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Master's Degree in Mechanical Engineering

Master's Degree Thesis

Development of a Virtual Test Rig for Evaluating Hybridisation Potential in Non-Road Mobile Machinery

Supervisors:
Prof. Luciano ROLANDO
Dott. Afanasie VINOGRADOV

Candidate:
Fiorelo ELMAZAJ

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Abstract

Non-Road Mobile Machinery (NRMM) accounts for about 3% of total CO_2 emissions in both the U.S. and the EU. Due to their high-power demand and long operating missions in harsh environments, NRMMs are considered a hard-to-abate sector. Therefore, a pragmatic pathway toward decarbonization must involve a combined approach that integrates a wide range of technologies. Among them, powertrain hybridisation can play a crucial role in reducing CO_2 and pollutant emissions, leveraging the experience gained from the electrification of passenger cars and commercial vehicles.

Within this framework, the present thesis aims to develop a virtual test rig to assess the potential CO_2 emission reductions of a skid-steer loader. This case study is particularly challenging due to the machine's versatility in performing diverse tasks and using multiple implements, which increase the variability of its duty cycles. A digital twin of the conventional powertrain was developed using the commercial software GT - SUITE, adopting a 1D modelling approach for both the hydraulics and the vehicle dynamics. This baseline configuration was established to benchmark the performance of an electrified powertrain across a wide range of mission profiles. Subsequently, the powertrain was electrified using a parallel P1 configuration, selected for its ability to enable engine downsizing, electric boosting, and load point shifting with minimal impact on the original layout.

The numerical simulations carried out on the proposed hybrid powertrains, featuring different degrees of electrification, indicated reductions in fuel consumption and associated CO_2 emissions for the investigated mission profiles. For example, a simple 48 V P1 configuration with one electric motor and a small battery achieved around 10% fuel savings, while the most electrified high-voltage layouts, with up to four electric machines for traction and hydraulics, exceeded 30% reduction. These improvements arise from a combination of engine downsizing, more favourable operating points and reduced hydraulic losses, but they also require higher system complexity, increased voltage levels and a larger number of electrical components.

Furthermore, the thesis highlights that the developed digital twin represents a valuable tool to support the optimisation of powertrain architectures for this class of machinery.

Acknowledgments

I would first like to thank my academic supervisors, Professor Luciano Rolando and Dottore Afanasie Vinogradov, for their guidance, patience, professionalism and encouragement throughout this project. Their deep technical insights and constant availability were essential to the successful development of this thesis and to my growth as an engineer.

I would also like to thank the entire faculty and staff of Politecnico di Torino for providing an excellent academic environment throughout my years here and for helping me fulfil my goal of becoming an engineer.

Do të doja të falënderoja të gjithë familjarët e mi, të cilët i dua shumë. Ju falënderoj për tërë këto vite mbështetje, që ju kam pasur pranë dhe të gatshëm për çdo problem. Ju falënderoj që jeni mbledhur sot nga anë e anës për të festuar këtë moment të veçantë në jetën time.

Një falënderim special dua të bëj edhe për motrat e mia, njëra që është doktoreshë dhe njëra që do të bëhet. Ju dua shume, dhe shpresoj të më asistoni pa pagesë monetare në lidhje me problemet shëndetësore që mund të kem në të ardhmen.

Një falënderim i veçantë shkon edhe për prindërit e mi, të cilët më kanë mbështetur gjithmonë dhe më kanë bërë njeriun që jam sot. Jam i sigurt që nuk ua kam bërë gjithmonë të lehtë, por ju falenderoj që keni ndenjur gjithmonë pranë meje. Jeni shembulli i njeriut që aspiroj të jem një ditë.

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Chapter 1

Introduction

Non-Road Mobile Machinery (NRMMs), commonly referred to as Off-highway or Off-Road machinery are machines engineered for specific tasks in construction, mining, agriculture, forestry and industrial sectors. These machines are essential for maintenance, production, and economy of the modern world, yet they often operate under duty cycles that are far from the engine's optimal efficiency region. As a result, NRMMs can contribute disproportionately to fuel use and local emissions compared to their share of the total vehicle fleet. Due to these characteristics, the interest to reduce CO_2 and pollutant emissions, as well as fuel consumption, has expanded from the on-road sector to the wide family of Non-Road Mobile Machinery (NRMM).

Recent analyses of non-road mobile machinery emissions indicate that Off-Highway equipment in the European Union is responsible for about 108 Mt of CO_2-equivalent per year, corresponding to roughly 3.1 % of the EU's total greenhouse gas emissions [16].

An earlier market analysis for the NRMM sector reports that NRMMs is responsible for "roughly 100 million tons of CO_2-equivalent annually," about 2% of total EU GHG emissions, and notes that in the EU, NRMMs contributes about 15% of total NOx and 5% of total PM emissions [17].

Similar figures are reported in the United States, where NRMMs contribute to about 3% of Greenhouse gas emissions, as illustrated in figure 1.1.

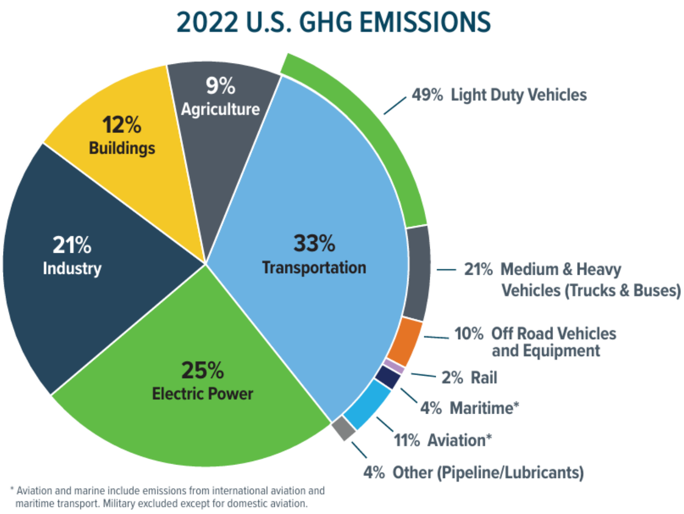

Figure 1.1: EPA Greenhouse Gas Emissions [1]

NRMMs often work at low vehicle speed with highly variable load: digging, lifting, loading, short travel, lots of idling, etc. Therefore, these machinery's engines spend a considerable amount of time idling, or at low load, and at high speeds, where Brake Specific Fuel Consumption (BSFC) is not the best. Undergoing rapid load changes also hurts efficiency, and such is the case for these machinery.

Using the experience gained from the electrification of passenger cars and commercial vehicles, we can leverage that knowledge to improve many aspects of NRMM operations.

Adding an electric component can reduce consumption and emissions, but also regain some energy during operations, so the overall efficiency increases. Hybrid powertrains can provide improved fuel efficiency, reduced emissions, regenerative braking, and better operational control. They allow for downsizing, while not losing on performance.

Other benefits of powertrain hybridization include reduced wear and tear on the powertrain, smother operation of engine, improved performance of the machinery and lower operating costs.

1.1 Non Road Mobile Machinery Electrification

In the European Union, the Stage V regulation for Non-Road Mobile Machinery imposes strict limits on CO, NO_x, and PM emissions for engines in the range of 19–560 kW, as summarized in table 1.1. These limits aim to reduce the environmental footprint of off-road equipment.

Ign.	**Power** [kW]	**CO** [g/kWh]	**HC** [g/kWh]	**NO$_x$** [g/kWh]	**PM** [g/kWh]	**PN** [1/kWh]
CI	$P < 8$	8.00	7.50^a	-	0.40^b	-
CI	$8 \leq P < 19$	6.60	7.50^a	-	0.40	-
CI	$19 \leq P < 37$	5.00	4.70^a	-	0.015	1×10^{12}
CI	$37 \leq P \leq 56$	5.00	4.70^a	-	0.015	1×10^{12}
All	$56 \leq P < 130$	5.00	0.19	0.40	0.015	1×10^{12}
All	$130 \leq P < 560$	3.50	0.19	0.40	0.015	1×10^{12}
All	$P > 560$	3.50	0.19	3.50	0.045	-

Notes: ^{a} HC+NO$_x$; ^{b} 0.60 for hand-startable, air-cooled direct injection engines.

Table 1.1: Stage V emission standards for off-road engines [15]

Engines for NRMM applications are tested using a standardized duty cycle called a Non-Road Mobile Transient Cycle (NRTC), shown in figure 1.2. In this test, a normalized, predefined speed and torque working profile is applied to the engines, once in a cold start, and once in a hot start. The resulting emissions from both tests must not exceed the values shown in the table 1.1.

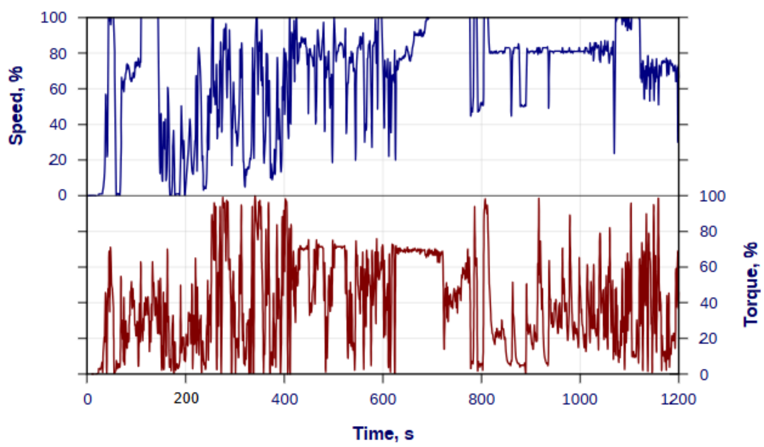

Figure 1.2: NRTC Test (normalized torque and speed profile)

Although the push for electrified powertrains may not be primarily from emission regulations at the moment, the potential fuel savings as well as productivity, controllability, noise reduction, and enabling new machine functions are strong electrification drivers. NRMMs can also benefit from the advancements done in the passenger vehicles, and can use that experience to implement these systems easier and without decades of research and development.

1.1.1 Off-Highway Challenges

Unlike passenger vehicles, in Off-highway applications, a unique set of challenges in hybridizing are presented.

For one, since the machinery and their work varies a lot in this sector, hybrid powertrain sizing, and the methodology used to implement it varies between different typologies of machinery.

These challenges in designing a unified approach are extended to the working cycle or mission profile. No unified working cycle exists across all applications, and this is again due to the vastly different architectures and work these machinery do. However, the simulation approach can be an important element in increasing the efficiency of the machinery, since in simulations all the particular elements and sub-system of Off-Highway machinery can be looked at individually, and optimizations can be made.

Another challenge this sector faces is the powertrain. Off-Highway machinery traditionally use a diesel engine as the primary driver. Diesel engines are preferred since they can provide the low-end torque, reliability, and productivity needed in these machinery. Diesel engines however, are already quite efficient. Therefore, the strategies on reduction of emissions cannot be based solely around the optimization of operations of the diesel engine, but rather, adding a degree of electrification. Hybridization also helps the engine work around a more efficient region in its map.

1.2 Aim and proposed Methodology

Within this framework, the main aim of this thesis is to develop a virtual test rig for assessing the potential CO_2 emission's reduction and performance improvements in Off-highway are achievable through powertrain hybridization.

Vehicle modeling and simulation of operations is very useful when experimental data are unavailable or where not enough research has been made on the different typologies and scenarios. This approach also allows for better optimization of the vehicle systems, and power distribution. For this reason, powertrain modeling is used widely in the industry.

A digital twin of the conventional powertrain machinery is initially developed using commercial software GT - SUITE and adopting a 1D modeling approach for both the hydraulics and the vehicle dynamics. This baseline configuration is used to establish a benchmark in terms of the working cycle torque and power requirements, as well as provide a baseline of the performance to later compare that to the performance and results of an electrified powertrain.

The modeling approach that was adopted is forward-facing. While backward-facing models can estimate fuel use across defined cycles, they rely on idealized control assumptions, limiting their value for dynamic studies [7]. Forward facing simulations also offer a more realistic approach to casual system interactions. This is because, unlike backward facing models, the working cycle in a forward-facing model is considered to be a target rather than an input. They replicate the behaviour of a real machine, in which the command is given, and then the machine does the movement. Therefore they reproduce causal system interactions, providing the means to study control strategies, component dimensioning, and transient machine operation [7]. Case studies on compact wheel loaders and skid-steer loaders confirm that forward-facing methods are capable of showing fuel savings and representing dynamic effects more realistically [18][11]. Other studies also suggest that in hybrid off-highway contexts, where duty cycles are irregular and highly transient, forward-facing modeling is particularly advantageous [19].

A forward model of a Off-highway machinery was built, particularly a Skid-Steer Loader model, and then used to evaluate the performance of each hybrid architecture, starting from simpler and easier to implement ones like P1, with increasing levels of electrification, and internal combustion engine downsizing.

The virtual test rig is finally used to compare different hybrid configurations and operating strategies, quantifying potential fuel savings over a realistic working cycle and highlights the most influential design and control parameters.

