or urban operation, the average diesel pilot energy ratio (PER) in the HPDI H₂ engine increases (Table 7.8). Under these conditions, the engine operates more time in low-torque regions (i.e., $T < 0.5 T_{\max}$), resulting in higher CO₂ emissions, as demonstrated in Figure 7.2(b). By modifying the engine operating points, a hybrid electric powertrain can help to reduce average PER by limiting low-torque operation that leads to high CO₂ emissions.

Table 7.8: Average energy share of diesel fuel per total injected fuel, mass consumption of diesel and/or hydrogen fuel(s), and engine-out CO₂ emissions among the designed Class 8 regional-haul truck configurations over three HD truck drive cycles under different cargo loads.

Class 8 Truck Configuration	Highway-2			HHDDT-65			CSHVR		
	35% Cargo	70% Cargo	100% Cargo	35% Cargo	70% Cargo	100% Cargo	35% Cargo	70% Cargo	100% Cargo
Average diesel pilot energy ratio* (%)									
Conv. diesel	100	100	100	100	100	100	100	100	100
Conv. HPDI H ₂	2.91	2.10	1.69	3.32	2.42	1.93	4.75	3.38	2.69
Parallel HPDI H ₂	1.60	1.32	1.24	1.97	1.54	1.41	2.37	2.04	1.71
Diesel and/or Hydrogen Fuel Mass Consumption (kg/100 km)									
Conv. diesel	Diesel: 28.23	Diesel: 32.65	Diesel: 36.85	Diesel: 29.12	Diesel: 32.84	Diesel: 35.98	Diesel: 38.08	Diesel: 50.42	Diesel: 60.93
Conv. HPDI H ₂	Diesel: 0.73 H ₂ : 8.67	Diesel: 0.61 H ₂ : 10.16	Diesel: 0.56 H ₂ : 11.58	Diesel: 0.82 H ₂ : 8.49	Diesel: 0.68 H ₂ : 9.72	Diesel: 0.60 H ₂ : 10.76	Diesel: 1.43 H ₂ : 10.14	Diesel: 1.34 H ₂ : 13.55	Diesel: 1.29 H ₂ : 16.56
Parallel HPDI H ₂	Diesel: 0.38 H ₂ : 8.23	Diesel: 0.36 H ₂ : 9.52	Diesel: 0.38 H ₂ : 10.74	Diesel: 0.46 H ₂ : 8.06	Diesel: 0.40 H ₂ : 9.16	Diesel: 0.41 H ₂ : 10.06	Diesel: 0.49 H ₂ : 7.13	Diesel: 0.55 H ₂ : 9.40	Diesel: 0.56 H ₂ : 11.51
Tailpipe CO ₂ Emissions (g/tonne.km)									
Conv. diesel	107.38	62.10	49.06	110.78	62.48	47.91	144.86	95.90	81.12
Conv. HPDI H ₂	2.78	1.17	0.75	3.12	1.29	0.79	5.42	2.54	1.72
Parallel HPDI H ₂	1.44	0.68	0.50	1.74	0.77	0.54	1.86	1.05	0.75

* Time-averaged energy share of diesel fuel per total injected fuel.

Powertrain hybridization reduces the individual mass consumption of diesel and hydrogen fuels in the HPDI H₂ engine relative to the conventional HPDI H₂ model across all evaluated drive cycles and cargo load conditions (Table 7.8). For hydrogen fuel, expressed in kg/100 km, the reductions range from 5.1% to 7.3% over the highway cycles (Highway-2 and HHDDT-65) and increase to as much as 30.5% over the urban cycle (CSHVR). These reductions result from the higher net powertrain energy-conversion efficiency of the parallel hybrid design, enhanced by greater regenerative

braking energy recovery and a larger engine BTE advantage over the conventional counterparts.

7.4.2 Hybridization Effects on Engine Operating Points

In this section, the ICE operating points and their distribution on the engine BSFC maps are analyzed among the three Class 8 regional-haul truck designs. The analysis highlights the ability of the P₂ hybrid configuration to improve the average engine BTE compared to the conventional counterparts, irrespective of variations in cargo load and drive cycles.

Despite the presence of an electrical power source (high-voltage battery) in the P₂ hybrid electric configuration, the ICE remains the primary source of power generation. The limited availability of regenerative braking energy over highway drive cycles increases reliance on the ICE to provide battery charging and maintain charge-sustaining operation throughout the trip. One of the main advantages of the parallel hybrid architecture is the partial decoupling of engine torque from the driving wheels, allowing the engine to operate more frequently within its high-efficiency region. Figure 7.7 shows the ICE operating points overlaid on the engine BSFC contour maps for the three designed configurations across the selected drive cycles under 100% cargo load, along with the corresponding time-averaged percentages of operation within different BSFC regions. In the conventional configurations, engine operation spans a wide range, from low-torque with high BSFC to high-torque with low BSFC values. In contrast, the P₂ hybrid configuration concentrates most of the engine operations within higher torque regions with lower BSFC values (i.e., $\text{BSFC} \leq 165 \text{ g/kWh}$), resulting in an improved average engine BTE. The involvement of the electrical driveline, including the battery and e-motor, to provide propulsion when the ICE would otherwise operate outside its high-efficiency region reduces engine operation in low-torque regimes. As shown in Figure 7.7, 86.8% to 88.4% of engine operation for the parallel HPDI H₂ design occurs within the high-efficiency region ($\text{BSFC} \leq 165 \text{ g/kWh}$) over the highway cycles (Highway-2 and HHDDT-65), whereas this range is significantly lower, at 34.2% to 36.6%, for the conventional HPDI H₂ design.

In the conventional configurations, where the ICE is the sole power source, reductions in drive cycle power demand shift engine operation toward lower torque regions, associated with lower BTEs. Drive cycle power demand decreases either due to a transition from highway to urban cycles or a reduction in cargo load. Conversely, for the P₂ hybrid configuration, shifting toward less power-demanding driving conditions results in only a negligible reduction in the percentage of engine operation within the

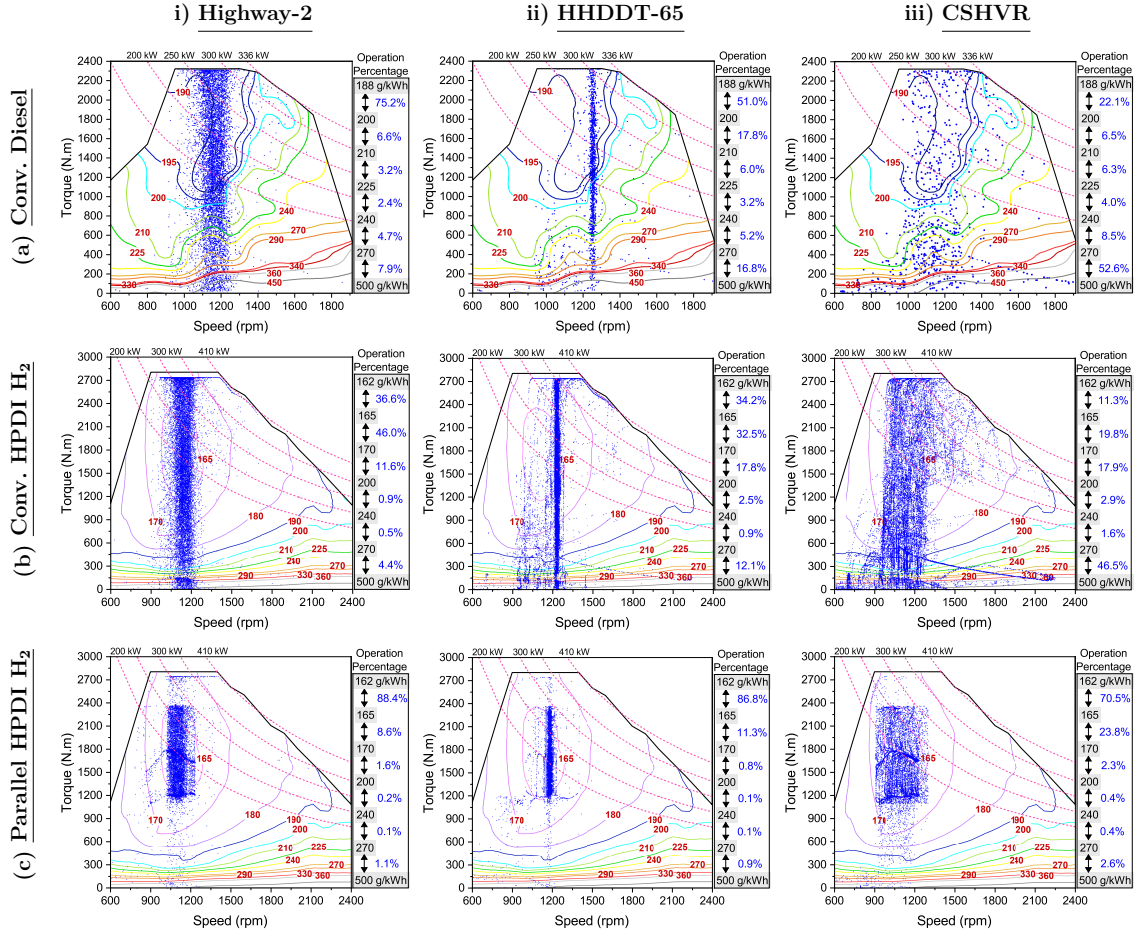


Figure 7.7: Weighted engine operations on BSFC (diesel equivalent) contour plots over the three drive cycles: (i) Highway-2, (ii) HHDDT-65, and (iii) CSHVR, evaluated for the designed Class 8 regional-haul truck configurations at 100% cargo load. Sub-figures correspond to: (a) conventional diesel, (b) conventional H₂, and (c) pre-transmission parallel (P₂) hybrid powertrains. Each point represents the ICE operating state during engine-on periods over a one-second interval. The percentage tables shown beside each sub-figure indicate the fraction of engine operating time spent within the corresponding BSFC ranges. The conventional diesel model uses the Cummins X15 engine, while the other models use the 13 liter HPDI hydrogen engine.

high-efficiency region. Hence, as drive cycle power demand decreases, the reduction in average engine BTE of the conventional configurations relative to the hybridized configuration becomes more pronounced. For example, when transitioning from the Highway-2 cycle to the CSHVR cycle under 100% cargo load, the time-averaged percentage of engine operation within the low-torque region (i.e., $T < 0.4 T_{\max}$) for the conventional HPDI H_2 model increases substantially from 26.7% to 72.3% (Table 7.9). However, this increase is marginal for the parallel HPDI H_2 model, rising only from 3.2% to 8.3%. Similarly, reducing the cargo load of the conventional HPDI H_2 model from 100% to 35% leads to a 16.6% increase in the percentage of engine operation within the low-torque region over the Highway-2 cycle and a 12% increase over the CSHVR cycle (Table 7.9). While the corresponding increases for the parallel hybrid model are only 0.6% and 0.5%, respectively. Appendix D provides the weighted ICE operating points among the designed Class 8 truck configurations over the selected drive cycles when the cargo load decreases to 35%.