The thesis is organized in the following chapters:

- **Chapter 2** provides an overview of the NRMM sector, examples of hybridization in the sector and how they were implemented, current market availability of electrified powertrain NRMMs, and presents the Tracked Skid-Steer Loader as the case study.
- **Chapter 3** shows the model implementation of the the skid-steer loader, based on a forward model, and the integration of the hybrid component, showing the power and torque requests of each component based on the conventional mode.
- **Chapter 4** analyses the results, comparing the different implemented architectures, as well as component sizing and their influence on the hybridization effect.
- **Chapter 5** discusses the main outcomes and future works.

Chapter 2

Theoretical Background: Non-Road Mobile Machinery

According to the Vehicle Certfication Agency, Non-Road Mobile Machinery designation encompasses a broad category of equipment powered by internal combustion engines, including construction, agricultural, and industrial machines that are not intended for use in public roads [20]. They are generally grouped based on their field of use. While they do share some characteristics, the working cycles, as well as power requirements from each machinery field of application differs largely, and thus, the powertrain will be specific to the type of machinery and application. The largest vehicle type contributors to the market, categorized by their fields of application, are:

- **Construction and earthmoving machinery:** This category includes excavators, wheel loaders, bulldozers, backhoe loaders, motor graders and similar equipment used for excavation, grading, trenching and material transport, often in harsh environments and on unprepared terrain.
- **Mining machinery:** This group comprises large off-highway trucks, mining excavators, electric rope shovels and other equipment used in surface and underground mining operations, typically operating with very high payloads and continuous duty cycles.
- **Agriculture and forestry machinery:** This category covers tractors, combines, forage harvesters, sprayers, skidders and forwarders used for soil preparation, crop cultivation and harvesting, as well as timber extraction and transport.
- **Material handling machines:** These include forklifts, telehandlers, reach stackers and container handlers used in warehouses,

ports and industrial sites for lifting, stacking and transporting pallets, containers and other unit loads.

- **Municipal and property maintenance:** This group comprises sweepers, compact loaders, mowers, snow ploughs and similar equipment used for street cleaning, winter maintenance, landscaping and other public service tasks in urban and suburban areas.

Construction and Earthmoving Machinery

Construction and Earthmoving machinery represent one of the largest and most diverse groups of NRMMs, not only in application, but in typologies as well. In the US, they also represent the largest contributor to Greenhouse gas emissions among the NRMM category, as shown in figure 2.1.

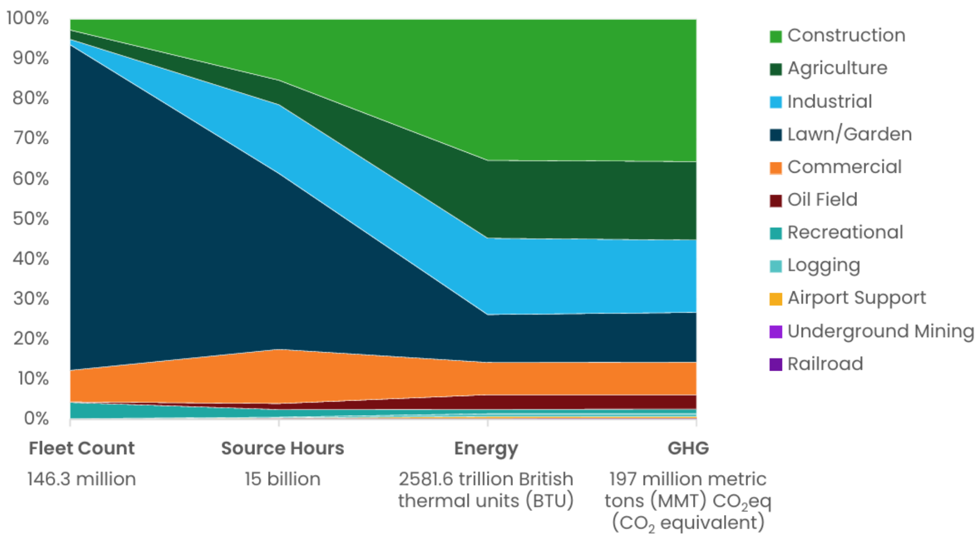

Figure 2.1: U.S. off-road equipment composition based on percent of fleet number, source hours, energy, and use-phase GHG emissions [1]

This category of NRMMs is designed for excavation, material transportation, grading, trenching, often in harsh and unsuitable terrain and conditions. Their cycles are characterized by highly transient operations, frequent and sudden direction changes, repeated short travel phases, and prolonged periods of stand-by or idling. It includes vehicles like Crawler or wheeled excavators, backhoe loaders, wheeled or articulated loaders, bulldozers, Skid-Steer Loaders, etc.

Wheeled Loader

The Wheeled Loader or Articulated Loader is a very important machine in the field of construction and a very widely used example in the field of hybridization. They traditionally utilize a driveline based on a torque converter with a powershift transmission, and more recently have started to utilize CVTs (Continuously Variable Transmission), or hydromechanical transmission. Therefore, the engine has a slightly higher degree of freedom from the vehicle speed and kinematic demands. Their duty cycle however still contains strong transients and idling.

Figure 2.2: Wheeled Loader completing its working cycle

The Wheeled Loader is a particularly excellent reference point in this study, as not only has the electrification of these vehicles begun, they are also very well studied and documented in terms of working cycle, consumption and other operational parameters.

(a) Liebherr mid-sized wheel loader. Source: Liebherr [21].

(b) Volvo wheeled loader in operation. Source: Volvo CE [22].

Figure 2.3: Examples of Loaders performing different duties

The classification of wheel loaders used in construction is shown in table 2.1.

Size class	**Weight [t]**	**Power [kW]**	**Engine disp. [L]**
Compact	4–10	35–80	3–4
Medium	10–25	100–220	4.5–7
Large	25–35	220–350	8–15

Table 2.1: Construction wheel loader size classes

Backhoe Loader

Backhoe loaders are use to bring multifunctionality on construction sites, especially when both material handling and shallow to medium-depth excavation are required within the same job. The machine integrates a front-mounted loader bucket for loading, carrying, and grading tasks much like a wheel loader, and a rear backhoe boom for digging and trenching operations, like an excavator. Thanks to this dual functionality, backhoe loaders are widely used in urban works, utility installation, road maintenance, and small-to-medium construction projects, where deploying two separate machines would be inefficient. Compared to dedicated wheel loaders or excavators, backhoe loaders typically offer lower peak productivity in each individual task, but compensate by reducing site footprint, and idle time associated with machine changes.

Figure 2.4: Caterpillar backhoe loader in operation. Source: Caterpillar Inc.[2]

Excavators

Excavators are the some of the most versatile and widely used machines within this category of NRMMs. They are available in several typologies with distinct driveline layouts and hydraulic architectures.

The most common, the Crawler excavator, utilizes an internal combustion engine, which drives the hydrostatic transmission, as well as the working pumps, and auxiliaries pumps. Modern hydraulic excavators can be equipped with a wide range of attachments, including digging and grading buckets, hydraulic breakers (shown in figure 2.5b), crushers, shears, grapples, augers and forestry tools, which allows the same base machine to perform excavation, demolition, material handling and vegetation management tasks with minimal reconfiguration.

(a) Liebherr generation 8 crawler excavator in earthmoving application. Source: Liebherr [23].

(b) Caterpillar crawler excavator with hydraulic hammer attachment. Source: Fiore Ricambi ShPK

Figure 2.5: Examples of different uses and attachments.

The second type of excavator, the wheeled excavator, shown in figure 2.6a, has wheels and driven by a normal mechanical transmission, or hydro-mechanical transmission. These types of excavators are used when the machine needs to move at speeds higher that the typical crawler, and generally employed in construction sites where excessive damage to the surface needs to be avoided.

(a) Volvo EW145B Prime wheeled excavator. Source: Volvo Construction Equipment [24].

(b) Komatsu PC240LC/NLC-11 crawler excavator. Source: Komatsu [25].

Figure 2.6: Main excavator typologies

Excavators are categorized based on their weight, as shown in table 2.2.

Size class	Weight [t]	Engine Power [kW]	Engine disp. [L]
Micro	< 1	$\sim$ 7–15	$\sim$ 0.5–1.0
Mini / Compact	1–6	$\sim$ 10–40	$\sim$ 0.8–2.5
Midi	6–10	$\sim$ 40–75	$\sim$ 2.5–4.0
Standard / Full-sized	10–45	$\sim$ 70–250	$\sim$ 4.0–7.5
Large / Heavy-duty	45–80	$\sim$ 250–400	$\sim$ 9–15
Ultra-large / Mining	> 80	> 400	> 15 (up to $\sim$ 50)

Table 2.2: Typical crawler excavator size classes by operating weight, engine displacement, and net power (representative ranges).

Mining Machinery

In mining, some vehicle types - such as large excavators - are also widely used in construction, in addition to dedicated mining trucks and other equipment. These machines operate exclusively off-road, often in continuous or near - continuous duty cycles with very high payloads and power demands.

The mining machines are much larger than the ones used in general construction, with a mining excavator being able to produce up to 2500 kW power from 80000 cm^3, V-16 engines, and weight being from 300 to 1000 tons [26].

(a) Caterpillar 793D mining truck [27].

(b) Liebherr R 9600 mining excavator [26].

Figure 2.7: Examples of large-scale mining machinery: (a) haul truck and (b) hydraulic mining excavator.

Hybridization is already well established in mining NRMMs, where diesel–electric powertrains are now common; consequently, many modern haul trucks, wheel loaders, and excavators integrate multiple electric machines for propulsion and auxiliary functions.

Agriculture and Forestry Machinery

These include vehicles like the tractor, but also other machinery like combines, skidders, and forwarders. They are typically used for field preparation, crop harvesting, material handling and timber extraction, often operating at low speeds with highly variable engine loads, which makes their duty cycles particularly relevant from an energy - efficiency and emissions perspective.

(a) Combine (b) Tractor

Figure 2.8: NRRMs for Agricultural Use: Tractor and Combine [3]

The tractor is an essential machinery in this subcategory. Tractors are often looked at when considering hybridization, due to their wide spread use, but also since they contribute about 85% of the Greenhouse gas emissions within the agricultural sector. Conventionally they utilize engines that range from 15 kW to 400 kW. They also come in different typologies, from wheeled tractors, tracked tractors, and articulated high-power tractors, which are optimised for different soil conditions, traction requirements, and field operations.

While the biggest power consumer in a tractor is traction, they also typically provide power to implements through the Power-Take-Off (PTO) and hydraulic circuits, meaning that propulsion and working loads often overlap during real agricultural duty cycles. This makes their energy demand highly variable and strongly dependent on the specific implement and task. Their variable load demand and long operating hours make tractors an important target for hybrid solutions, especially when PTO and traction loads overlap.

Material Loading and Handling machines

Material Handling machines are represented by forklifts, and the larger telehandler. They are used for transporting large weights in worksites or magazines and have a telescopic arm and back-wheel (forklifts) or four-wheel steering which are useful in stacking weights in tight positions.

(a) Telehandler. Source: Merlo

(b) Forklift. Source: Linde

Figure 2.9: Examples of NRRMs material handling: Forklift and Telehandler

In the case of Material Handling machines, the electrification has been developing for a while, especially in the case of forklifts, which for indoors use must utilize a fully electric powertrain, to avoid harmful gases.

2.1 NRMM Electrification

Beyond the regulatory and operational drivers introduced in Chapter 1, Off-Highway hybridization must satisfy several practical constraints before it can be commercially viable:

- **Reliability**: The machine needs to ensure constant and stable operations, as well as durability. The machines of this sector can cost up to hundreds of thousands of euros, and therefore need to have the reliability and guarantee that comes with such a purchase. They also work in very difficult conditions and often in extended or even around the clock operations. Low air quality environments are also common. For example, excavators need double-heavy duty air filters for better filtering, but also to avoid tears to the filter due small rocks or other terrain particles.
- **Performance**: This means that hybridization of the machine at the bare minimum doesn't impede the machine's performance compared to a similar machine of the same size and overall power output. Ideally, the hybridization process should not only reduce the emissions, but also able to perform better than before. With varying levels of success, it should offer the same working capacity (be that digging, bulldozing, transporting piles of material, etc.), work

more smoothly, enable functions such as energy regeneration, provide faster transient response during peak-load events, preserve or improve cycle times, and ensure consistent tractive and hydraulic capability across the working envelope. In other words, any hybrid solution must guarantee no loss in productivity (e.g., tonnes moved per hour or m^3 excavated per hour), while potentially improving drivability, controllability, and operator responsiveness under highly dynamic loads. This can enable the potential buyer to spend extra, and get the hybrid version instead of the conventional one.

- **Longevity**: The machine must be able to work for a specified amount of total hours, which is comparable to the conventional powertrain machines. In the case of excavators for example, the manufacturer typically must be able to guarantee about 10000 to 15000 hours of work, and around 3000 hours of work in the powertrain under warranty conditions. If major overhauls need to be made before this period the producer will incur a loss on the warranty.
- **Battery Life**: Expanding on the previous point, in a hybrid, this requirement extends to the battery pack: it must be sized and managed so that it can withstand a similar number of operating hours in charge-sustaining use without unacceptable loss of capacity or power, otherwise the manufacturer would incur additional costs due to premature component replacement under warranty.