Table 7.9: Time-averaged percentage of ICE operation (when is on) within low-torque region, and time-averaged percentage of engine-on duration for the designed Class 8 regional-haul truck configurations across the selected drive cycles under 35% and 100% cargo load conditions.

Class 8 Truck Configuration	Highway-2		HHDDT-65		CSHVR	
	35% Cargo	100% Cargo	35% Cargo	100% Cargo	35% Cargo	100% Cargo
Time-Averaged Percentage of Engine Operation* within Low-Torque Region ($T < 0.4 T_{\max}$) (%)						
Conv. diesel ($T < 932$ N.m)	52.9	30.4	67.4	43.6	86.0	76.0
Conv. HPDI H_2 ($T < 1120$ N.m)	43.6	26.7	54.7	37.1	84.3	72.3
Parallel HPDI H_2 ($T < 1120$ N.m)	3.8	3.2	2.4	2.2	8.8	8.3
Time-Averaged Percentage of Engine-On Duration (%)						
Conv. diesel	100	100	100	100	100	100
Conv. HPDI H_2	100	100	100	100	100	100
Parallel HPDI H_2	81.0	84.9	77.8	79.4	27.6	37.1

* Percentages of engine operation within the specified torque region are calculated during periods the ICE is on.

In hybridized powertrains, the ICE can be shut down when the high-voltage battery has sufficient charge and the e-motor alone can satisfy the road power demand. As a result, engine turn-on duration over the trip is reduced, and when active, the engine operates predominantly within its high-efficiency region. Table 7.9 presents the time-averaged percentage of engine-on duration for the designed truck configurations across the selected drive cycles and cargo loads. The engine-on duration in the parallel hybrid design decreases as the drive cycle becomes less power-demanding and the availability

of regenerative braking energy increases, as observed in the CSHVR cycle. Reducing the cargo load from 100% to 35% over the urban cycle (CSHVR) results in a 9.5% reduction in engine-on duration for the parallel HPDI H₂ configuration, whereas this reduction is less significant (up to 3.9%) over the highway cycles (Highway-2 and HHDDT-65). The reduction in road power demand resulting from a decrease in cargo load is more pronounced over the CSHVR cycle, which features lower average speeds and no road grade profile, compared to the more power-demanding highway cycles. Consequently, a larger portion of the CSHVR cycle can be propelled by the e-motor in all-electric mode, leading to a shorter engine-on duration.

Overall, improving the average engine BTE by operating predominantly within the high-torque region associated with lower BSFC values in the hybridized powertrain equipped with the HPDI H₂ engine leads to a corresponding reduction in both hydrogen and diesel fuel consumption. Reduced diesel consumption, together with the ability to shut down the engine during portions of the drive cycle, directly contributes to lower carbon-related tailpipe emissions, such as CO₂.

7.4.3 Parasitic Load of the On-Board Hydrogen Compression System

This section evaluates the auxiliary load imposed by the on-board hydrogen compression system on the engine crankshaft in the Class 8 conventional and hybridized regional-haul truck models equipped with the HPDI H₂ engine. This analysis further quantifies the associated fuel consumption penalty required to maintain a 300-bar hydrogen injection pressure across both highway and urban driving conditions under different cargo loads.

The rate of hydrogen pressure depletion in the storage tanks and the timing of compression system engagement, when the tank pressure drops below 300 bar, are influenced by the engine hydrogen consumption across the drive cycle. Figure 7.8 shows that the rate of tank pressure decrease is slower and the engagement of the compression system occurs later in the parallel HPDI H₂ configuration than in the conventional counterpart over the Highway-2 cycle under identical cargo load conditions. This difference results from the lower hydrogen consumption of the hybridized configuration, which leads to reduced cumulative compressor energy compared to the conventional model. As summarized in Table 7.10, the cumulative compressor energy per 530 km driving distance over the Highway-2 cycle at 100% cargo load is 33.2 Wh/km for the hybridized configuration, compared to 41.5 Wh/km for the conventional counterpart. When the cargo load is reduced to 35%, the corresponding

values decrease to 12.1 Wh/km and 14.0 Wh/km for the hybridized and conventional configurations, respectively. Reducing the cargo load results in lower power-demand driving conditions, requiring a smaller fuel mass per injection event. Consequently, the compression system consumes less energy to compress the reduced amount of hydrogen to the required injection pressure of 300 bar for the HPDI H₂ engine.

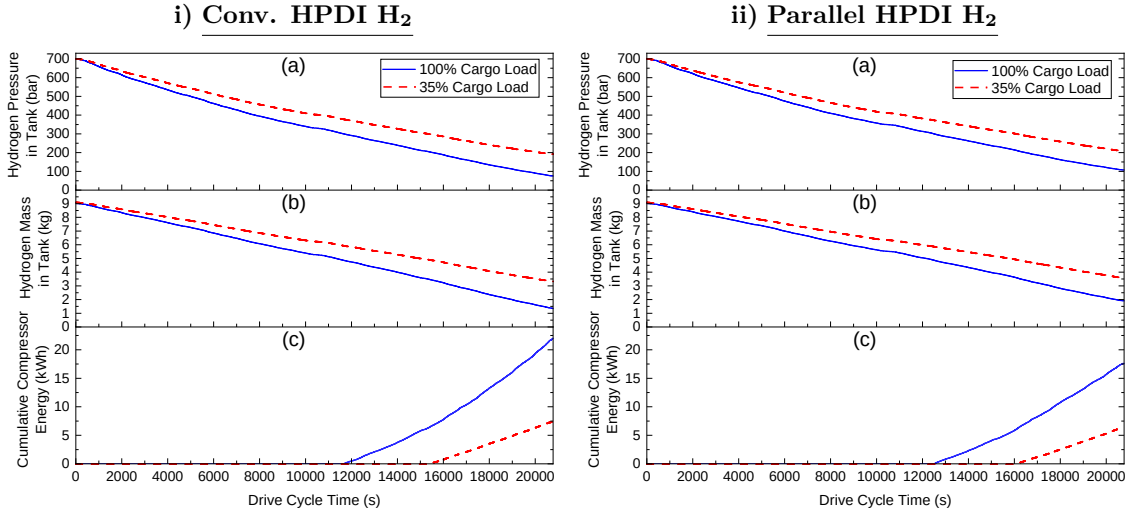


Figure 7.8: On-board hydrogen compression system performance for the Class 8 (i) conventional and (ii) pre-transmission parallel (P₂) hybrid electric regional-haul truck models equipped with the 13 liter HPDI hydrogen engine, evaluated over two consecutive repetitions (total distance: 530 km) of the Highway-2 drive cycle under 100% and 35% cargo load conditions. (a) Hydrogen pressure vs. time in each on-board storage tank, (b) hydrogen mass vs. time in each on-board storage tank, (c) cumulative compressor energy consumption vs. time.

For the conventional HPDI H₂ configuration at 100% cargo load, the cumulative compressor energy per total driving distance is significantly higher over the CSHVR drive cycle than over the other cycles (Table 7.10). However, when the cargo load is reduced to 35%, this trend is reversed, and the corresponding value for the CSHVR cycle becomes lower than for the other cycles. At higher vehicle mass, the traction energy per unit distance over the CSHVR cycle significantly exceeds that of the other drive cycles due to frequent stop-and-go operation, amplifying the mass-dependent acceleration and rolling resistance effects. This results in higher hydrogen consumption over the CSHVR cycle at 100% cargo load, as previously reported in Table 7.8, and consequently a faster drop in tank pressure below 300 bar. Notably, the fuel energy required for the conventional HPDI H₂ configuration to achieve the target driving range of 530 km over the Highway-2 cycle at 100% cargo load is equivalent to driving ranges of 569.2 km and 366.9 km over the HHDDT-65 and CSHVR cycles,

Table 7.10: Cumulative compressor energy per driving distance, driving range until storage tanks hydrogen pressure drops below 300 bar, and compressor energy as a percentage of traction energy at wheels for the Class 8 conventional and pre-transmission parallel (P₂) hybrid electric regional-haul truck configurations equipped with the 13 liter HPDI hydrogen engine across the selected drive cycles under 35% and 100% cargo load conditions.

Class 8 Truck Configuration	Highway-2		HHDDT-65		CSHVR	
	35% Cargo	100% Cargo	35% Cargo	100% Cargo	35% Cargo	100% Cargo
Cumulative Compressor Energy per Driving Distance* (Wh/km)						
Conv. HPDI H ₂	14.0	41.5	16.3	38.7	8.2	60.0
Parallel HPDI H ₂	12.1	33.2	15.1	31.8	0	15.0
Percentage of Driving Range Until Hydrogen Tank Pressure Drops Below 300 bar (% of Total Distance)						
Conv. HPDI H ₂	73.8	56.1	71.2	55.3	84.6	55.0
Parallel HPDI H ₂	77.4	60.2	75.7	58.5	100	79.0
Cumulative Compressor Energy as Percentage of Traction Energy at Wheels (%)						
Conv. HPDI H ₂	1.0	2.3	1.2	2.3	0.6	2.5
Parallel HPDI H ₂	0.9	1.8	1.1	1.9	0	0.6

* The total driving distance equivalent to 530 km over the Highway-2 cycle at 100% cargo load corresponds to 569.2 km over HHDDT-65 and 366.9 km over CSHVR.

respectively. The percentage of driving range until the hydrogen tank pressure drops below 300 bar for the conventional configuration, as detailed in Table 7.10, ranges from 55.0% to 56.1% at 100% cargo load and increases to 71.2% to 84.6% when the cargo load is reduced to 35%.

The reduction in cumulative compressor energy per driving distance for the parallel HPDI H₂ model, relative to the conventional counterpart, is more pronounced over urban cycles such as CSHVR than over the highway cycles (Table 7.10). Consequently, the hybridized configuration can sustain a longer driving range over the CSHVR cycle before the hydrogen compression system is activated when the tank pressure drops below 300 bar. More frequent operation in electric-only mode over the CSHVR cycle reduces engine-on duration and minimizes the power demand of the hydrogen compression system in the hybridized configuration.

Hybridization reduces the auxiliary power demand of the on-board hydrogen compression system applied to the crankshaft of HPDI H₂ engines, thereby lowering the associated fuel consumption penalty over the drive cycle. To illustrate this effect across the considered drive cycles and cargo loads, the cumulative compressor energy as a percentage of traction energy at the wheels is summarized in Table 7.10. A higher value of this percentage indicates a greater fuel consumption penalty associated with hydrogen compression during the trip. The traction energy at the wheels reflects the energy required to overcome aerodynamics, rolling, braking, acceleration,

and grade resistive losses, along with losses from the final drives. The associated fuel consumption penalty of the hydrogen compression system over the highway cycles (Highway-2 and HHDDT-65) ranges from 1.0% to 2.3% for the conventional configuration, while it is marginally lower, at 0.9% to 1.9%, for the hybridized configuration. Over the urban cycle (CSHVR), the corresponding ranges are 0.6% to 2.5% for the conventional and zero to 0.6% for the hybridized configuration, indicating a more pronounced reduction due to hybridization under urban driving conditions.

7.4.4 Powertrain Energy Balance and Component Efficiencies

The variations in component-level energy efficiency and the energy flows between the powertrain components for the designed conventional and hybridized regional-haul truck configurations are analyzed in this section.

Hybridization enables shifting engine operation toward higher efficiency regions (discussed in Section 7.4.2), while decreasing the average engine-on duration and enabling energy recuperation during deceleration. However, each energy conversion introduces efficiency penalties (energy losses), which reduce the net fuel consumption and CO₂ emissions benefits of the hybridized powertrain. The relative importance of these factors can be assessed using powertrain energy-flow diagrams, as illustrated over the Highway-2 drive cycle under 35% and 100% cargo load conditions in Figure 7.9. In these diagrams, the ‘Fuel’ block denotes the total chemical energy input from the fuel used to propel the vehicle. The ‘Traction Energy at Wheels’ block represents the energy required at the driving wheels downstream of the gearbox, including losses associated with aerodynamic drag, rolling resistance, braking, acceleration, grade resistance, and the final drives. The conventional and parallel hybrid configurations equipped with the HPDI H₂ engine exhibit relatively higher energy to meet the required traction energy at wheels compared to the conventional diesel counterpart over 530 km of driving on the Highway-2 cycle. This difference is attributed to the higher powertrain weight of the configurations with the HPDI H₂ engine, as previously detailed in Table 7.7. Notably, part of the additional energy required to propel the heavier hybridized vehicle is recovered during deceleration through regenerative braking.

The diesel-equivalent fuel consumption for each powertrain design is proportional to the energy supplied by the ‘Fuel’ block shown in Figure 7.9. Despite the heavier powertrain weight and the additional mechanical auxiliary load associated with the hydrogen compression system in the conventional HPDI H₂ design, the total fuel

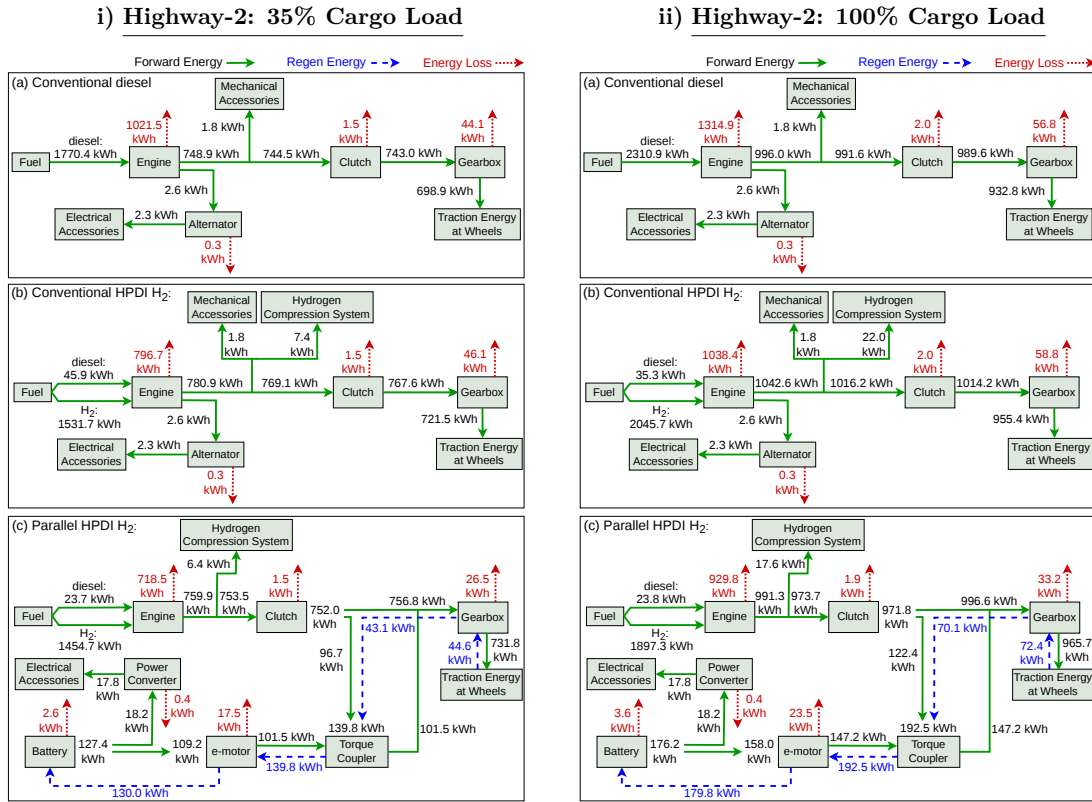


Figure 7.9: Energy flow diagrams for the designed Class 8 regional-haul truck configurations across the Highway-2 drive cycle at: (i) 35% and (ii) 100% cargo load conditions. Sub-figures correspond to: (a) conventional diesel, (b) conventional hydrogen, and (c) pre-transmission parallel (P₂) hybrid powertrains. The term ‘Traction Energy at Wheels’ reflects the energy required to overcome aerodynamics, rolling, and grade resistive forces, as well as the required energy for acceleration and losses from the final drives. Note that the conventional diesel model uses Cummins X15 engine, while the other models use 13 liter HPDI hydrogen engine.

energy consumed is lower than that of the conventional diesel configuration under the same cargo load. This is due to the more than 7% higher average engine BTE of the HPDI H₂ engine compared to the Cummins diesel engine over the Highway-2 cycle under different cargo loads, as shown earlier in Figure 7.6.

The presence of the electrical driveline in the parallel hybrid configuration, including the high-voltage battery, e-motor, and power converter, introduces additional powertrain losses compared to the conventional configurations. However, the capability to recover regenerative braking energy and operate the engine at a higher average BTE contributes to lower overall powertrain losses in the hybridized configuration. Moreover, downsizing of the gearbox in the parallel hybrid configuration leads to lower transmission losses compared to the conventional counterparts. As a result, the parallel HPDI H₂ design requires 6.3% and 7.7% less fuel energy, respectively, than the conventional HPDI H₂ design to propel the vehicle over the Highway-2 cycle at 35% and 100% cargo load conditions (Figure 7.9). The lower hydrogen fuel consumption in the parallel hybrid design also results in reduced auxiliary losses associated with the hydrogen compression system compared to the conventional configuration with the same HPDI H₂ engine under the same drive cycle and the cargo load condition.

The sizes of the primary powertrain components for the parallel hybrid configurations were optimized in Section 7.3 to minimize the diesel-equivalent fuel consumption over 530 km of driving over the Highway-2 drive cycle at full cargo load. However, variations in cargo load and drive cycles can affect the fuel consumption reduction achieved by the hybridized model relative to the conventional configurations. To clarify this effect, the cycle-averaged efficiencies of the primary powertrain components in both the parallel hybrid and the conventional design with the HPDI H₂ engine are compared in Table 7.11 across the considered drive cycles and cargo loads.

The propelling efficiency of the 10-speed manual gearbox in the conventional designs varies within a range of 93.7% to 94.2% over the considered drive cycles and cargo loads, as shown in Table 7.11. A higher range of 95.2% to 96.9% is observed for the parallel hybrid design equipped with a more efficient 6-speed automatic gearbox. The percentage of regenerative braking energy per total traction energy at the wheels for the hybridized configuration ranges from 4.8% to 7.5% over the highway cycles (Highway-2 and HHDDT-65), whereas this percentage exceeds 40% for the urban cycle (CSHVR). Although the highway cycles provide less than 10% tractive energy recovery potential through regenerative braking, this value is on the order of 50% for the urban cycle. This is one of the key reasons that the hybridized configuration achieves larger fuel consumption savings over the CSHVR drive cycle. The

Table 7.11: Cycle-averaged powertrain performance metrics for the designed conventional and pre-transmission parallel (P₂) hybrid Class 8 regional-haul truck configurations equipped with the HPDI H₂ engine across the considered drive cycles under 35% and 100% cargo load conditions.

Class 8 Truck Configuration	Highway-2		HHDDT-65		CSHVR	
	35% Cargo	100% Cargo	35% Cargo	100% Cargo	35% Cargo	100% Cargo
Total Traction Energy at Wheels over the Drive Cycle* (kWh)						
Conv. HPDI H ₂	721.5	955.4	752.1	944.8	544.4	887.7
Parallel HPDI H ₂	731.8	965.7	761.0	953.8	559.4	902.6
Cycle-Averaged Gearbox Efficiency during Propulsion (%)						
Conv. HPDI H ₂	94.0	94.2	94.0	94.2	93.7	93.9
Parallel HPDI H ₂	96.7	96.9	96.5	96.8	95.0	95.2
Cycle-Averaged Electric Motor Efficiency during Propulsion and Regeneration (%)						
Parallel HPDI H ₂	Prop: 92.9 Regen: 93.0	Prop: 93.2 Regen: 93.4	Prop: 92.8 Regen: 92.9	Prop: 93.0 Regen: 93.3	Prop: 92.3 Regen: 91.9	Prop: 92.5 Regen: 91.5
Total Regenerative Braking Energy as a Percentage of Traction Energy at Wheels (%)						
Parallel HPDI H ₂	6.1	7.5	4.8	5.7	45.4	40.9

* The total driving distance equivalent to 530 km over the Highway-2 cycle at 100% cargo load corresponds to 569.2 km over HHDDT-65 and 366.9 km over CSHVR.

small variation in cycle-averaged e-motor efficiency ($< 2\%$) across the considered drive cycles and cargo loads is attributed to the wide high-efficiency operating region of e-motors, which maintain near-constant efficiency over a broad range of speed and torque conditions.

7.5 Summary of Chapter

This chapter presents a near-zero CO₂ emission electrified truck solution using hydrogen as the primary fuel. A system-level design framework is employed for modeling and optimal component sizing of conventional and hybridized Class 8 regional-haul truck powertrains equipped with a high-efficiency diesel-pilot high-pressure direct injection (HPDI) hydrogen (H₂) engine. The HPDI H₂ engine requires a supply of H₂ at up to 300 bar. The model includes an on-vehicle hydraulically-driven booster compressor to maintain this pressure while drawing the vehicle tanks down to 75 bar. The focus is to reduce fuel consumption while maintaining tank-to-wheel (tailpipe) CO₂ emissions below the European Union (EU) zero-emission vehicle threshold of 3 g CO₂/tonne_{cargo}.km across various driving conditions and cargo load levels. The hybridized powertrain is implemented as a pre-transmission parallel (P₂) hybrid electric configuration specifically designed for heavy-duty (HD) truck applications. The key findings of this study are:

- **Conventional hydrogen vs. Conventional diesel:** The HPDI H₂ engine exhibits 7–9% higher brake thermal efficiency (BTE) than the Cummins diesel engine across the entire torque range. Consequently, the conventional H₂ model achieves lower diesel-equivalent fuel consumption than the conventional diesel counterpart. As the cargo load decreases from 100% to 35% over the highway cycles, the fuel consumption of the conventional diesel model ranges from 1.8 to 4.2 L/tonne_{cargo} · 100 km, while that of the conventional H₂ model ranges from 1.6 to 3.6 L/tonne_{cargo}, 100 km. The higher BTE of the HPDI H₂ engine is mainly attributed to the favorable thermodynamic properties of hydrogen combustion, combined with its high compression ratio of 23:1.