2.1.1 Mission Identification

As briefly mentioned in Chapter 1, Off-Highway present a different set of challenges, compared to passenger vehicles. One of the biggest ones are lack of testing cycles.

Passenger vehicles are tested using the a myriad of laboratory or real world driving cycles.

Testing Cycles of Road Vehicles

- **WLTP (Worldwide Harmonised Light Vehicles Test Procedure):** Standardized chassis-dynamometer laboratory test used mainly in the EU to certify official CO_2/fuel consumption and regulated pollutant emissions. The official CO_2 fuel-consumption values shown for EU cars come from WLTP [28][29].
- **RDE (Real Driving Emissions):** EU on-road test performed in real traffic using a PEMS unit (Portable Emissions Measurement System) to verify that NO_x and particle-number emissions remain within limits outside laboratory conditions. RDE is not used to

publish official CO_2 values, but it is used to ensure real-world pollutant compliance (NO_X/PN conformity factors) [28][30].

- **FTP-75 (Federal Test Procedure 75):** Main US city/urban chassis-dyno certification cycle, including cold start, stop-go driving, and hot start phases; forms the core of US pollutant and CO_2 certification [31].
- **Cold FTP (Cold-start FTP-75:** US cold-temperature version (run at $-7\,°C$) of the urban cycle to capture winter penalties on emissions and fuel consumption [32].
- **HWFET (Highway Fuel Economy Test):** US highway chassis-dyno cycle with steadier speeds, paired with FTP-75 to determine official fuel-economy and emissions ratings [33].
- **US06 (Supplemental Federal Test Procedure US06):** US high-speed/aggressive chassis-dyno cycle representing hard accelerations and higher speeds, added to better capture real-world driving effects [32].
- **SC03 (Supplemental Federal Test Procedure SC03):** US air-conditioning hot chassis-dyno cycle conducted at high ambient temperature with A/C on, used to quantify added fuel use and emissions from HVAC loads [32].

Testing Cycle of Off-Highway Vehicles

Off-Highwat Machinery lack such standardized cycles. While engine test cycles like the NRTC do help regulate emissions from such vehicles, its difficult to prescribe a one-size-fits-all testing cycle to such different machinery. This is because, unlike passenger vehicles, Off-Highway machinery have different mission profiles, and therefore different power demands and powertrain sizing. For example, an excavator will mostly stay in one place during it's working cycle, and overlap between the two primary power consuming subsystems is not as common as in loaders. On the other hand, machinery like loaders and tractors need to be constantly moving, as well as operate any other attachments necessary for completing their working cycle. This means that they also have different power demands, and power distribution in their subsystems. This means that, in the case of NRMM applications, no unified approach can be established.

Nevertheless, some test cycles like the US Environmental Protection Agency Non-Road Duty Cycles do exist, as mentioned in Chapter 1. However, these tests are not used for regulatory purposes, they are only used to estimate real-world CO_2, NO_X, PM, fuel consumption. In figure 2.10 testing cycles of different Off-Highway Machinery are depicted. While under the same category of NRMMs, they couldn't be operating more differently from each-other. If anything, they showcase how difficult it is to create a unified approach in testing these machines.

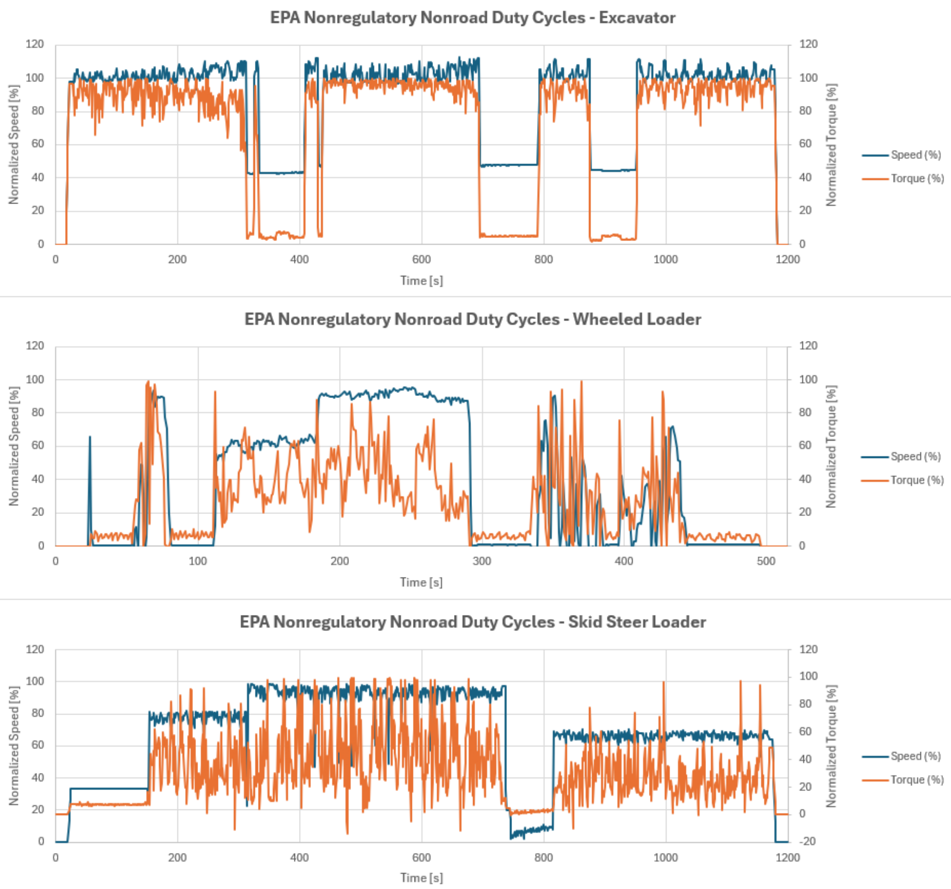

Figure 2.10: EPA Non-Road Duty Cycles for three different NRMM typologies: Tracked Excavator, Wheeled Loader, and Skid-Steer Loader [4]

2.1.2 EMS - Energy Management System

In a hybrid system, since there is no longer one primary driver, certain control strategies need to be implemented to decide the operating mode of each powertrain component. This is done by the Energy Management System (EMS) unit. The EMS is part of the powertrain hierarchy of the powertrain control that has to define the operating modes and/or power split strategies between different sources.

The control hierarchy of the EMS in automotive applications is shown in figure 2.11.

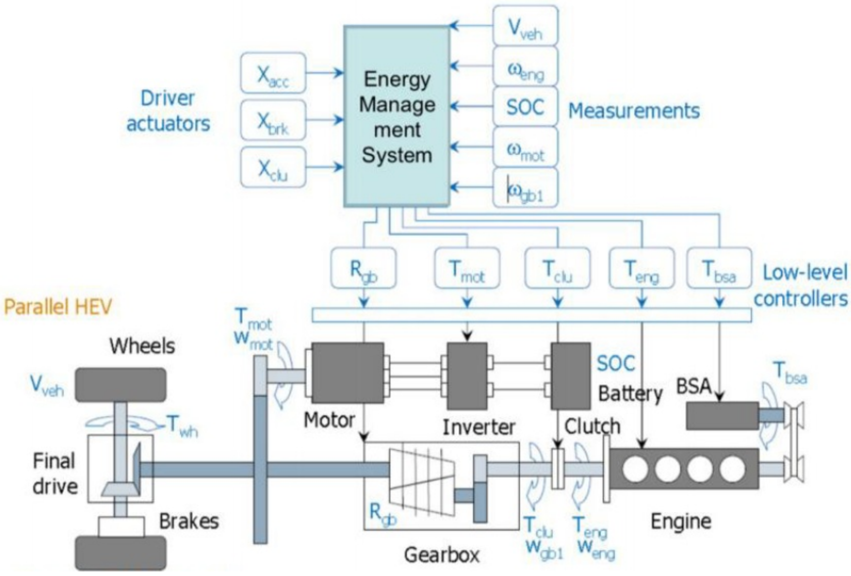

Figure 2.11: EMS Hierarchy in road going parallel hybrid vehicles [5]

The duties of the EMS in automotive applications are:

- In case of complex powertrains, it should select the most suitable operating mode, depending on power request, engine conditions and other operational and environmental parameters.
- For each operating mode it needs to define the power split among the power users.
- The EMS receives information from the interface of the vehicle, such as the accelerator, clutch and brake pedals to understand the conditions. It will interact also with low level controller (controller of each individual system for example, engine, battery, etc).
- Other secondary optimisation of other systems of the vehicle, such as thermal management, drive mode selection, etc.

The EMS is made out of two main domains: the Supervisory control decides the best operating mode (ICE only, EV, etc.) based on the driver demand and operating conditions of the components (ex. battery SOC).

The Energy Management control splits the power among the engine and electric motors in order to satisfy the overall power demand, based on the mode chosen by the Supervisory Control.

Depending on the computational resources of the vehicle and desired level of optimality, several energy management strategies can be implemented:

- **RB - Rule Based**: This is the simplest control algorithm implemented in EMS. It's based around thresholds, and state logic. These include engine on/off, state of charge window, engine maps, and so on. The thresholds or rules used in defining the control are generally due to experience or real world examples of the programmer. However, more sophisticated methods of procuring said rules do exist. One such EMS is Sub-Optimal Rule Based, where the rules are derived from using Dynamic Programming and then implemented the rules for similar scenarios. Rule Based is fairly simple to implement, and can be implemented in real time.
- **ECMS - Equivalent Consumption Minimization Strategy**: This is a more complex EMS control strategy. It centres around a optimisation that is done by converting the energy usage of the hybrid powertrain into an equivalent fuel consumption, and trying to minimize the total fuel consumption of the car. Variants of ECMS such as A-ECMS (Adaptive ECMS), where the equivalence factor used to get the equivalent fuel consumption can be tunned online (unfeasible), or offline, and then implemented to match similar patterns as compared to the ones with which it was programmed with.
- **MPC - Model Predictive Control**: This control strategy goes a step further by formulating a short-horizon control problem that is solved repeatedly in real time, relying on a system model to forecast near-future power demands and enforce component and state constraints.
- **DP - Dynamic Programming**: This is a global optimisation strategy, in which the dynamic nature of the system is considered for the optimisation and an optimal solution is found over a predefined driving cycle, which must be known before. Therefore, while this offers the optimal strategy, it cannot be implemented online, and can be used only for calibration of other heuristic strategies, such as Optimal Rule Based, or to give a benchmark to compare with strategies such as ECMS and Adaptive ECMS.

The EMS used in this case is a simple Rule Based Control. In RB Control, the energy management algorithm focuses mainly on engine characteristics. The definition of rules is highly dependent on the powertrain architecture. This energy management strategy is beneficial because it can work in real-time conditions, and has a low computational requirement. However, the control is not optimal and the calibration can be long.

2.1.3 Hybridization Examples

Despite strong ongoing development toward hybrid NRMM powertrains, commercially available hybrid machines are still scarce. At the same time, several production models have entered the market, and a growing number of OEMs are advancing prototypes toward pre-production and series manufacturing.

Excavators:

Excavators are some of the most successful in hybridization because of its capabilities to regenerate some of the lost energy. Hybrid concepts, especially those targeting swing energy recuperation and hydraulic load buffering, have reached series production earlier in excavators than in many other machine types.

Komatsu was the first OEM to begin R&D on hybrid construction machinery in 1997 and developed the first hybrid excavator in 2008, with a reported 25% fuel savings and improved performance [8]. A few examples of hybrid excavators from Komatsu are the Komatsu PC200-8 Hybrid and the Komatsu Hybrid Crawler Excavators series.

Hitachi ZH210LC-5 Hybrid combines a hydraulic swing motor with an electric motor and capacitor/battery storage (TRIAS HX concept), targeting swing-energy recovery and engine assist; Hitachi literature and press materials shows roughly up to 31% fuel and CO_2 reduction compared to non-hybrid versions [6].

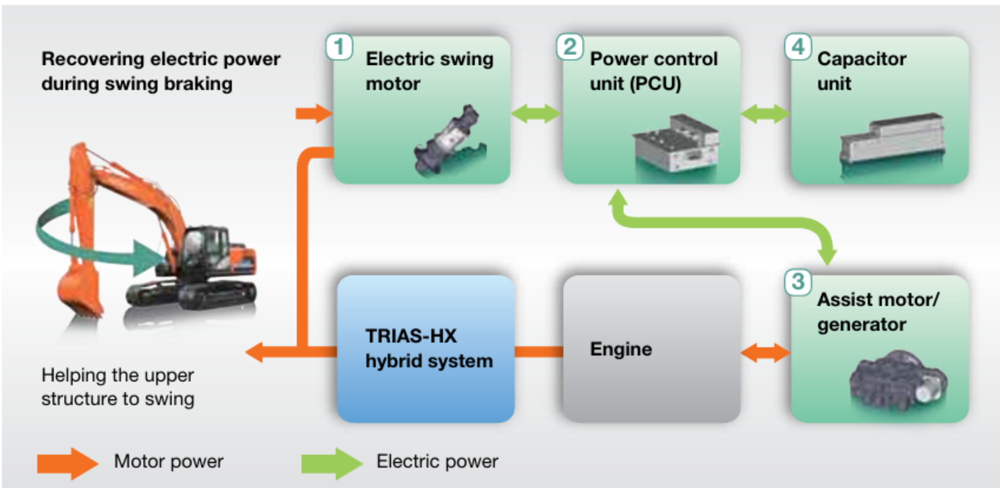

Figure 2.12: Hitachi ZH210LC-5B Hybrid Excavator [6]

Kobelco continues to develop parallel/compound hybrids as a core product family. Their SK200H-10 / SK210HLC-10 Hybrid integrates electrical swing regeneration with battery storage and hydraulic energy-saving circuits. Sumitomo offers a comparable production machine, the SH200HB-7 Hybrid [34].