Due to the small amount of diesel fuel injection (< 5% of total fuel energy) used as pilot for ignition in the HPDI H₂ engine, the tailpipe CO₂ emissions of the conventional H₂ truck model are almost negligible compared to those of the conventional diesel counterpart. Over a real-world highway drive cycle under full cargo load condition, the tailpipe CO₂ emissions of the conventional diesel model are 47.9 g/tonne_{cargo}.km, whereas those of the conventional H₂ model are less than 1 g/tonne_{cargo}.km. Over urban cycles, the corresponding values are 95.9 g/tonne_{cargo}.km for the conventional diesel model and less than 2.5 g/tonne_{cargo}.km for the conventional H₂ model.

- **Parallel hybrid vs. Conventional hydrogen:** Using a highly efficient HPDI H₂ engine, the parallel hybrid electric powertrain reduces diesel-equivalent fuel consumption by up to 7.7% over a real-world highway drive cycle and up to 31.6% over urban cycles compared to a conventional truck with the same engine. The larger fuel consumption reduction under urban driving is attributed to greater regenerative braking opportunities, shorter engine-on durations, and larger improvements in average engine BTE in the hybridized model compared to the conventional one.

Lower diesel-equivalent fuel consumption of the HPDI H₂ engine in the parallel hybrid model results in reduced tailpipe CO₂ emissions, meeting the EU zero-emission threshold over both highway and urban driving across various cargo load conditions. Over a real-world highway drive cycle under full cargo load condition, the hybridized model achieves a 33.3% reduction in tailpipe CO₂ emissions relative to the conventional H₂ model, while this reduction becomes more pronounced as 56.4% over urban cycles.

- **Parasitic load of the on-board hydrogen compression system:** By

reducing the hydrogen fuel consumption of the HPDI H₂ engine, the parallel hybrid model delays the pressure drop of gaseous hydrogen in the storage tanks below 300 bar. This leads to later activation of the on-board hydrogen compression system during the trip, resulting in a lower cumulative auxiliary load on the engine. Over highway driving, when the cargo load increases from 35% to 100%, the energy consumption of the hydrogen compression system in the conventional H₂ truck model ranges from 14.0 to 41.5 Wh/km. Whereas, this range decreases to 12.1 to 33.2 Wh/km for the hybrid electric truck model. Here, Wh/km represents the compressor energy required per unit distance traveled.

Owing to its higher net powertrain energy efficiency relative to the conventional H₂ model, the hybridized configuration achieves a 5.4% to 7.3% increase in driving range over real-world highway drive cycles per refueling event, for the same on-board quantities of diesel and hydrogen fuels.

This chapter did not conduct well-to-wheel (WTW) assessment of CO₂-equivalent GHG emissions for conventional and hybridized Class 8 regional-haul trucks equipped with a diesel-pilot HPDI hydrogen engine.

Chapter 8

Conclusions, Limitations, and Future Work

8.1 Conclusions

This thesis created a new vehicle-level design framework for modeling and optimal component sizing of conventional and hybridized Class 8 heavy-duty (HD) truck configurations equipped with natural gas (NG) or hydrogen (H_2)-fueled compression ignition (CI) engines. The goal was to reduce fuel consumption and engine-out emissions (CO_2 , CH_4 , and NO_x) while maintaining propulsion system performance comparable to that of conventional diesel trucks.

A high-fidelity simulation model of a conventional baseline truck configuration was developed and validated against collected on-road test data from a Class 8 Peterbilt diesel truck operated over a real-world regional-haul drive cycle (Highway-2). Conventional truck models powered by low-carbon NG-CI and near-zero-carbon H_2 -CI engines were subsequently created by replacing the baseline diesel engine with high-pressure direct injection (HPDI) NG or H_2 engines, along with the corresponding modifications to the on-board fuel systems and storage tanks. These commercially available HPDI engines use a small amount of diesel ($< 5\%$ of total fuel energy) as a pilot for ignition while maintaining overall brake thermal efficiencies (BTEs) comparable to, or even higher than, that of the baseline diesel engine. The results demonstrated up to 21.1% reduction in tailpipe CO_2 emissions for the conventional NG and up to 98.5% reduction for the conventional H_2 model relative to the baseline diesel counterpart over the Highway-2 drive cycle.

Next, three full hybrid electric architectures with high potential for reducing fuel consumption and GHG emissions in Class 8 HD trucks were selected, and their powertrain configurations and energy management systems (EMSs) were designed. These

architectures included a pre-transmission parallel (P_2) hybrid, a complex power-split hybrid, and a series plug-in hybrid powertrain, all specifically tailored to accommodate the high torque demands associated with varying cargo loads and the wide operating range of Class 8 HD truck applications. A multi-stage component sizing framework was then created to optimize the sizes of the primary hybrid powertrain components and to tune the key EMS-related variables. This framework aimed to minimize net powertrain energy consumption over the Highway-2 drive cycle while maintaining propulsion system performance comparable to that of the conventional baseline model. Propulsion system performance was assessed based on the criteria specified by the SAE J2807 standard, including level-road acceleration time, flat-road top speed performance, and gradeability over steep road conditions.

A summary of the fuel consumption and tailpipe CO_2 emissions of the hybrid electric Class 8 truck configurations developed in this thesis, incorporating different variants of NG-CI and HPDI H_2 engines, over both highway and urban drive cycles is presented in Figure 8.1. The main conclusions of this thesis are summarized as follows:

8.1.1 Hybridization of Low-Carbon NG-CI Engines

The hybrid electric powertrains were optimized for integration with different variants of NG-CI engines to further reduce GHG emissions in Class 8 HD truck applications. These engines included a downsized diesel-pilot premixed NG (micro-pilot) engine, as well as downsized and full-size technology-enhanced variants of the diesel-pilot HPDI NG engine. The technologies added to the HPDI NG engine include: increasing compression ratio (CR), advancing start of injection (SOI), and integrating a series turbo-compound (TC) system. These technologies were implemented to improve engine BTE, particularly at mid-to-high torque levels, where the engine in hybrid configurations operates predominantly.

- **Hybridization effects on improving net powertrain energy conversion efficiency:** First, hybridization enabled the recovery of regenerative braking energy, thus reducing total powertrain energy losses; however, the associated braking potential was limited over highway-cruising drive cycles. Second, the results showed that hybridization allowed the engine to either remain off or operate for more than 85% of the engine-on time within high-torque regions ($T > 0.5 T_{\max}$) associated with higher BTE, regardless of drive cycle (highway or urban) or cargo load condition. In contrast, the corresponding percentage

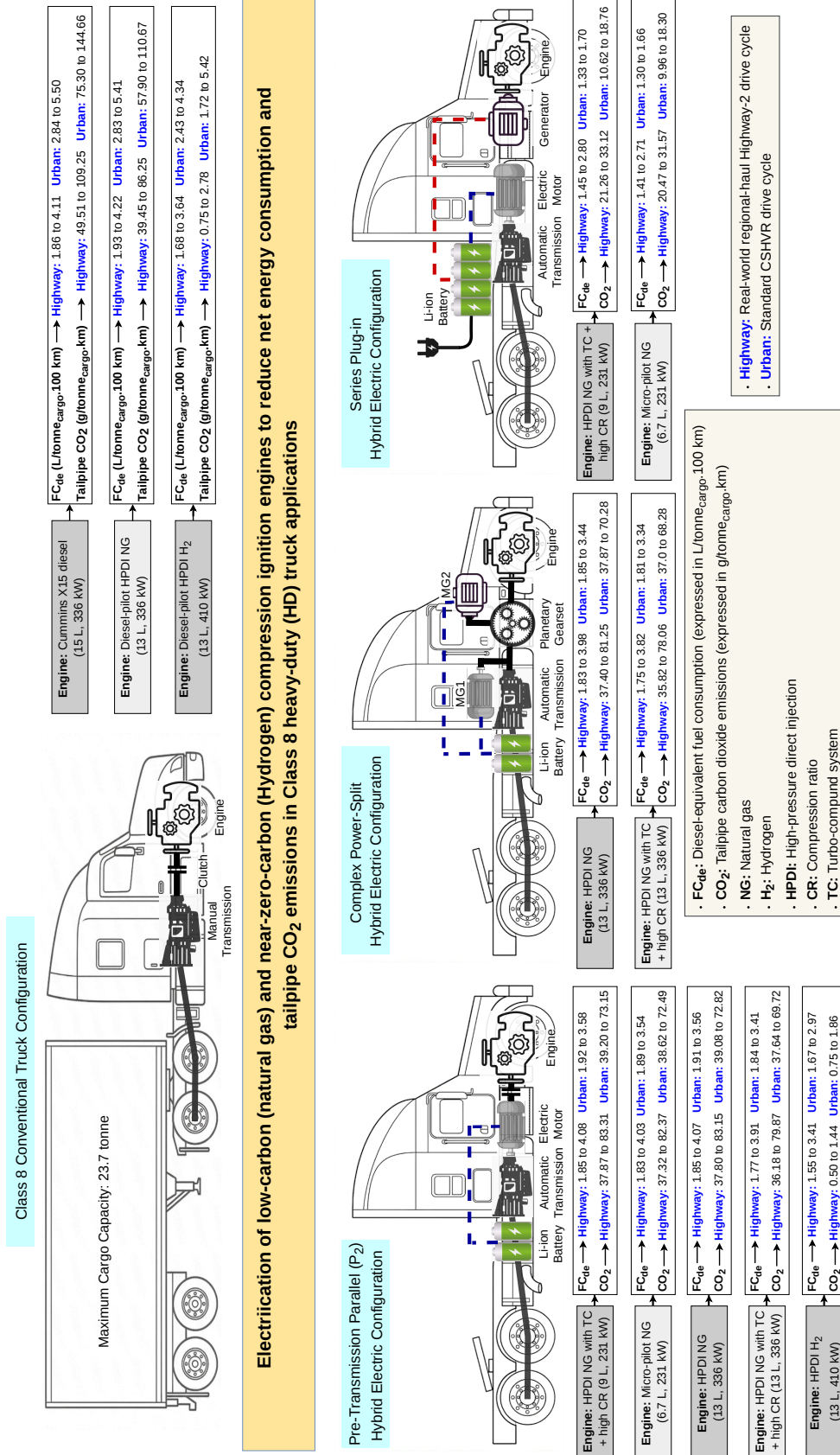


Figure 8.1: Comparison of diesel-equivalent fuel consumption and tailpipe CO₂ emissions for the designed conventional and hybrid electric Class 8 truck configurations in this thesis, including diesel-, natural gas (NG)-, and hydrogen (H₂)-fueled compression ignition (CI) engines. Results are presented over representative highway and urban drive cycles across different cargo load conditions.

for conventional configurations ranged from 13% to 75%, with lower values observed under less power-demanding conditions such as low cargo loads and urban driving. As a result, the hybridized configurations achieved a higher cycle-average BTE and improved net powertrain energy efficiency, with the greatest benefits observed when technology-enhanced engines with higher BTE at high-torque operating conditions were incorporated.

- **Fuel consumption and tailpipe CO₂ emissions:** Among the designed hybridized configurations incorporating different NG-CI engine variants (Figure 8.1), the series plug-in hybrid powertrain equipped with the downsized (6.7 L, 231 kW) micro-pilot NG engine achieved the lowest diesel-equivalent fuel consumption and tailpipe CO₂ emissions. Relative to the conventional diesel model, it achieved fuel consumption and tailpipe CO₂ emission reductions of 24.2% and 58.7%, respectively, over the Highway-2 drive cycle under full cargo load. The corresponding reductions for the plug-in hybrid powertrain equipped with the downsized (9 L, 231 kW) HPDI NG engine incorporating both TC and high CR were slightly lower at 22.1% and 57.1%, respectively. This difference was primarily attributed to the 2-2.5% higher BTE of the micro-pilot NG engine at high-torque regions ($T > 0.6 T_{\max}$), resulting in lower powertrain energy losses over the drive cycle.

Among the designed non-plug-in hybrid configurations, the greatest reductions were achieved by the power-split hybrid powertrain equipped with the full-size (13 L, 336 kW) HPDI NG engine with both TC and high CR. The corresponding reductions were 5.9% and 27.7%, respectively. The reductions achieved by the P₂ hybrid powertrain equipped with the same engine were slightly lower at 4.8% and 26.9%, respectively. These results highlighted the importance of incorporating technology-enhanced full-size HPDI NG engines in non-plug-in hybrid applications. Since these configurations relied primarily on the engine as the main energy source, improvements in engine BTE translated directly into lower fuel consumption and greenhouse gas (GHG) emissions.

- **Engine-out NO_x and CH₄ emissions:** Co-optimization of the engine and hybrid powertrain significantly reduced engine-out NO_x and CH₄ emissions primarily by concentrating engine operation in high-efficiency regions with minimum emissions, and avoiding low-load, inefficient regions. The greatest reductions were achieved by the plug-in hybrid configurations equipped with downsized NG-CI engines, as these architectures relied most heavily on the electri-

cal power source and exhibited the shortest engine-on duration over the drive cycle. Relative to the conventional HPDI NG configuration, engine-out NO_x and CH_4 emissions were reduced by up to 70.8% and 73.4%, respectively, over the Highway-2 drive cycle under full cargo load. Under less power-demanding operating conditions, such as urban driving or reduced cargo loads, the corresponding reductions became more pronounced, reaching up to 87.7% and 98.8%, respectively.

Among the designed non-plug-in hybrid configurations, the greatest reductions were achieved by the power-split hybrid powertrain equipped with the full-size HPDI NG engine incorporating both TC and high CR. The corresponding engine-out NO_x and CH_4 emission reductions were 12.6% and 39.1%, respectively, over the Highway-2 drive cycle under full cargo load. For the P_2 hybrid powertrain equipped with the same engine, the corresponding values were slightly lower at 10.4% and 36.9%, respectively. This advantage was primarily attributed to the greater flexibility of the power-split architecture in maintaining engine operation within high-efficiency, low-emission regions, together with its slightly shorter engine-on duration over the drive cycle.

Overall, the designed plug-in hybrid Class 8 truck configurations equipped with high-efficiency NG-CI engines achieved up to a 56.9% reduction in tailpipe net GHG emissions (CO_2 -equivalent), including CH_4 , relative to the conventional diesel baseline model over the Highway-2 drive cycle under full cargo load. This result indicated that such configurations could satisfy both the U.S. EPA Phase-3 interim GHG reduction target for 2032 and the European Union (EU) GHG reduction target of 45% by 2030. The designed power-split and P_2 hybrid configurations equipped with high-efficiency advanced NG-CI engines achieved up to a 27.3% reduction in tailpipe net GHG emissions under the same operating conditions, which was sufficient to satisfy only the U.S. EPA Phase-3 target.

8.1.2 Hybridization of Near-Zero-Carbon H_2 -CI Engines

The P_2 hybrid powertrain were designed and optimized for integration with a commercially available diesel-pilot HPDI H_2 engine with a maximum BTE of $\sim 52\%$. This engine uses a small amount of diesel fuel (1–5% of total fuel energy) as a pilot for ignition and offer 2–8% higher overall BTE than the baseline diesel engine. New models were created to simulate the H_2 -CI engine and the auxiliary load imposed on

the engine by the on-board H₂ compression system to maintain a 300 bar hydrogen injection pressure as on-board storage tanks are depleted.

- **Fuel consumption and tailpipe GHG emissions:** Results demonstrated a 17.1% reduction in diesel-equivalent fuel consumption for the P₂ hybrid configuration equipped with the HPDI H₂ engine relative to the conventional diesel model over the Highway-2 drive cycle under full cargo load. The corresponding reduction became more pronounced, reaching up to 45.8% over the urban drive cycle, primarily due to the greater availability of regenerative braking and the larger improvement in cycle-average engine BTE relative to the conventional diesel model.

Tailpipe GHG emissions were reduced by 98.9% relative to the conventional diesel model over the Highway-2 drive cycle under full cargo load, demonstrating that the hybridized model can meet the EU zero-emission vehicle threshold of less than 3 g CO₂/tonne_{cargo}·km (tank-to-wheel). The corresponding reduction became more pronounced, reaching up to 99.1% over the urban drive cycle.

- **Parasitic load of the on-board hydrogen compression system:** By reducing the hydrogen fuel consumption of the HPDI H₂ engine, the parallel hybrid model delayed the pressure drop of gaseous hydrogen in the storage tanks below 300 bar. This led to later activation of the onboard hydrogen compression system during the trip, resulting in a lower cumulative auxiliary load on the engine. Over highway driving, as the cargo load increased from 35% to 100%, the energy consumption of the hydrogen compression system in the conventional H₂ truck model ranged from 14.0 to 41.5 Wh/km, whereas the corresponding range decreased to 12.1 to 33.2 Wh/km for the hybrid electric truck model.

Owing to its higher net powertrain energy efficiency relative to the conventional diesel baseline model, the hybridized configuration achieved a 5.4% to 7.3% increase in driving range over real-world highway drive cycles per refueling event for the same on-board quantities of diesel and hydrogen fuel.

8.1.3 Development of a Real-World Gradeability Performance Criterion for Hybridized HD Trucks

The existing gradeability criterion established by the Society of Automotive Engineers (SAE) for conventional HD trucks was revised to address the testing requirements of

hybridized Class 8 HD trucks with downsized engines. Unlike conventional SAE gradeability assessments that rely on constant road grades, hybridized truck performance is strongly influenced by the continuously varying nature of real-world road profiles, which affect battery energy utilization and regenerative braking opportunities. Therefore, real-world road grade profiles for 15 identified steep highways used by HD trucks across different regions (U.S., Canada, Europe, and China) were derived. The optimal component sizing of the conventional and P₂ hybrid models was then refined using each of these identified steep highways as the gradeability criterion.

The results demonstrated that the conventional diesel model with a 336 kW engine could be converted to a P₂ hybrid electric configuration equipped with a 231 kW engine, a 200 kW e-motor, and a 60.8 kWh battery without sacrificing propulsion system performance or maximum cargo load capacity. Under these conditions, the equivalent fuel consumption of the hybridized model with the downsized diesel engine over the Highway-2 drive cycle was 3.4% to 8.9% lower than that of the conventional counterpart under identical cargo load conditions. This underscores the feasibility of engine downsizing for hybridized Class 8 HD trucks while maintaining propulsion system performance comparable to that of conventional counterparts.

8.2 Modeling Assumptions, Uncertainties, and Limitations

Although this thesis developed and validated a vehicle-level modeling framework for evaluating conventional and hybridized Class 8 HD truck configurations, several modeling assumptions and uncertainties should be considered when interpreting the results. The main limitations are summarized below:

8.2.1 Powertrain Modeling

- The vehicle models are developed as vehicle-level simulation models using map-based component representations. Therefore, high-frequency dynamics, detailed thermal transients, component aging, and actuator-level control dynamics are not fully captured.
- The aerodynamic drag coefficient and rolling resistance coefficients are assumed to be constant for each vehicle configuration, while the frontal area is fixed based on the nominal tractor-trailer geometry. In real-world operation, aerodynamic drag can vary with crosswind, yaw angle, tractor-trailer gap, trailer height,

aerodynamic fairings, and vehicle speed, while rolling resistance can vary with tire pressure, tire temperature, tire wear, pavement condition, road roughness, and vehicle loading distribution.

- Road grade is considered as the primary external disturbance in the vehicle longitudinal dynamics. Other real-world disturbances, such as wind, traffic interactions, driver behavior, road surface variability, and ambient temperature effects, are not explicitly modeled.
- Auxiliary loads are represented using simplified or fixed power-demand assumptions, except for the on-board hydrogen compression system, which is modeled using a map-based approach. Time-varying HVAC loads, cooling-system demands, and thermal-management loads under extreme ambient conditions are not fully represented.
- The battery models are based on equivalent-circuit/map-based formulations and do not include long-term battery degradation, capacity fade, internal-resistance growth, thermal aging, or the effect of battery temperature on performance and usable energy.
- The powertrain component mass and efficiency assumptions are based on representative values from the literature, manufacturer data, or Autonomie component models. Detailed packaging constraints, structural integration requirements, and component-level design variations are outside the scope of this thesis.
- The baseline conventional diesel truck model was validated against real-world Highway-2 data; however, the hybridized NG- and H₂-fueled configurations are evaluated through simulation. Therefore, additional experimental validation would be required before direct implementation in production vehicles.

8.2.2 Energy Management Strategies

- The rule-based EMSs developed for the parallel and power-split hybrid configurations rely on predefined thresholds for SOC, engine operating region, torque demand, and vehicle speed. These rules are effective for transparent vehicle-level comparison, but may require further calibration for real-time implementation under broader operating conditions.
- The EMS parameters are optimized mainly over selected representative drive cycles and cargo load conditions. Therefore, the resulting control parameters

may not be globally optimal for all possible routes, traffic conditions, weather conditions, or payload distributions.