Two other examples of hybridization implementation used in excavators is shown in figure 2.13. Komatsu, shown on the left, uses a parallel architecture for traction and working hydraulics, while utilizing an electric motor for the swing motor and for energy recovery during boom lowering. On the other hand, Hitachi (shown on the right) utilizes a parallel architecture for the working hydraulics and series for the cabin swing and traction.

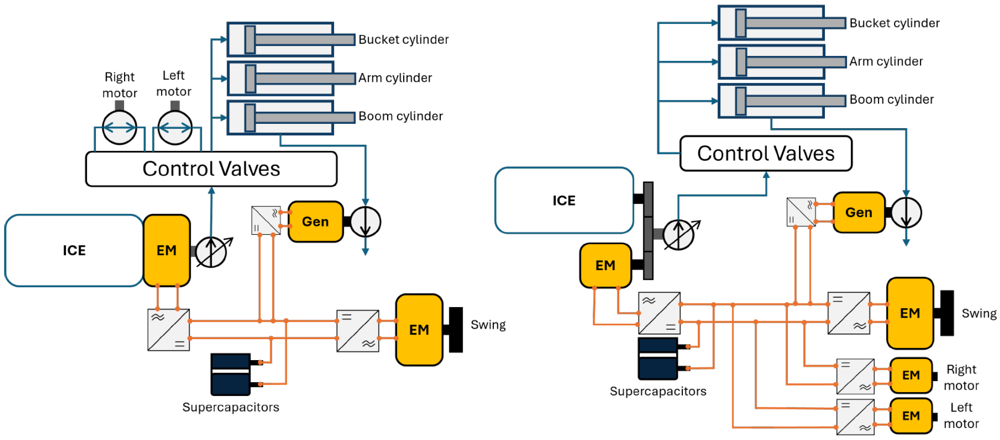

Figure 2.13: Hybrid configurations for excavators: Komatsu (Left); Hitachi (Right)[7] [8]

Wheel Loaders:

Volvo L220F is one of the earliest examples of Parallel architecture in a wheel loader. The core of the system was Volvo's Integrated Starter Generator unit mounted between the 12L engine and the transmission. Features such as Start-stop are implemented to allow the engine to shut off during idling periods. This machinery can also provide electric power boost of 50 kW at low speed operations, and energy recovery during braking or swing-down phases [35].

Another more recent example of hybridization in wheel loaders is the Hitachi ZW220HYB-5, medium class wheel loader, utilizing a series hybrid for traction (2 axle mounted electric motors) and a parallel configuration for the working hydraulics. It also utilises a supercapacitor energy storage system, with reported fuel-use reductions of 26% to 31% (in-house testing) compared to its conventional ZW220 counterpart [36].

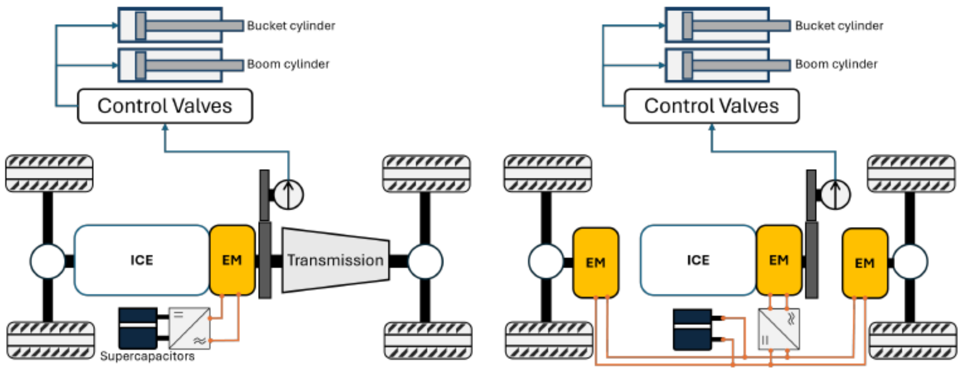

Figure 2.14: Hybrid configurations for wheel loaders: Volvo L220F (Left) and Hitachi ZW220HYB-5 (Right) [7], [9]

Some other examples include the Komatsu WE-series hybrid wheel loaders (WE1850, WE2350, WE475). They use a fully regenerative series-hybrid electric drive with a diesel engine that runs a generator, electric traction, and regenerative braking. Komatsu claims up to 45% lower fuel consumption for these mining loaders compared to similar conventional drive machines [37]. John Deere 644K Hybrid and 944K Hybrid are also available, utilizing a series hybrid powertrain and electric traction.

Skid-Steer Loaders:

One of the NRMM Construction domain that is has received little attention is the Skid-Steer Loader. While some concepts of hybrid or fully electric like the IMER IHIMER 3S Hybrid[38], Yuchai S35 Max (electric), or Bobcat S7X and T7X (electric) [39] do exist, they are not yet available. However, fully electric machines suffer from issues with reduced working hours, and the inability to have a more than one shifts. Especially in Skid-Steer Loaders, development of hybrid powertrain is crucial since no available hybrid powertrain machines are commercially available.

Other NRMM include:

HAMM HD+ 90i PH tandem roller, Caterpillar D6 XE dozer, Tadano AC 4.070HL-1 and AC 5.120H-1 all-terrain cranes, Liebherr MK mobile construction cranes (hybrid operation: grid/electric on site with diesel backup).

These machines exploit the electrification or hybridization primarily to improve the overall energy efficiency of the powertrains. In the case of excavators for example, the electric machines are often used to recover energy during cabin swing, boom-down, or deceleration, support the engine during high power demand and employ EMS strategies such as Load Point Shift, reducing the consumption, while maintaining the required performance.

2.2 Case Study - Skid-Steer Loader

The reference machine adopted in this work is a skid-steer loader, one of the most widely used machines in the construction sector. Its power rating makes it a suitable candidate for hybridisation, since electrified components can be integrated without extensive modifications to the existing layout and within a range where the adoption of SCR after-treatment is not strictly required.

In particular, the tracked variant (Compact Tracked Loader), shown in figure 2.17a, is taken as the reference configuration.

(a) Tracked Loader performing bucket filling part of the duty cycle. Source: Caterpillar Inc.

(b) Tracked Loader performing bucket emptying part of the duty cycle. Source: Doosan Bobcat

Figure 2.15: Examples of two Tracked Loaders performing the duty cycle.

Skid-Steer Loaders are particularly interesting, since they offer hydraulic working components, and rapidly changing and short working cycle, yet they remain comparably under-represented in the domain of Hybrid NR-MMs. They are compact and manoeuvrable and extensively used in construction and material handling. Skid-Steer Loaders can also be used for a number of applications, and are highly customizable and agile, with a large number of attachments that can be fitted to them, depending on the work. Some of the available attachments are shown in figure 2.16.

(a) Tracked Loader with a grapple bucket attachment

(b) Tracked Loader with a chain trencher attachment

Figure 2.16: Examples of two Tracked Loaders with attachments [10].

Skid-Steer Loaders and Tracked Loaders utilize a hydrostatic transmission, similar to a crawler excavator. Much like other tracked or HST powered vehicles, in a Skid-Steer Loader, these machinery employ a throttle notch, at a fixed working RPM. Speed/torque at the tracks and at the cylinders is then modulated by variable pump/motor displacement and valve positions, not by continuously varying engine speed. Keeping RPM high and steady keeps pump flow available and reduces response lag when the operator suddenly asks for traction or implement movement.

The duty cycle is characterised by a short (30 - 100 seconds) highly transient manoeuvres. Therefore, the powertrain needs to be comfortable to idle for periods of time, as well as with high bursts of torque and vehicle speed variation. Since the engine needs to drive both hydraulic components, the duty cycle can sometimes overload the engine, when multiple actions are done at the same time.

More traditionally they are wheeled as shown in figure 2.17b, with a single hydraulic motor for each side driving both wheels of their side via a chain. They can also be tracked (a single hydraulic motor on each side will drive each side of the tracks, left and right) as shown in figure 2.17a. While technically called a Compact Tracked Loader, or a Tracked Loader here thereafter, it will only be referred to as a Skid-Steer Loader or SSL.

The tracked version has a few advantages and disadvantages compared to the more traditional wheeled one. It offers higher traction on loose or soft ground because the contact patch is larger. The increased ground contact also improves stability on slopes and reduces bouncing on rough surfaces. As a result, tracked SSLs typically provide higher push capability than similarly sized wheeled machines.

(a) T770 Tracked Loader (b) S770 Skid-Steer Loader

Figure 2.17: Examples of two identically sized Skid-Steer Loaders from Bobcat: Tracked and Wheeled. Source: Doosan Bobcat

The wheeled version has its benefits. It is cheaper to maintain, since tracks and undercarriage needs much more care and cost more to replace then wheels. The wheeled version can also travel faster in smoother surface such as asphalt or concrete.

Due to their characteristic work which involves lifting piles of materials, transporting them to the drop zone, and dropping them, conventional Skid-Steer Loaders need two separate circuits to conduct their work:

- **Hydrostatic Transmission**: This is a closed loop hydraulic circuit that drives the vehicle. Its sole function is vehicle propulsion.
- **Working Hydraulic Circuit**: This is an open loop hydraulic circuit that powers the loader's arm cylinders, bucket cylinders, and other auxiliary attachments.

Vehicle and Powertrain Specifications

The Skid-Steer Loaders tend to exhibit a widely scattered engine speed–load distribution, with a substantial period of the cycle spent at high engine speeds and partial load. By contrast, bigger machines, like wheeled loaders operate the engine more frequently within a narrower band of medium–high load and moderate speed, which facilitates operation closer to the optimal efficiency region and enhances the potential benefits of advanced engine or hybrid control strategies.

When choosing downsized engine replacements for hybridization, one of the considerations of this thesis is in downsizing to engines below the bracket of 56 kW and up. Since the engines in the 56 kW-and-up power output bracket need to employ an array of sensors and NO_x after-treatment systems according to table 1.1, that can increase the costs and complexity of the machinery, while sometimes slightly impeding the machine's performance. The largest one is the 55.7 kW (rounded up to 56 kW) 2.5L engine, shown in table 2.3.

	Conventional	**Hybrid**
Lifting capacity [kg]	1000–1650	1000–1650
Approximate weight [kg]	5700	5700
CI engine displacement [cm^3]	3400	1900–2500
Maximum torque [N-m]	470	450
Maximum power [kW]	80 (w/ SCR)	55.7 (+20 el.)
Electric motor nominal/ peak power [kW]	–	16 / 20
Battery energy capacity [kWh]	–	8

Table 2.3: Conventional and hybrid specifications of the SSL under analysis

2.2.1 Mission Profile Identification

Lack of a standardized testing cycle aside, a few approximations have been made, with a working cycle, commonly called a Y-Cycle. The name is taken from the fact that it looks like the letter Y as shown in figure 2.18.

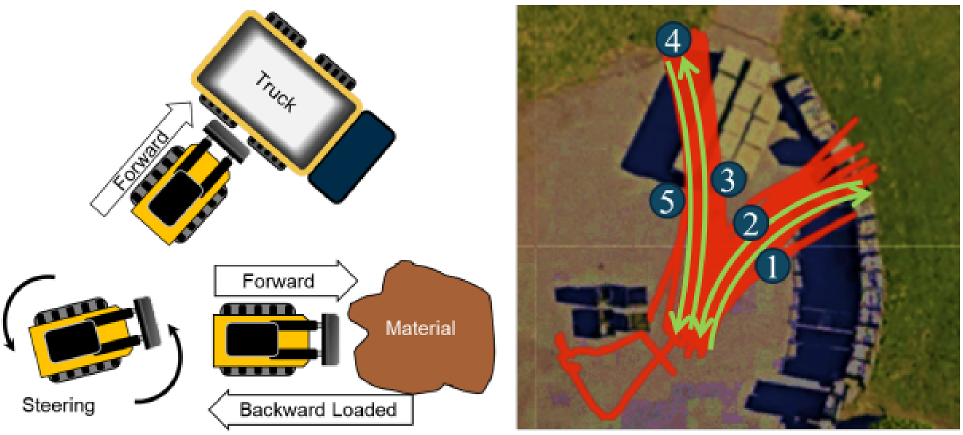

Figure 2.18: Y - Cycle of a Wheel Loader [7]

This mission profile is based on the typical load cycle of loaders, and other machinery doing typical loading/unloading work in highly repeatable and fixed intervals. While not standard, and thus cannot be used for regulatory purposes, this estimated working cycle can yield good results. For example, Bertini et al. use a version of the Y cycle to show the potential in hybridization of a Skid-Steer Loader in their paper about exploring electrification of powertrains in NRMMs [11].