- The simulations assume accurate knowledge of key states, such as vehicle speed, torque demand, battery SOC, engine speed, and component operating points. In real vehicles, sensor noise, estimation errors, communication delay, and actuator response limitations may affect EMS performance.
- The plug-in hybrid EMS is formulated using a predefined drive cycle and target blended charge-depletion trajectory. Therefore, its performance depends on prior knowledge of trip distance and expected energy demand, and may differ under unknown or highly variable routes.
- The optimization-based control results are used as offline simulation benchmarks. Real-time implementation would require additional observer-based feedback, route prediction, adaptive control, or model predictive control to account for uncertainties in road grade, vehicle mass, battery condition, and traffic disturbances.

8.2.3 Emissions Quantification

- Engine fuel consumption and emissions are quantified using steady-state engine maps as functions of engine speed and torque. Therefore, transient effects such as turbocharger lag, EGR dynamics, injection-system response, exhaust temperature variation, and rapid load changes are not fully captured.
- The NO_x and CH_4 emissions reported for the NG-CI engines are engine-out values. Detailed exhaust aftertreatment models, including DOC, DPF, SCR, TWC, catalyst light-off, urea dosing, ammonia slip, and catalyst temperature dynamics, are not included.
- The tailpipe GHG analysis focuses on tank-to-wheel emissions. Upstream emissions associated with fuel production, fuel transportation, hydrogen production pathways, methane leakage, electricity generation for plug-in hybrid operation, and vehicle/component manufacturing are not included.
- The current analysis does not include a full life-cycle assessment or techno-economic assessment. Therefore, the results should be interpreted as vehicle-level tank-to-wheel performance indicators rather than complete well-to-wheel or cradle-to-grave environmental impacts.

8.3 Future Work

Building on the findings and limitations of this thesis, several directions for future research are identified to further improve the performance, efficiency, and applicability of hybridized Class 8 HD truck powertrains. These include:

- A comprehensive well-to-wheel (WTW) assessment of CO₂-equivalent GHG emissions for conventional and hybrid electric class 8 HD trucks equipped with NG- or H₂-CI engines is recommended. This analysis would require the development of an aftertreatment system model to quantify tailpipe GHG emissions, along with the inclusion of upstream fuel production pathways to capture full life-cycle CO₂-equivalent impacts.
- A techno-economic assessment is recommended to evaluate the total cost of ownership (TCO) of the proposed hybridized Class 8 HD truck configurations. This analysis should include capital and operating costs, as well as key indicators such as payback period and net present value (NPV), to assess the economic feasibility relative to conventional diesel counterparts.
- A detailed assessment of the time-varying impact of heating, ventilation, and air conditioning (HVAC) loads on the optimal operation of electrified Class 8 sleeper trucks is recommended, particularly in cold climates. This analysis should quantify the additional energy demand and its impact on energy management strategies, battery usage, and overall efficiency.
- Development and validation of an electrically driven on-board hydrogen compression system for HPDI H₂ engine truck configurations are recommended to replace the mechanically driven compressor used in this thesis. This approach would reduce additional engine torque and speed transients and enable improved system control and efficiency.
- A systematic uncertainty and sensitivity analysis is recommended to quantify the influence of key modeling assumptions on fuel consumption, emissions, and component sizing results. These assumptions include aerodynamic drag coefficient, rolling resistance coefficients, auxiliary loads, component efficiency maps, vehicle mass, battery parameters, and road-load disturbances.
- Development of real-time adaptive energy management strategies is recommended for future implementation of the proposed hybrid powertrains. Such

strategies could incorporate feedback from measured or estimated states, including battery SOC, road grade, vehicle mass, traffic conditions, and component temperatures, to improve robustness under real-world disturbances, sensor noise, and route uncertainty.

- Investigation of the proposed hybridization strategies for other commercial vehicle applications, including transit buses, is recommended. Since transit buses predominantly operate over urban drive cycles with frequent stop-and-go conditions, they offer high potential for regenerative braking and engine load optimization, leading to greater energy-saving opportunities and reductions in GHG emissions.

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Appendix A: Ph.D. Publications

A.1 Peer Reviewed Journal Papers from Thesis

A.1.1 Published Journal Papers

1. **S. Moghadasi**, N. Balazadeh Meresht, G. McTaggart-Cowan, M. Shahbakhti, “Fuel consumption and emissions reductions of hybrid electric regional-haul trucks equipped with a high-pressure direct injection natural gas engine in real-world driving conditions”, *Energy Technology*, Impact Factor: 3.6, vol. 14, 202502541, 2026, 31 pages.
2. **S. Moghadasi**, A. Yasami, S. Munshi, G. McTaggart-Cowan, M. Shahbakhti, “Real-world steep drive cycles and gradeability performance analysis of hybrid electric and conventional class 8 regional-haul truck”, *Energy*, Impact Factor: 10.1, vol. 320, 135128, 2025, 24 pages.
3. **S. Moghadasi**, Y. Long, L. Jiang, S. Munshi, G. McTaggart-Cowan, 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*, Impact Factor: 11.8, vol. 309, 118451, 2024, 20 pages.
4. N. Balazadeh Meresht, **S. Moghadasi**, S. Munshi, M. Shahbakhti, G. McTaggart-Cowan, “Advances in vehicle and powertrain efficiency of long-haul commercial vehicles: A review”, *Energies*, Impact Factor: 3.9, vol. 16, 6809, 2023, 37 pages.

A.1.2 Submitted Journal Papers

1. N. Balazadeh Meresht, **S. Moghadasi**, M. Shahbakhti, G. McTaggart-Cowan, “Joint optimization of high-efficiency natural gas engines and hybrid-electric powertrains for heavy-duty trucks”, Submitted to *Applied Thermal Engineering*, Elsevier, Impact Factor: 7.5, Submitted on June 2, 2026, 27 pages.
2. **S. Moghadasi**, M. Nikkhah Koojehri, N. Balazadeh Meresht, G. McTaggart-Cowan, M. Shahbakhti, “Energy saving and near-zero greenhouse gas emissions in heavy-duty trucks via design of hybrid electric powertrains with high-pressure direct injection hydrogen engines”, Submitted to *Energy Institute*, Elsevier, Impact Factor: 6.5, Submitted on July 21, 2026, 38 pages.

A.2 Peer Reviewed Journal Papers from Other Collaborations

A.2.1 Published Journal Paper

1. L. Jiang, **S. Moghadasi**, M. Shahbakhti, “Platooning electrified trucks: benefits of saving energy and minimizing emissions”, *Energy*, Impact Factor: 10.1, vol. 358, 141332, 2026, 17 pages.

A.3 Peer Reviewed Conference Papers from Other Collaborations

1. N. Balazadeh Meresht, **S. Moghadasi**, M. Shahbakhti, G. McTaggart-Cowan, “Phenomenological modeling and system-level evaluation of a high-efficiency hydrogen engine for class 8 heavy-duty trucks”, Under review in the *ASME International Mechanical Engineering Congress and Exposition (IMECE 2026)*, Submitted on May 4, 2026.
2. **S. Moghadasi**, A. Salahi, H. Borhan, C. R. Koch, M. Shahbakhti, “Model

predictive control using machine learning for voltage control of a PEM fuel cell stack”, *Proceedings of the Canadian Society for Mechanical Engineering International Congress (CSME)*, May 28–31, 2023, Sherbrooke, Quebec, Canada.

3. A. Salahi, **S. Moghadasi**, H. Borhan, C. R. Koch, M. Shahbakhti, “Imitative learning control of a LSTM-NMPC controller on PEM fuel cell for computational cost reduction”, *Proceedings of the Canadian Society for Mechanical Engineering International Congress (CSME)*, May 28–31, 2023, Sherbrooke, Quebec, Canada.

A.4 Technical Presentations (Refereed Abstracts)

1. **S. Moghadasi**, Y. Long, S. Munshi, G. McTaggart-Cowan, M. Shahbakhti, “Investigation of Davis Dam drive cycle application for performance assessment of electrified and conventional long-haul trucks”, *Proceedings of the Canadian Society for Mechanical Engineering International Congress (CSME)*, June 5–8, 2022, Edmonton, Alberta, Canada.
2. Y. Long, L. Jiang, **S. Moghadasi**, S. Munshi, G. McTaggart-Cowan, M. Shahbakhti, “Performance analysis of a heavy-duty hybrid electric truck with a micro-pilot natural gas engine in real-world highway driving conditions”, *Proceedings of the Canadian Society for Mechanical Engineering International Congress (CSME)*, June 5–8, 2022, Edmonton, Alberta, Canada.

Appendix B: Road Grade Data of the Identified Real-World Steep Highways for Gradeability Testing of Heavy-Duty Trucks

This appendix provides supporting materials related to the identified real-world steep highways presented in Chapter 5 for the gradeability performance assessment of Class 8 HD trucks.

The global positioning system (GPS) data for the 15 identified steep highways, including latitude and longitude, were digitized from highway route profiles in Google Earth Pro[®], with an average horizontal distance of 73 meters between consecutive points. The GPS-based data of steep highways were then extracted to generate grade-distance profiles, using the algorithm outlined in Section 5.3.3 of Chapter 5. A repository containing detailed information on these steep highways, including latitude, longitude, altitude, distance, and road grade, is available at the following link:

Link: https://github.com/yasami-amirreza/Real-world_Steep_Drive_Cycles_GPS_and_Grade_Dataset

The accuracy of altitude data in Google Earth Pro[®] varies based on factors such as the elevation data source and the region mapped. To achieve higher altitude data accuracy, digital elevation models (DEMs) [229] were utilized. High-accuracy altitude data (approximately 30 meters) for highways in the U.S. and Canada was extracted from DEM data sources provided by the National Aeronautics and Space Admin-

istration (NASA) and the U.S. Geological Survey (USGS) [251]. For the identified highways in Europe and China, the ODP1 DEM tool managed by the USGS was used [252].

Appendix C: Supplementary Contents for Hybridization of Dual-Fuel Diesel–Natural Gas Engines

This appendix provides supporting materials for the integration of powertrain hybridization with low-carbon natural gas engine variants, as presented in detail in Chapter 6 of this thesis.

C.1 Calibrated engine torque-speed operating boundaries for the designed non-plug-in hybrid electric configurations

The calibrated engine torque–speed operating curves for the different dual-fuel diesel–natural gas engine variants used in the parallel and power-split hybrid electric truck configurations in this thesis are shown in Figure C.1. These curves include the low, optimum, and high torque-speed lines, which constrain engine operation to the high-efficiency region as part of the constrained rule-based energy management system (EMS) for the hybrid powertrains. Details of the EMS control strategies for the parallel and power-split hybrid configurations are provided in Chapter 3, Sections 3.2.2 and 3.3.2, respectively. The optimum torque line at each engine speed is determined by identifying the torque corresponding to the maximum thermal efficiency, or equivalently the minimum BSFC. The low- and high-torque lines are then obtained by optimizing the engine efficiency threshold (δ_e) as an EMS-related parameter within the optimal component sizing framework, as detailed in Chapter 4, Section 4.3.