In such loading and unloading cases, the working cycle or mission profile is split into the following phases:

1. **Advancing and Loading:** the machine moves forward from its initial position to and pushes into a pile of material.
2. **Reverse Travel:** The machine, now with the bucket loaded, backs away from the pile.
3. **Forward Movement:** The machine will drive towards the discharge area.
4. **Unloading:** The arms are raised and bucket is tilted to dump the material in the dumping position.
5. **Return and Arm lowering:** The machine will reverse back and the arm is lowered to the initial position.

A representation of the total Mission profile implemented in this model, in terms of physical quantities, is shown in figure 2.19.

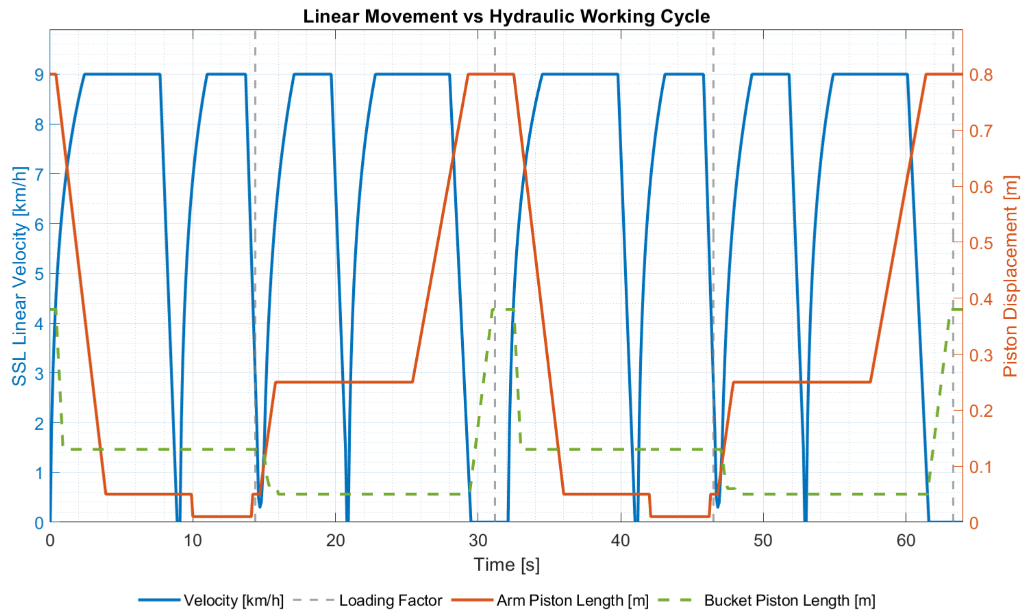

Figure 2.19: Y - Cycle Linear Velocity and Arm Piston's position relation

Chapter 3

Vehicle Modelling

Vehicle modelling and simulation of operations is particularly useful when experimental development and data acquisition from real models is costly, or new architectures are still in early development stages. And this is exactly the case for hybrid Skid-Steer Loaders; no publicly available data or commercially available models exists.

The simulation approach also allows us to check where there can be better optimization of certain elements which are drawing too much power, or causing issues with the system, as each element can be checked individually. This allows us to know where to introduce an electrifying element (for example, electric motor), and what sort of power is needed to supply this added element. This is very important for component sizing (electric motors, batteries, etc) and overall architecture of the vehicle.

This approach is also useful when considering subsystems that can allow for energy recouperation, and so on. In a real machine, a number of expensive gauges and other measuring equipment are needed. While simulation approach is one of the first steps in R&D it allows for huge potential savings in costs of development before a viability study has been made.

This is why a model based approach is required in order to investigate the hybridization potential of these machinery, and eventually produce a working prototype, and then hopefully a commercially available machine.

Simulation however is not free of limitations. Side effects related to human operator behaviour, soil conditions, temperature and other boundary conditions can only be simulated so accurately, and are generally taken as a constant, approximate parameters, or an educated guess based on experimental data. While they are good enough to conduct a viability study, such as this one, more research and development has to take place in order for a real world accurate model. Another issue with simulation is that some simplification has to be included for many of the components, as its not only time consuming and computationally demanding, but also generally provides minimal additional insight into the system. Many models and components are simulated using a lumped-parameter methods, where secondary components such as valves, piping, etc., are considered as functional blocks with a fixed efficiency and described by flow-pressure relationships. Such models can be implemented in the available 1D simulation software, such as GT - SUITE, AMESim or others, and are widely utilized in the industry for quick and cost-less results.

In this chapter, the main elements of the simulation, machinery subsystem, as well as their individual system results will be looked at. The parameters used are taken from manufacturer datasets when available, or referenced from other papers.

3.1 Conventional Skid-Steer Loader Layout

Typically in a Skid-Steer Loader, the machine has two separate hydraulic circuits. One is the Closed Hydrostatic Transmission, used to drive the hydraulic motors, and drive the machine, where as the other is the Working Hydraulics Circuit, which feeds the linear actuators that raise the arms and allow the bucket tilting.

A simplified conventional architecture would look like shown in Figure 3.1.

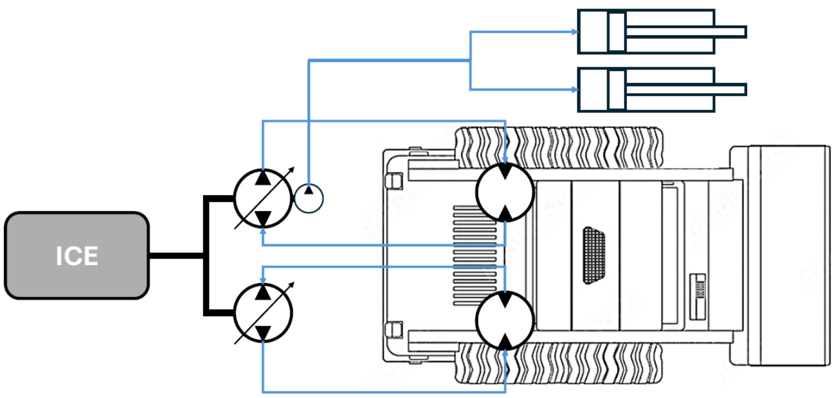

Figure 3.1: Simplified Schematic of Conventional SSL

Hydraulic systems are essential for the operations of NRMMs, and especially Skid-Steer Loaders. A detailed schematic with all the important components of both circuits is shown in fig 3.2:

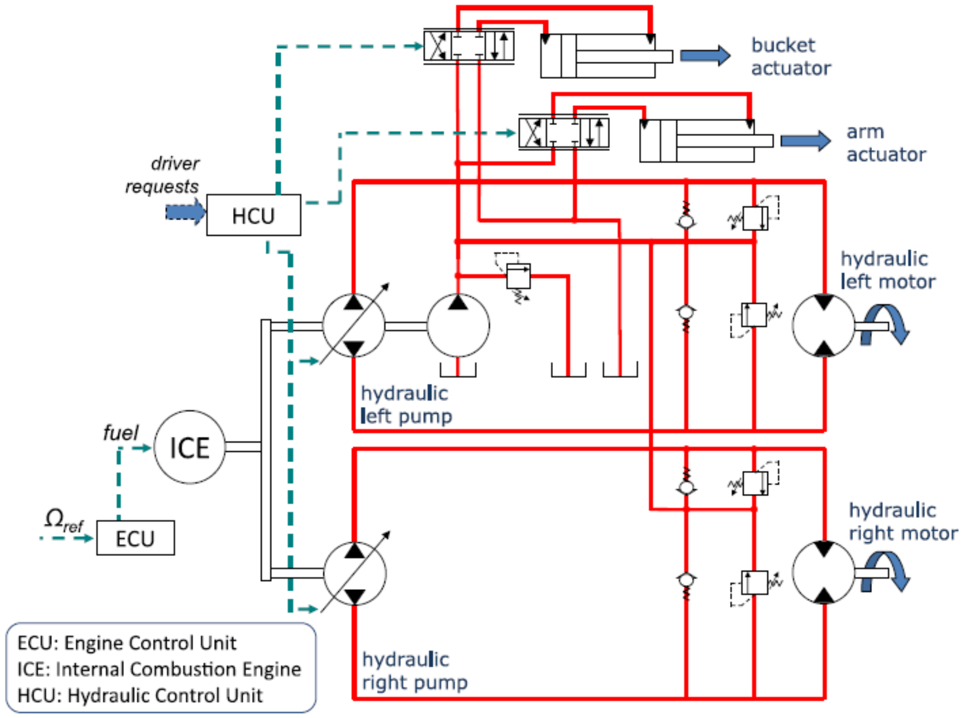

Figure 3.2: More detailed Schematic of SSL [11]

The engine will drive both the Hydrostatic Transmission (HST), used for traction and the Working Hydraulics Circuit, which is used for controlling other vehicle functions such as the arm and bucket assembly, or other hydraulic attachments that the Skid-Steer Loader can be equipped with.

3.2 Modelling Considerations in Skid-Steer Loaders

For creating accurate models able to predict the performance of a machine and evaluating new architectures, it is important to identify the four main aspects of modelling a Skid-Steer Loader:

- **Traction**: These include traction modelling (HST or otherwise).
- **Working Hydraulics**: These include an accurate kinematics modelling of the SSL, as well as hydraulic implementation. Special considerations need to be taken for the arm and bucket assembly of the SSL.
- **Y-Cycle**: A properly defined Y-Cycle, that's realistic and well split between the two hydraulic systems is very important in getting ac-

curate consumption results.

- **Conventional and Hybrid powertrain**: ICE, Electric components, and EMS need to be determined accurately to ensure proper operations, and longevity of the system.

3.3 Traction System

The traction system accounts for the predominant share of the Skid-Steer Loader's overall power and torque demand. In conventional models driven solely by an engine, this is done via the hydrostatic transmission. A system as accurate as possible is crucial for finding accurate consumption data, and needs to be based on kinematics - force equations of the SSL, as well as a realistic and well-designed Y-Cycle. Wheel loader simulations have shown that subsystem interactions between hydraulics and primary drivers strongly influence efficiency [18].

3.3.1 Hydrostatic Transmission Functional Description

The hydrostatic transmission is what translates the energy of the prime mover into movement of the vehicle. In these circuits, the outlet port of the hydraulic motor, is connected directly to the inlet port of the pump. The speed inversion in hydraulic motors in these systems is done via a gearbox (for conventional engine prime drivers), or a reversible hydrostatic unit. This allows the inversion of the direction of flow without changing the direction of the prime mover. The closed circuit is ideally a valveless circuit. The HST loops will have a high pressure and low pressure side, which are interchangeable during inversion of movement.

Each track is driven by a separate hydraulic motor, and each motor is connected to a theoretically closed loop to the pump. However, in real cases, the system is not fully closed due to some issues:

- **System leaks**: pumps and motors as hydraulic rotary devices have high leakages. The loop works under the idea that the system has a constant and fixed level of working fluid (oil).
- **Fluid Conditioning System**: these include oil filters and Heat exchangers. These are necessary in hydraulic circuits, but cannot be implemented in the main line.
- **LP line pressure definition**: Pressure in the low pressure line is undefined, and therefore throughout the circuit its undefined. This is because hydraulic motors and pumps generate torque and

force based on a pressure drop, and therefore it's important that the pressure throughout the circuit remains stable and fixed.

To address these issues, a flushing valve and a pressure relief valve is set in the system[1]. The flushing valve will connect the low pressure line with the fluid conditioning unit, as well as set a pressure for the low pressure line $p_{LP} = p_{RV1}$.

A Charge Pump (generally with a displacement of 10 to 15% of the main pump) as well as an array of pressure relief valves have to be implemented in each HST loop. A real HST loop will look something like so:

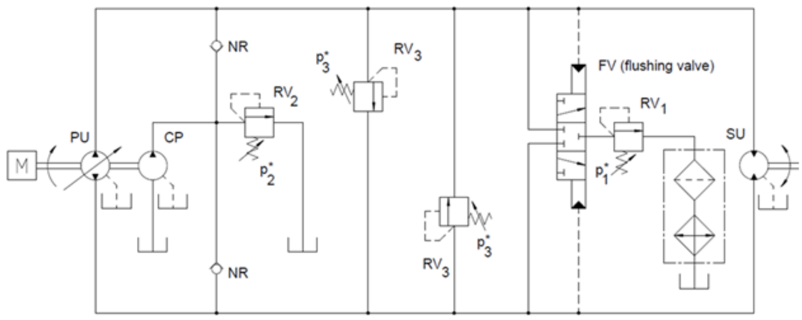

Figure 3.3: Real HST Loop Components [12]

3.3.2 Longitudinal and Steering Forces in SSLs

Longitudinal motion of NRMMs can be described using similar equations to those for on-road vehicles. To propel a vehicle, the hydraulic motor must generate sufficient force to overcome inertia, rolling resistance, aerodynamic drag, and road inclination. In addition to this, off-road machinery often requires the addition of specific terms. For a Skid-Steer Loader, these factors include steering resistance and bulldozing force when pushing into material piles.

In the case of tracked vehicles, rolling resistance can be included within the steering resistance term, and the total resistance force can be expressed as:

$$F_r = F_{in} + F_{aero} + F_{grade} + F_{bulldozing} + F_{steering} \tag{3.1}$$

$F_{inertia}$ is the inertial force, considering time-varying mass ($M_{SSL}(t)$) due to load variation. It is the largest element comprising the traction forces

[1]This is not implemented into this system, and the fluid conditioning parameters are set to constant.

and torque needed at the sprocket.

$$F_{inertia} = M_{SSL}(t)\frac{dv}{dt} \tag{3.2}$$

F_{aero} is the aerodynamic drag, which is typically negligible at SSL operating speeds.

$$F_{aero} = \tfrac{1}{2}\rho_{air} A_f C_d v^2 \tag{3.3}$$

F_{grade} is the grade resistance due to slope inclination θ. The slope was considered zero in this model.

$$F_{grade} = mg\sin(\theta) \tag{3.4}$$

$F_{bulldozing}$ is the force arising when the loader is pushing into granular materials like gravel, sand, or rocks. It is considered a peak resistive force at the end of the deceleration phase, due to the tractive force exerted by the tracks when filling the bucket. In this case was considered a total value of 20 kN.

Steering forces

Differential steering forces a core concept of vehicles utilizing a hydrostatic transmission, and more specifically tracked vehicles. Rather than steering through a front axle, vehicles like the Skid-Steer Loader yaw by generating differential speeds between the two speeds, often holding one track nearly stationary while driving the other. This enables extremely tight turning radius, almost in-place pivots. The steering resistance term is given by:

$$F_{steering} = F_{out} + F_{in} \tag{3.5}$$

Here, F_{out} and F_{in} represent the thrust needed on the outer and inner tracks, respectively.

The skid steering mechanism model from Kotiev [40] is then used to estimate the thrust forces of the tracked system. The outer track thrust force is given as:

$$F_{out} = \frac{M_{SSL}(t)\cdot g}{2}\left(f_s\left(1+\frac{2V^2 H_z}{B\cdot R_f\cdot g}\right)+\frac{\mu\cdot L}{2B}\left(1-\left(\frac{V}{V_{crit}}\right)^4\right)\right)+\frac{M_{SSL}(t)\cdot V^4\cdot L}{4\cdot R_f^3\cdot\mu\cdot g} \tag{3.6}$$

Similarly, the inner track thrust force is:

$$F_{in} = \frac{M_{SSL}(t) \cdot g}{2} \left(f_s \left(1 - \frac{2V^2 H_z}{B \cdot R_f \cdot g} \right) - \frac{\mu \cdot L}{2B} \left(1 - \left(\frac{V}{V_{crit}} \right)^4 \right) \right) + \frac{M_{SSL}(t) \cdot V^4 \cdot L}{4 \cdot R_f^3 \cdot \mu \cdot g} \tag{3.7}$$

Where:

- $M_{SSL}(t)$ = time-varying mass of the SSL
- f_s = resistance coefficient for straight-line motion on soil
- V = vehicle speed
- H_z = height of the centre of mass
- B = track width
- R_f = turning radius
- μ = lateral resistance coefficient
- L = track contact length

The critical velocity before drift is:

$$V_{crit} = \sqrt{\mu_{max} \cdot g \cdot R_f} \tag{3.8}$$

with the actual turning resistance coefficient expressed as:

$$\mu = \frac{\mu_{max}}{0.925 + 0.15 \frac{R_t}{B}} \tag{3.9}$$

with $R_t = \frac{R_f \cdot B}{k \cdot L}$ being the theoretical radius, and k a proportional soil coefficient.

Finally, to evaluate track kinematics, the track speeds are expressed as:

$$V_2 = V_{out} = V \left(1 + \frac{B}{2R_f} \right) \qquad V_1 = V_{in} = V \left(1 - \frac{B}{2R_f} \right) \tag{3.10}$$

The sprocket angular speeds can then be calculated as:

$$n_2 = \frac{V_2}{r}, \qquad n_1 = \frac{V_1}{r} \tag{3.11}$$

and the input/output torques as:

$$T_{in} = F_{in} \cdot r, \qquad T_{out} = F_{out} \cdot r \tag{3.12}$$

where r is the sprocket radius.

The main machine characteristic variables are graphically shown in figure 3.4.

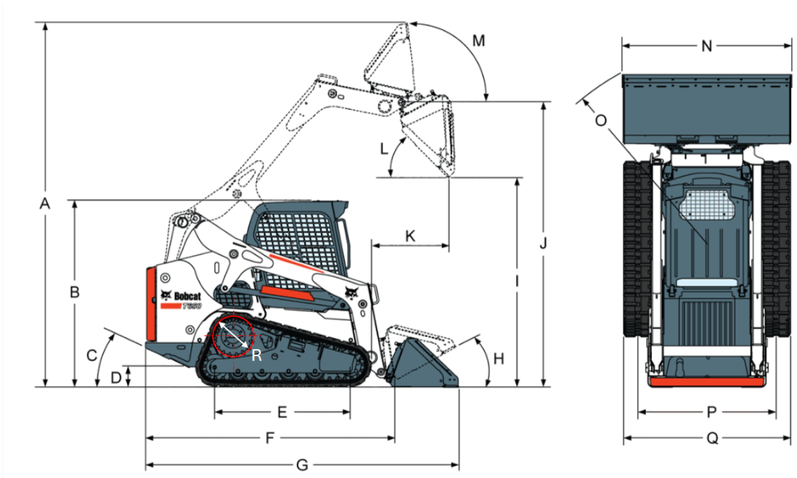

Figure 3.4: Graphical representation of the dimensions of the SSL [13]

These dimensions in figure 3.4 correspond to the following dimensions used in the equations above:

- Track width B = Dimension P
- Height of centre of mass H_z - not directly shown, estimated relative to dimension A/B
- Track contact length L = Dimension E
- Sprocket radius r = Dimension R

The dimensions have to be carefully selected, based on existing machines, as small changes in some of them can lead to an unstable model and unreliable data.

Vehicle capacities such as maximum speed, turning radius, bulldozing force, and so on, need to also be estimated based on operations of real machines. This is to ensure that no unrealistic forces and torque requirements are put in the model.

3.3.3 Y - Cycle for Traction system

Based on the Y - cycle described in Chapter 2, a graphical representation of the inputs of the cycle is shown in figure 3.5. These are desired behaviour in terms of speed, turning and bulldozing during the Y-Cycle:

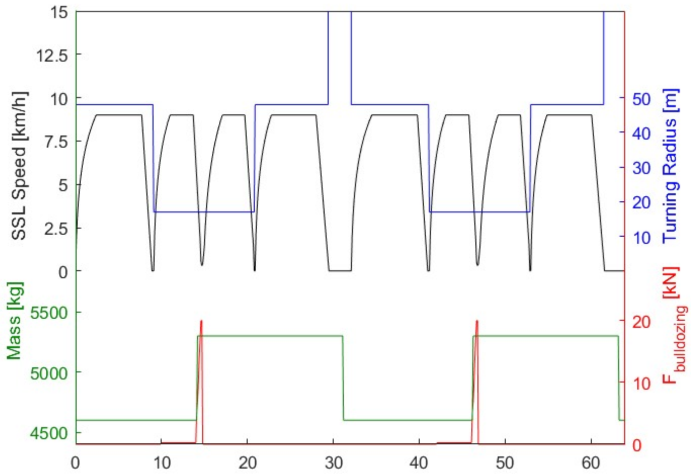

Figure 3.5: HST duties during the Y-Cycle

3.3.4 Implementation Strategy for the Hydrostatic Transmission

The control logic is a combination of the two main types of loops: feedback and feedforward.

The feedforward loop will be used for the input Y-Cycle of the vehicle, as to simulate how an operator would work with the machine. This means that the torque request at the sprocket is calculated using the actual velocity of the vehicle, it's mass, and turning rate. The hydraulic motor will need a flowrate from the pump to overcome this torque, and the engine will provide the torque to drive the pump supplying this flowrate to the hydraulic motor.

The feedback loop will be used in the internal control of the system, as a feedback control loop to continuously adjust the necessary parameters (such as pump displacement) so the system behaves as the user requires. The feedback works constantly through a PID which adjusts the displacement of the pump, and based on the speed give the hydraulic motor the flowrate to produce the torque necessary to overcome the torque applied at the sprocket.

Figure 3.6 illustrates a similar control logic implemented by Han et al [14]. In this paper, the desired vehicle motion, provided as a target linear velocity and yaw rate is converted into reference commands for the left and right motor speeds. These references provide the feedforward action. At the same time, the measured motor speeds are fed back and compared with the target. This yields an error ϵ, which is processed by a low-level controller (PI/PID) to adjust. The final displacement command to the variable pump combine the forward component that enforces the required steering behaviour, with the backwards, corrective, component that guarantees the parameters are adjusted accordingly.

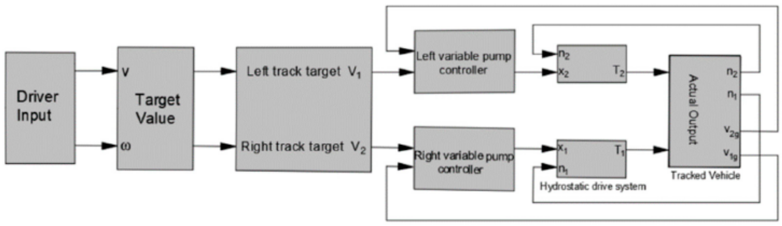

Figure 3.6: Feedback Loop according to Han et al. [14]

In this model, the machine will firstly need to move forwards with a linear velocity. V – the velocity from the driver – will be turned into required sprocket angular velocity by:

$$\omega = \frac{V}{r} \tag{3.13}$$

Where:

- V is the requested velocity by the driver
- ω is the angular velocity that the sprockets need
- r is the radius of the sprocket

If the vehicle is turning, refer to equation (3.10). Next the model needs to calculate how much torque is needed at each track to move the vehicle. Considerations:

- Vehicle mass will vary during the cycle (loading/unloading). This can be part of the cycle uploaded.
- Terrain introduces resistive forces (off-road conditions).
- Turning radius input needs to properly adjust each side via PID controllers.

The thrust forces are calculated with equations (3.7) and (3.6). Torque request is based on equation (3.12).

As feedback, we use the actual angular speeds of the sprockets:

- If $\omega_{motor} > \omega_{target}$: increase pump displacement
- If $\omega_{motor} < \omega_{target}$: reduce pump displacement

This feedback goes to the PID Controller, constantly adjusting the displacement of the Main pump. This process is repeated for both tracks, each of which will have an independent loop.

The control or implementation logic used in this model is shown in figure 3.7.

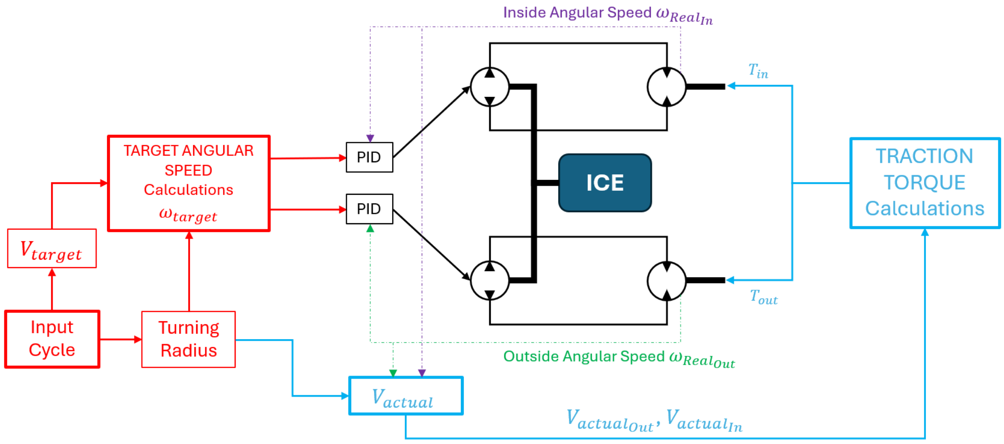

Figure 3.7: Control Logic for the Hydrostatic Transmission

System Behaviour in Different Driving Conditions

To showcase how the HST will behave during two main manoeuvres, which are driving straight and turning, the individual behaviour of the components of the HST is shown below.

Driving Straight

- Equal pump displacement means equal flow which means equal speed.
- Motors rotate at the same speed.
- Relief valves inactive under ideal conditions.

Turning

- Increased displacement on outside track's pump which means higher flow and higher pressure difference in it's sides hydraulic motor. Since the pump is a variable displacement pump, the displacement control is done by the feedback loop PID. This ensures that the pump will change the displacement accordingly to account for error correction, and to account for the speed of the primary driver.
- Outside pump will have a lower or equal displacement, depending on turn radius. This will create a lower pressure drop in the hy-

draulic motor, and thus a differential force is created between the two tracks. The difference in forces will create a turning torque.