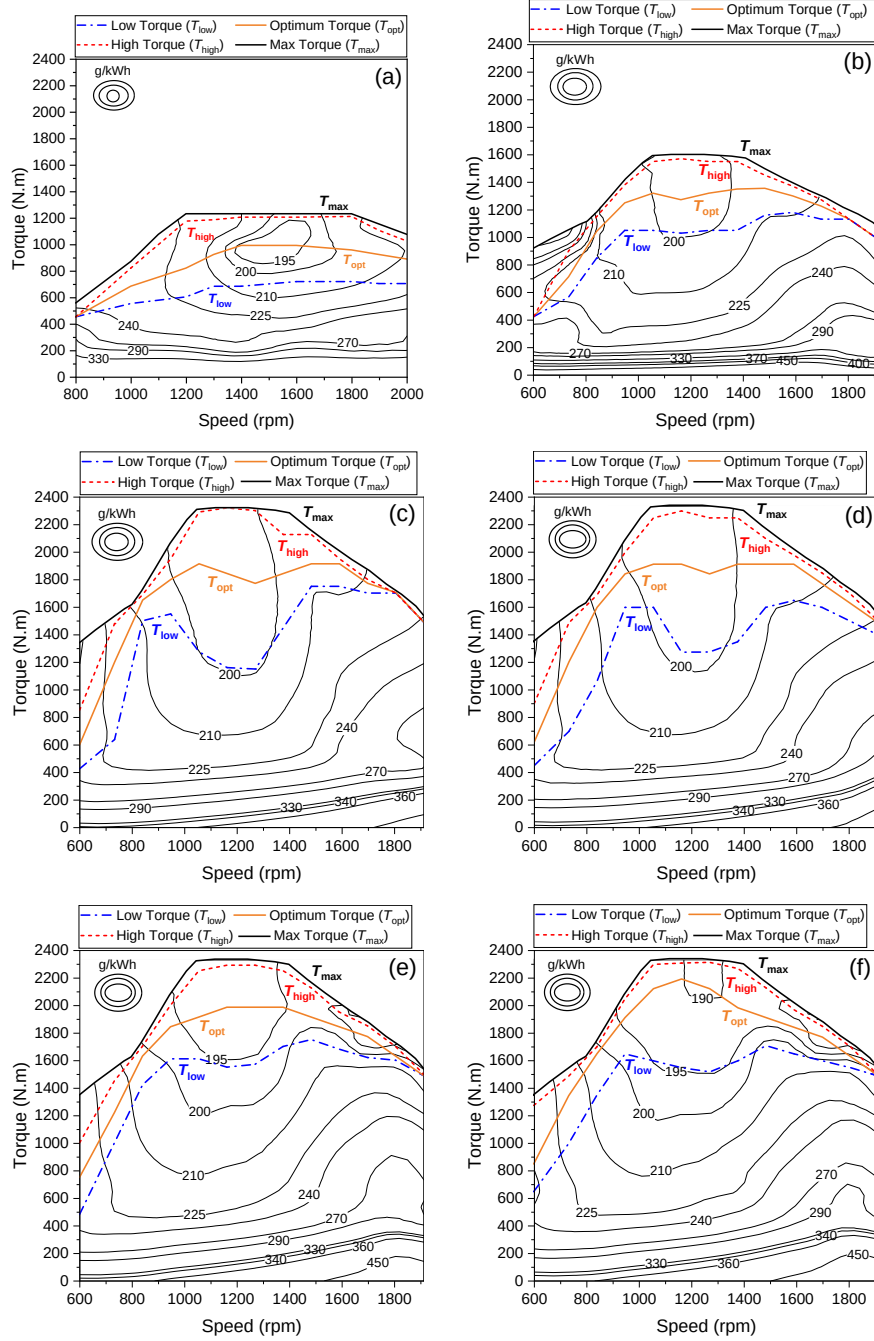


Figure C.1: Calibrated engine torque–speed operating lines as part of the constrained rule-based control strategy for the designed parallel and power-split hybrid Class 8 truck configurations, each equipped with different variants of the NG-CI engine: (a) 6.7 L micro-pilot NG engine (231 kW), (b) 9 L HPDI NG engine with turbo-compounding (TC) and high compression ratio (CR) (231 kW), (c) 13 L baseline HPDI NG engine (336 kW), (d) 13 L HPDI NG engine with advanced start of injection (SOI) (336 kW), (e) 13 L HPDI NG engine with TC (336 kW), and (f) 13 L HPDI NG engine with TC and high CR (336 kW). The optimum torque line at each engine speed corresponds to maximum BTE (minimum BSFC), while the low and high torque lines are defined using the engine efficiency threshold (δ_e) as deviations from the optimum line.

C.2 Powertrain component efficiencies of the hybridized configurations under different drive cycles and cargo loads

The cycle-average operational efficiencies of the e-motor(s) and transmission system, along with the percentage of regenerative braking energy at the wheels, for the designed hybrid electric Class 8 truck configurations equipped with different variants of the HPDI NG engine are summarized in Table C.1. These metrics are reported for each configuration across the three selected drive cycles under both 35% and 100% cargo load conditions.

Table C.1: Cycle-averaged powertrain performance metrics for the designed conventional and hybrid electric Class 8 truck configurations equipped with different variants of the HPDI NG engine, across the selected drive cycles under 35% and 100% cargo load conditions.

Designed Class 8 Truck Configurations	Highway-2		HHDDT-65		CSHVR	
	35% Cargo	100% Cargo	35% Cargo	100% Cargo	35% Cargo	100% Cargo
Cycle-Averaged Gearbox Efficiency during Propulsion (%)						
Conv. HPDING Baseline	94.0	94.3	94.0	94.3	93.6	93.9
P ₂ HPDING + TC + CR (9 L)	96.9	97.1	96.4	97.0	96.5	96.7
P ₂ HPDING (13 L)	96.9	97.1	96.4	97.0	96.5	96.7
P ₂ HPDING + TC + CR (13 L)	96.9	97.1	96.4	97.0	96.5	96.7
Power-Split HPDING (13 L)	Gb ¹ : 98.7 PG ² : 99.0	Gb: 98.9 PG: 99.2	Gb: 98.7 PG: 98.6	Gb: 98.9 PG: 98.8	Gb: 97.7 PG: 98.6	Gb: 98.2 PG: 98.8
Plug-in HPDING + TC + CR (9 L)	97.6	97.8	97.5	97.7	97.1	97.4
Cycle-Averaged e-Motor/MG1 Efficiency during Propulsion and Regeneration (%)						
P ₂ HPDING + TC + CR (9 L)	Prop: 91.5 Regen: 92.6	Prop: 91.9 Regen: 92.8	Prop: 91.2 Regen: 92.5	Prop: 91.5 Regen: 92.7	Prop: 91.1 Regen: 92.4	Prop: 91.3 Regen: 92.0
P ₂ HPDING (13 L)	Prop: 93.0 Regen: 92.2	Prop: 93.4 Regen: 92.7	Prop: 92.6 Regen: 92.1	Prop: 93.0 Regen: 92.6	Prop: 92.5 Regen: 92.0	Prop: 92.7 Regen: 91.4
P ₂ HPDING + TC + CR (13 L)	Prop: 93.3 Regen: 92.9	Prop: 93.7 Regen: 93.4	Prop: 92.9 Regen: 92.7	Prop: 93.3 Regen: 93.0	Prop: 92.7 Regen: 92.2	Prop: 92.9 Regen: 92.0
Power-Split HPDING (13 L)	Prop: 93.3 Regen: 93.5	Prop: 93.7 Regen: 92.9	Prop: 93.2 Regen: 93.1	Prop: 93.5 Regen: 92.9	Prop: 93.3 Regen: 93.3	Prop: 93.0 Regen: 92.9
Plug-in HPDING + TC + CR (9 L)	Prop: 94.0 Regen: 92.5	Prop: 93.9 Regen: 92.8	Prop: 93.8 Regen: 92.5	Prop: 93.7 Regen: 92.6	Prop: 93.0 Regen: 92.1	Prop: 92.9 Regen: 91.6
Cycle-Averaged Generator/MG2 Efficiency during Propulsion and Regeneration (%)						
Power-Split HPDING (13 L)	Prop: 92.5 Regen: 93.7	Prop: 92.4 Regen: 93.5	Prop: 92.4 Regen: 93.4	Prop: 92.3 Regen: 93.2	Prop: 90.9 Regen: 92.5	Prop: 92.8 Regen: 94.0
Plug-in HPDING + TC + CR (9 L)	Gen: 93.5	Gen: 93.7	Gen: 93.5	Gen: 93.7	Gen: 92.6	Gen: 92.8
Total Regenerative Braking Energy as a Percentage of Traction Energy at Wheels (%)						
P ₂ HPDING + TC + CR (9 L)	6.1	7.5	4.8	5.7	51.5	47.2
P ₂ HPDING (13 L)	6.1	7.5	4.8	5.7	51.5	47.2
P ₂ HPDING + TC + CR (13 L)	6.1	7.5	4.8	5.7	51.5	47.2
Power-Split HPDING (13 L)	6.3	7.7	5.0	5.9	52.9	48.5
Plug-in HPDING + TC + CR (9 L)	7.3	8.1	5.3	6.1	53.5	49.2

¹ ‘Gb’ refers to the multi-speed automatic gearbox in the hybridized configurations.

² ‘PG’ refers to the planetary gearset in the power-split hybrid electric configuration.

Appendix D: Supplementary Contents for Hybridization of the diesel-pilot HPDI Hydrogen Engine

This appendix provides supporting materials for the integration of powertrain hybridization with the near-zero-carbon diesel-pilot HPDI H₂ engine, as presented in detail in Chapter 7 of this thesis.

The engine operating points on the BSFC contour maps for the designed Class 8 truck configurations across the three selected drive cycles at 35% cargo load, along with the corresponding percentages of operation within different BSFC zones are shown in Figure D.1. The results indicate that the parallel hybrid model is capable of maintaining engine operation predominantly within the high-efficiency region (i.e., $\text{BSFC} < 165 \text{ g/kWh}$), irrespective of variations in drive cycle and cargo load. In contrast, for the conventional powertrains, a reduction in cargo load or transition to a lower power-demanding drive cycle causes a significant portion of engine operation to shift toward low-torque regions ($T < 0.4 T_{\text{max}}$), associated with lower BTEs.