3.3.5 GT - SUITE Model

Finally, the entire system is built in GT - SUITE, shown in figure 3.8.

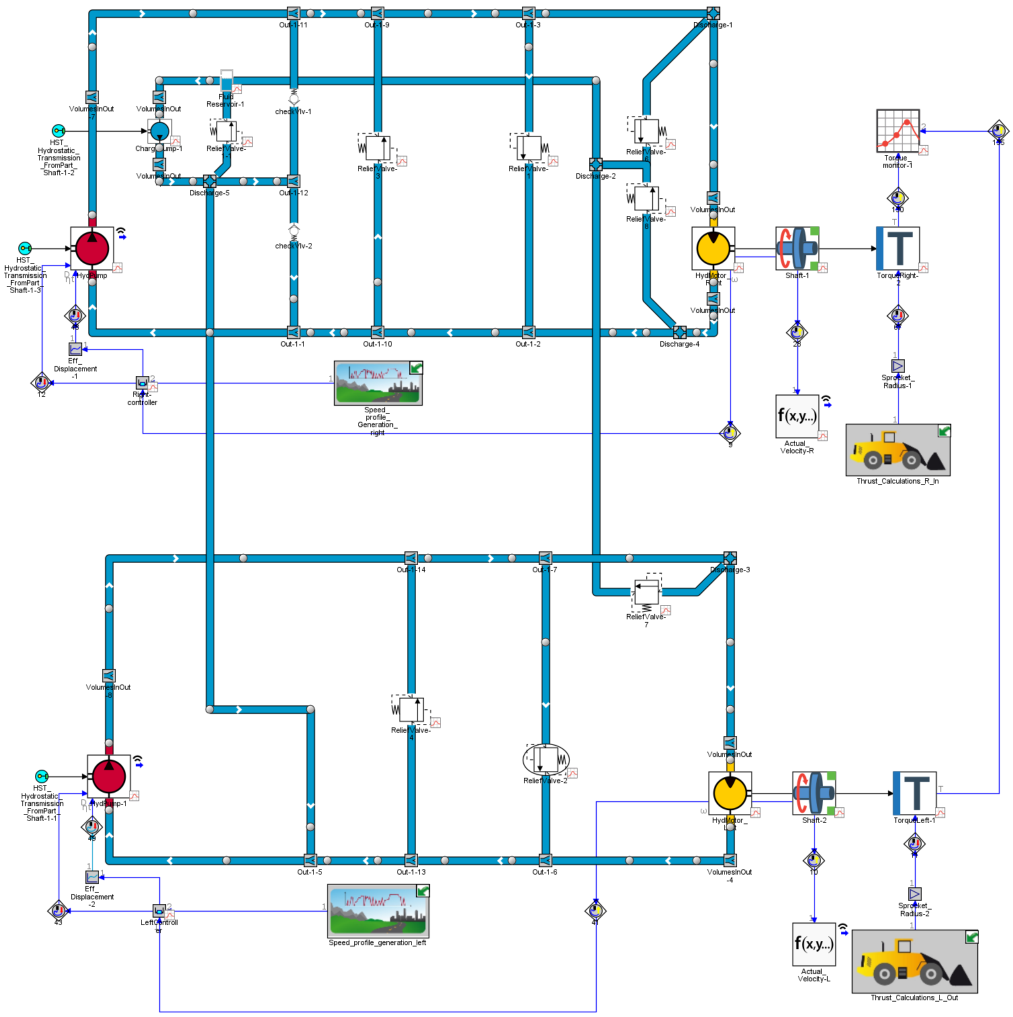

Figure 3.8: Hydrostatic Transmission built in GT - SUITE

Both control loops are inside the model. It also contains some calculation blocks, that according to the previous section describing the general traction and Kotiev model [40] will turn the requested linear speed and yaw rate into two hydraulic motor rotational speeds, and then into the torque applied at each track and sprocket.

3.3.6 Results - Traction

The system is initially run solely for quantifying the torque applied at each sprocket, and what torque request will each pump have in order to overcome this resistance torque.

Torque request at the sprockets

The system will treat the torque requests of each sprocket, calculated in the section *Longitudinal and Steering Forces in SSLs*, as a resistive torque applied to the sprocket, connected to the hydraulic motor. For this, several GT - SUITE calculation blocks such as MathFunction are utilized, so that the hydraulic motor is getting the torque request in each timestep, and then transmitting that information to the pump, which then, in turn will draw the torque from the Primary driver.

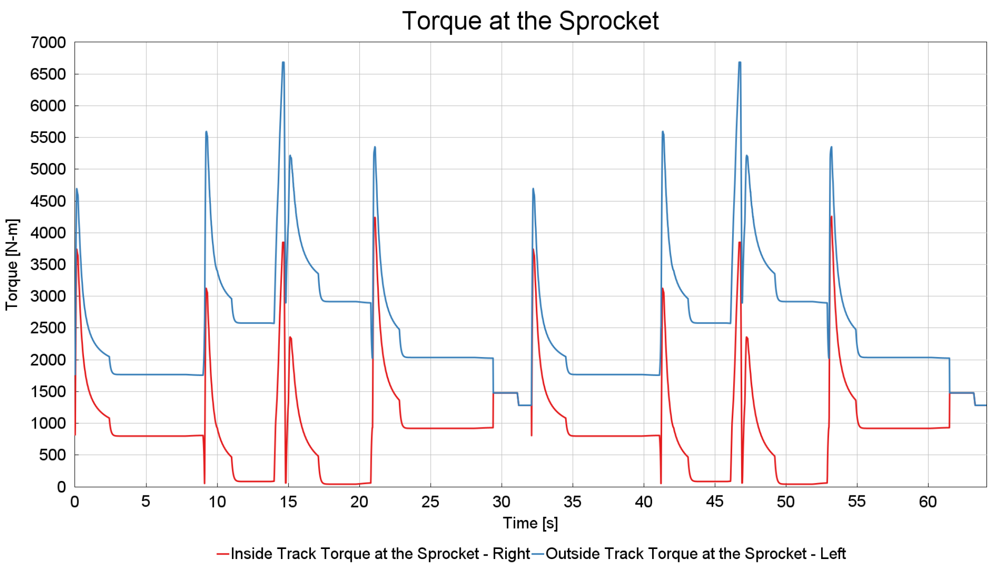

Figure 3.9: Torque at the sprockets

The peaks represent acceleration phases, where as the difference in torque such at time 12 to 14 seconds or 17 to 21 seconds represent turning phases. The bulldozing phases are shown by the peaks at 14 and 47 seconds.

Total Torque and Power request of the HST

The majority of the torque request comes from the outside track pump. In the system the left track is set as the outside track, and the right track as the inside track. The tracks remain outside and inside as specified as to simplify the system, and because this is an energetic study of the entire machine, this makes no difference to the results.

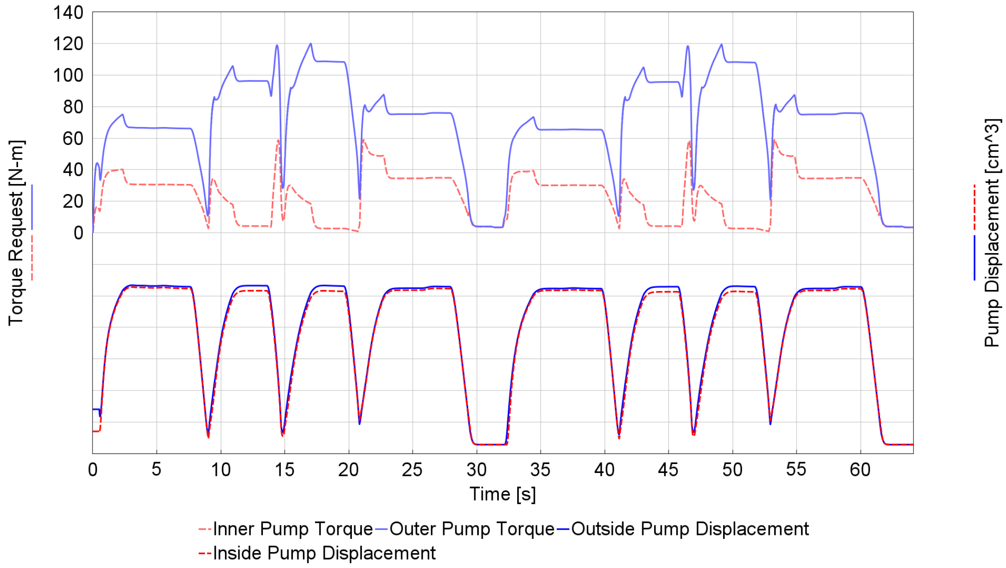

Figure 3.10: Torque Request by the pumps and normalized pump displacements

After the simulation is run, the two pumps plus the smaller charge pump will in total have the following torque requirements (along with the decomposition of each individual element's torque requirement), shown in figure 3.11:

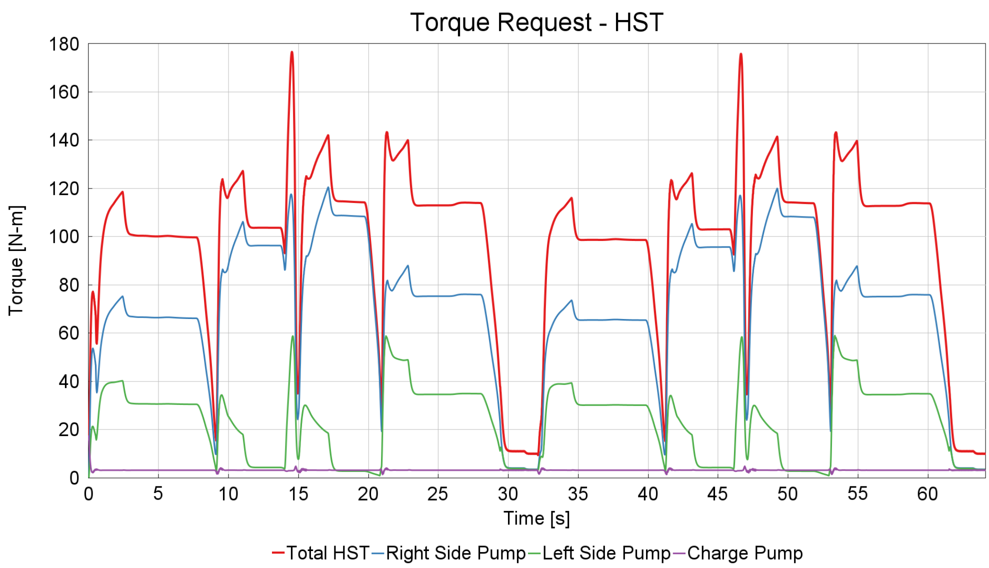

Figure 3.11: Total Torque Requirement for the Hydrostatic Transmission

3.4 Working Hydraulic Circuit

In parallel to the HST, the Working Hydraulic Circuit is an open circuit used to actuate the lift arms, tilt the bucket, and do other auxiliary functions of the vehicle.

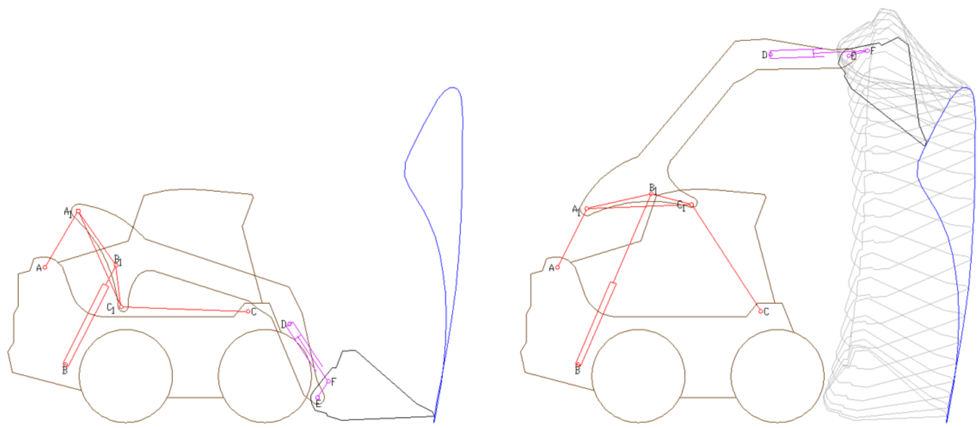

Figure 3.12: First and final frames of the filling and emptying the bucket cycle

The Y-Cycle duties of the working hydraulics system are related to lifting the arms, and tilting the bucket. For the Y-Cycle used in this model, the steps in the cycle are divided as follows:

1. The machine rams into the pile of material
2. The machine lifts the arm slightly and changes the angle of the bucket to hold the material better.
3. The machine transports the pile to the drop zone in this arm and bucket assembly position.
4. The machine then raises its arm to prepare for the drop.
5. The machine tilts the bucket (bucket piston at full extension) and drops the material.
6. The arm and bucket are returned to the original positions to redo the work again.

While the Skid-Steer Loaders kinematics of the arm and bucket are generally simpler than larger machines, like excavators for example, they are less documented in the literature. By combining data from similar articulated wheel loaders with observations of real SSL operations and Y-cycles, a reasonable kinematic approximation can be obtained, as shown in figure 3.13.