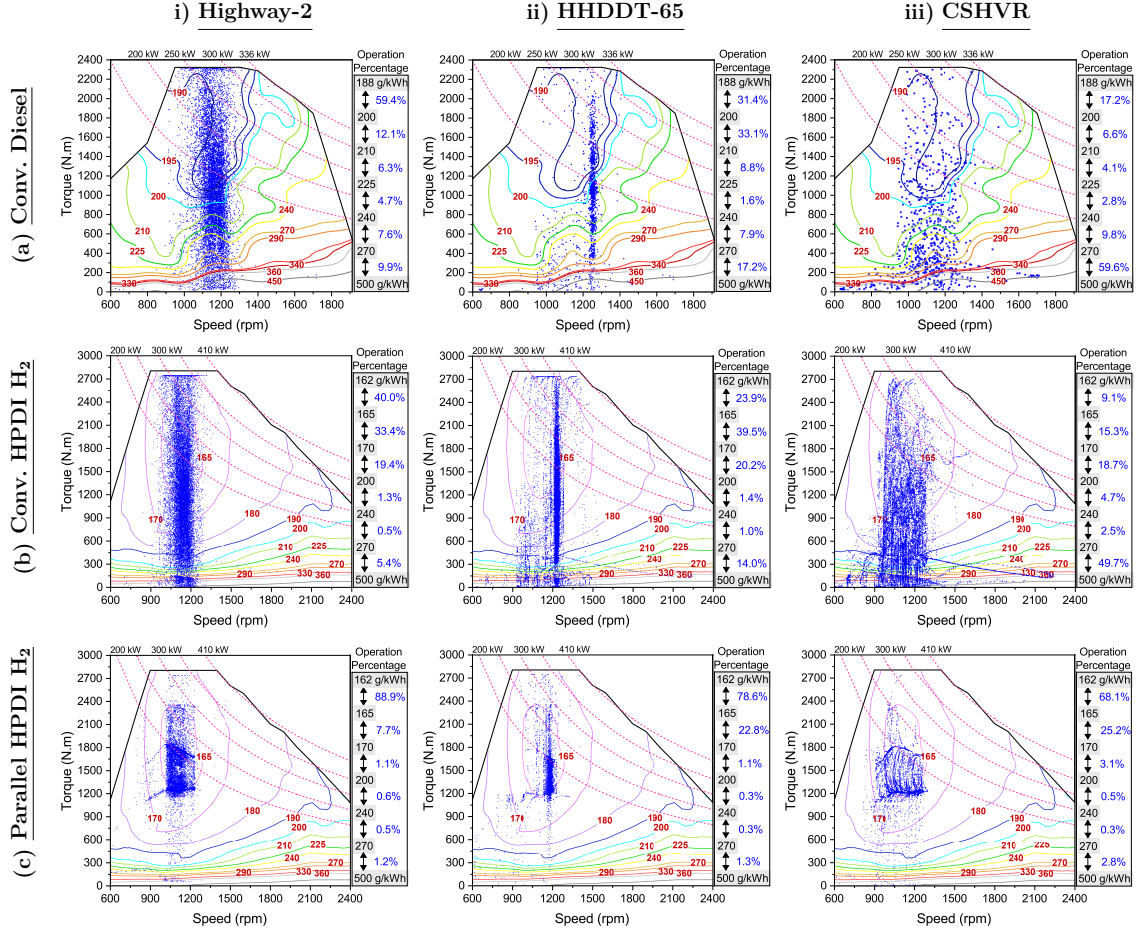


Figure D.1: Weighted engine operations on BSFC (diesel equivalent) contour plots over the three drive cycles: (i) Highway-2, (ii) HHDDT-65, and (iii) CSHVR, evaluated for the designed Class 8 regional-haul truck configurations at 35% cargo load. Sub-figures correspond to: (a) conventional diesel, (b) conventional H₂, and (c) pre-transmission parallel (P₂) hybrid powertrains. Each point represents the ICE operating state during engine-on periods over a one-second interval. The percentage tables shown beside each sub-figure indicate the fraction of engine operating time spent within the corresponding BSFC ranges. The conventional diesel model uses the Cummins X15 engine, while the other models use the 13 liter HPDI hydrogen engine.

Appendix E: Research Source Files

E.1 Chapter 1 Source Files

Table E.1: Source files of Chapter 1.

File name	File description
GHG reduction technology pathway in HD trucks.opju	Figure 1.1
NG-ICEs to meet EU GHG emission targets.opju	Figure 1.2
Aerodynamic and rolling resistance enhancements in HD trucks.pptx	Figure 1.3
Types of NG- and H2-ICEs in HD trucks.drawio	Figure 1.4
Full hybrid powertrain configurations for HD trucks.drawio	Figure 1.5
Various parallel hybrid configurations for HD trucks.pptx	Figure 1.6
Engine power range of conventional Class 8 trucks from OEMs.opju	Figure 1.7
Thesis structure and chapters contributions.drawio	Figure 1.8

E.2 Chapter 2 Source Files

Table E.2: Source files of Chapter 2.

File name	File description
Conventional diesel truck.slx	System model build
Conventional diesel truck parameters.mat	Model parameters
Conventional natural gas truck.slx	System model build
Conventional natural gas truck parameters.mat	Model parameters
Baseline diesel truck testing over the Highway-2 drive cycle.drawio	Figure 2.1

Continued on next page

Table E.2 (continued)

File name	File description
BSFC map of the Cummins X15 diesel engine.opju	Figure 2.2
Powertrain configuration of the conventional Class 8 truck.drawio	Figure 2.3
Conventional diesel baseline validation.opju	Figure 2.4
BSFC map of the 13 L HPDI NG engine.opju	Figure 2.5
Engine-out emission maps of the 13 L HPDI NG engine.opju	Figure 2.6

E.3 Chapter 3 Source Files

Table E.3: Source files of Chapter 3.

File name	File description
Parallel hybrid truck.slx	System model build
Parallel hybrid truck parameters.mat	Model parameters
Power split hybrid truck.slx	System model build
Power split hybrid truck parameters.mat	Model parameters
Plug in hybrid truck.slx	System model build
Plug in hybrid truck parameters.mat	Model parameters
Designed P ₂ hybrid electric powertrain configuration.drawio	Figure 3.1
Efficiency map of the MG1 in the P ₂ hybrid electric configuration.opju	Figure 3.2
Torque-speed curves of the rule-based EMS in the P ₂ hybrid electric configuration.opju	Figure 3.3
Designed power-split hybrid electric powertrain configuration.drawio	Figure 3.4
Efficiency map of the MG2 in the power-split hybrid electric configuration.opju	Figure 3.5
Designed series plug-in hybrid electric powertrain configuration.drawio	Figure 3.6
Efficiency map of the generator in the series plug-in hybrid electric configuration.opju	Figure 3.7

E.4 Chapter 4 Source Files

Table E.4: Source files of Chapter 4.

File name	File description
DOE optimization algorithm.m	Component sizing framework
GA optimization algorithm.m	Gearbox optimization framework
Optimal component sizing framework.drawio	Figure 4.1
Genetic algorithm optimization framework.drawio	Figure 4.2
Standard drive cycles profiles.opju	Figure 4.3

E.5 Chapter 5 Source Files

Table E.5: Source files of Chapter 5.

File name	File description
Grade vs distance calculation.m	Grade calculation algorithm
Steep highway information.xlsx	Road profile data
Conventional baseline model performance over the Davis Dam Road.opju	Figure 5.1
Geographical maps of the identified worldwide steep highways.drawio	Figure 5.2
Road grade profile calculation process.png	Figure 5.3
Dry weight estimation of diesel engines.opju	Figure 5.4
P ₂ hybrid model performance over the Davis Dam Road.opju	Figure 5.5
Grade versus distance profiles of the identified steep highways.pdf	Figure 5.6
Modified engine rated power of the conventional diesel model.opju	Figure 5.7
Refined component sizing results for the P ₂ hybrid design.opju	Figure 5.8
Fuel consumption of the conventional and P ₂ hybrid designs.opju	Figure 5.9
Engine BTE of the conventional and P ₂ hybrid designs.opju	Figure 5.10
Engine operating points of the conventional and P ₂ hybrid designs.opju	Figure 5.11

E.6 Chapter 6 Source Files

Table E.6: Source files of Chapter 6.

File name	File description
NO _x emissions map of the 13 L HPDI NG engine.opju	Figure 6.1
BSFC map of the 13 L HPDI NG engine with high CR.opju	Figure 6.2
BSFC map of the 13 L HPDI NG engine with advanced SOL.opju	Figure 6.3
NO _x emissions map of the 13 L HPDI NG engine with advanced SOL.opju	Figure 6.4
BSFC map of the 13 L HPDI NG engine with TC.opju	Figure 6.5
BSFC map of the 13 L HPDI NG engine with TC and high CR.opju	Figure 6.6
BSFC map of the Downsized NG engines.opju	Figure 6.7
NO _x and CH ₄ emission maps of the 9 L HPDI NG engine.opju	Figure 6.8
BSFC map of the 9 L HPDI NG engine with TC and high CR.opju	Figure 6.9
Engine operating points of the conventional HPDI NG model.opju	Figure 6.10
Engine operating points of the hybridized models with technology-enhanced HPDI NG engines at full cargo load.opju	Figure 6.11
Engine operating points of the hybridized models with technology-enhanced HPDI NG engines at partial cargo load.opju	Figure 6.12
Engine operating points of the power-split and P ₂ hybrid models with the 13 L HPDI NG engine.opju	Figure 6.13
Energy flow diagrams of the conventional and hybridized models with HPDI NG engine variants.drawio	Figure 6.14
Tailpipe GHG emissions of the conventional and hybridized model with HPDI NG engine variants.opju	Figure 6.15

E.7 Chapter 7 Source Files

Table E.7: Source files of Chapter 7.

File name	File description
Conventional hydrogen truck.slx	System model build
Conventional hydrogen truck parameters.mat	Model parameters
Parallel hybrid hydrogen truck.slx	System model build
Parallel hybrid hydrogen truck parameters.mat	Model parameters

Continued on next page

Table E.7 (continued)

File name	File description
BSFC map of the 13 L HPDI H ₂ engine.opju	Figure 7.1
CO ₂ emissions map of the 13 L HPDI H ₂ engine.opju	Figure 7.2
Schematic of the on-board H ₂ compression system.jpg	Figure 7.3
P ₂ hybrid powertrain configuration with the HPDI H ₂ engine.drawio	Figure 7.4
Fuel consumption of the conventional and P ₂ hybrid models with the HPDI H ₂ engine.opju	Figure 7.5
Engine BTE of the conventional and P ₂ hybrid models with the HPDI H ₂ engine.opju	Figure 7.6
Engine operating points of the conventional and P ₂ hybrid models with the HPDI H ₂ engine.opju	Figure 7.7
On-board H ₂ compression system performance.opju	Figure 7.8
Energy flow diagrams of the conventional and hybridized models with HPDI H ₂ engine.drawio	Figure 7.9