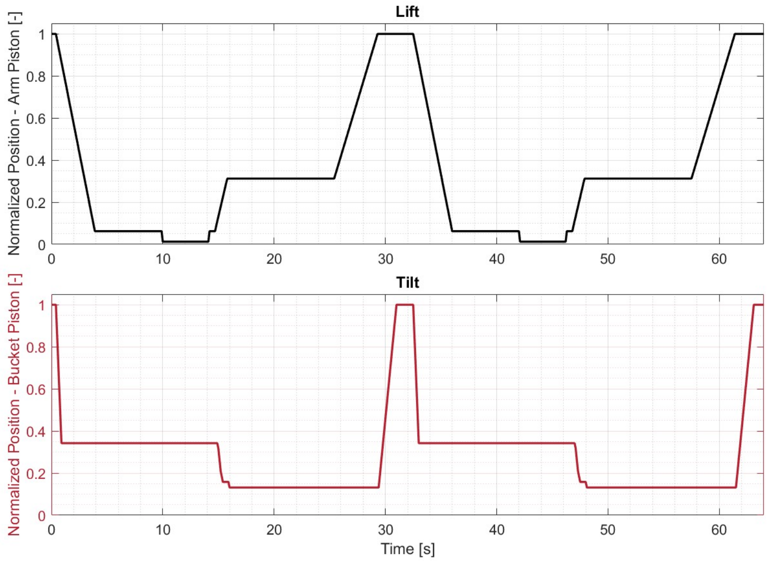

Figure 3.13: Normalized Piston Position vs Time

The Working Hydraulics Circuit is characterized by high, variable demand flows depending on the load cycle. Losses mainly occur through throttling in control valves, as well as heat dissipation in the valves.

System Functional Description

The system implemented here is a conventional Open-Centre system. While an Open-Centre system is less efficient than more complex systems (like Load Sensing), they allow the manufacturer to keep the costs down, since Open-Centre mostly requires the 2 DCV 6/3, as well as a fixed displacement (generally gear) pump. They also allow for easier maintenance, and smaller required area for fitting the system (which is crucial in SSLs, which are compact machines). While they have lower efficiency, this is acceptable for compact machines.

Open-Centre system allows for three different configuration of two directional valves:

- **Series**: Both valves act together and its impossible to move only one user; these provide synchronous motion.
- **Parallel**: Only one of the valves can act at a certain moment, with the valve upstream having priority.
- **Tandem**: Allows for different types of movement.

The two continuous directional control valves (DCV 6/3) utilized in this model are in the parallel configuration, meaning that they can act independently or together.

An example of a similar system is shown in figure 3.14.

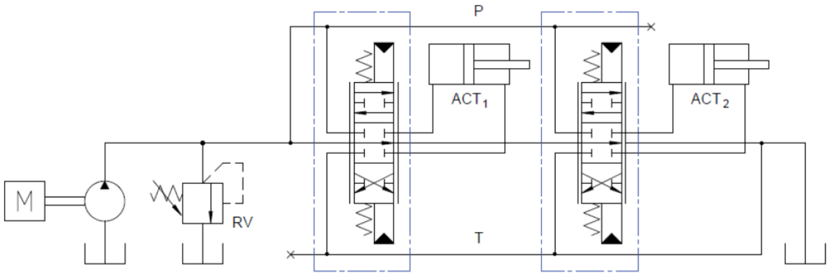

Figure 3.14: Open Centre System simplified [12]

The Continuous Directional Control valve has two main characteristics:

1. The central line is the Carry Over Line: In non-load conditions, the pump will be connected to the tank. This allows the pump to operate at *almost* atmospheric pressure of the tank, and saves energy from the prime driver.
2. The valves have continuous control. So, while it has 3 main positions like a regular 6 port, 3 position flow control valve, it will allow for the passage into each linear actuator to be restricted with the commands of the joystick by the operator.

This is shown in the figure 3.15.

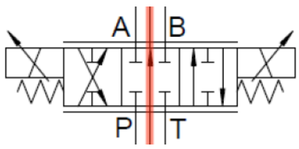

Figure 3.15: Continuous Directional Control Valve 6/3 valve highlighting the Carry-Over Line [12]

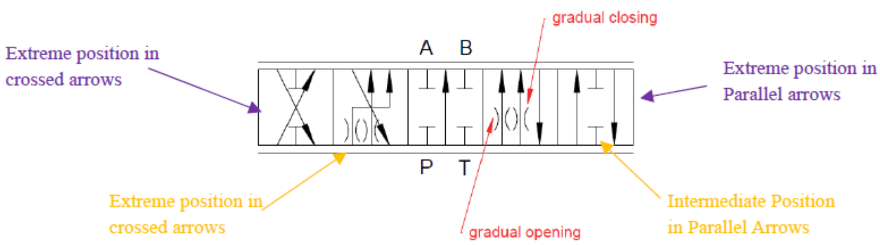

Figure 3.16: The positions of the Continuous Directional Control Valve in a Open Centre System [12]

In these types of Directional control valves these are the three main positions:

- Central Position: $p_{Pump} = 0$ bar; this is because the carry over line goes directly to the tank, and thus the delivery pressure is ideally zero. The hydraulic motor is blocked.
- Two Intermediate Position: P - A or P - B; the pump will delivery the flow to one of the inlets of the linear actuators depending on the command being requested (LA going inwards or outwards).

This is also shown in figure 3.16. These allow for precise control, and lower consumption and higher efficiency, since the pump doesn't have to work at the pressure level of the relief valve when at rest. For the system in general,

3.4.1 Implementation Strategy for the Working Hydraulics Circuit

To replicate the operator input, the model uses a PID controller to adjust the position of the arm and the tilt of the bucket. The PID will take the Input Cycle in the form of an outside cycle, which is provided as a target. The behaviour of the lift and tilt systems is shown in figure 3.17

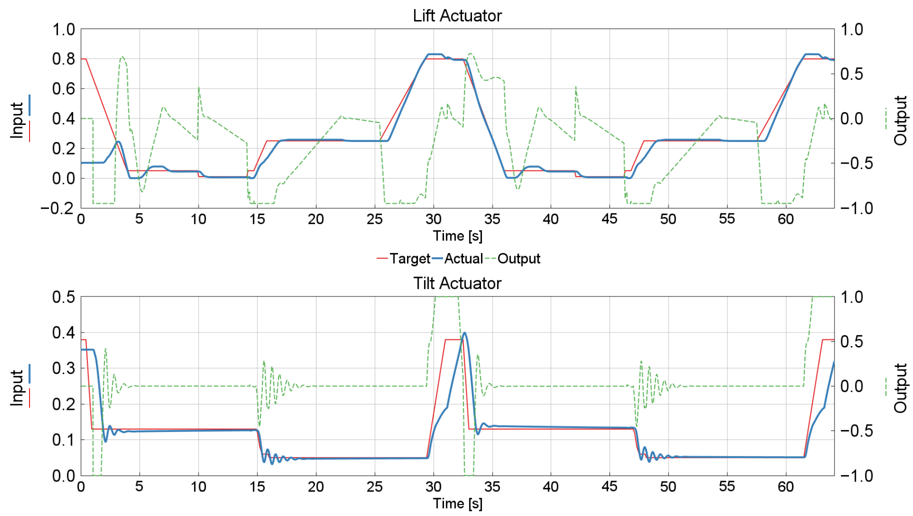

Figure 3.17: Lifting and Tilting of the Working Hydraulic

The use of a PID-based system allows repeatable actions and avoids any variances. This allows for more accurate evaluations of the hydraulic system configurations and more effective implementation of control strategies.

3.4.2 GT - SUITE Model

The fully built circuit implemented in GT - SUITE is shown in figure 3.18. Calculation blocks in the system calculate the forces in the linear actuator, while taking into consideration the kinematic relationships between the arm and bucket actuators, as well variables such as weight. The weight is considered to be constant during the lifting phase, and then gradual drop during the dumping phase. These movements would correspond to phases 1 through 4 and 5 - 6 in the description of the duties of the Working Hydraulics Circuit during the Y-Cycle.

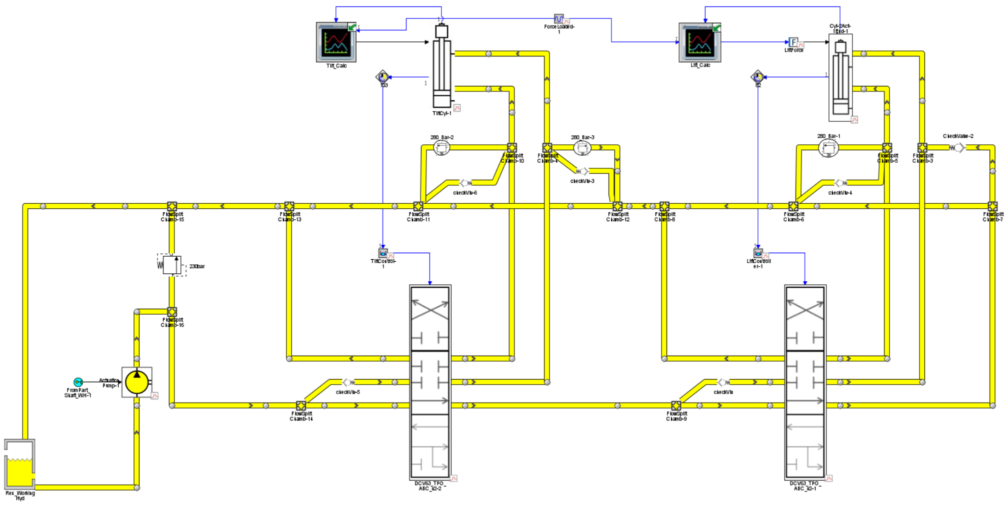

Figure 3.18: Open Centre System built in GT - SUITE

3.5 Primary Driver

Another key aspect for obtaining accurate results is the correct set-up and simulation of the powertrain. The imposed duty cycle, applied as an input to the hydraulic circuits, translates into a torque demand that must be supplied by the primary driver. The system is first operated with a conventional CI engine.

3.5.1 Internal Combustion Engine

In GT - SUITE the engine in this model is implemented as a quasi-static, map-based model. For each simulation time step the mechanical power is obtained from the speed–torque map, while the instantaneous fuel flow is computed from the BSFC map using the current operating point on the engine map. The same modelling approach is used for the downsized engines introduced in the hybrid configurations, with different torque and

BSFC maps corresponding to their rated power levels. The engine used in the conventional model is a four-cylinder 3400 cm^3 CI engine, producing $P_{\mathrm{ICE}} = 80$ kW and $T_{\mathrm{ICE,MAX}} = 490$ N-m (Fig. 3.19).

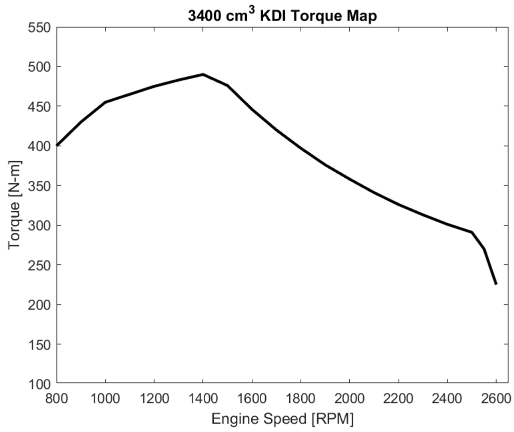

Figure 3.19: 3.4 L KDI Engine Mechanical Characteristic Graph

Generally, in machines that utilize the Hydrostatic transmission, such as Skid-Steer Loaders or Crawler excavators, the engine is set to a fixed velocity, using a notch throttle. A higher engine speed will make the machine more responsive, but also consume more fuel. The engine control is set to torque control, while maintaining a speed around 2300 RPM. The engine speed profile throughout the cycle can be seen in figure 3.20.

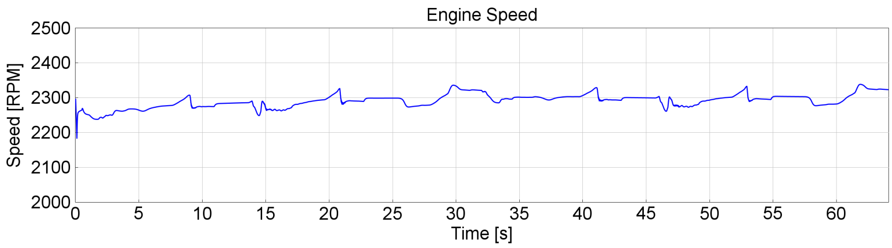

Figure 3.20: Engine speed throughout the cycle

The engine will follow the torque request of the individual sub-systems of the Skid-Steer Loader, like they were described in the sections before.

Torque Requirement

In the conventional system, both the hydraulic circuits are connected directly to the engine via rigid connections. The total torque and speed provided by the engine must satisfy the requests of each component.

As shown in the theoretical part, it can be seen now that the majority of the torque requirement is from the hydrostatic transmission. All combined, the engine will have the torque request as shown in figure 3.21.

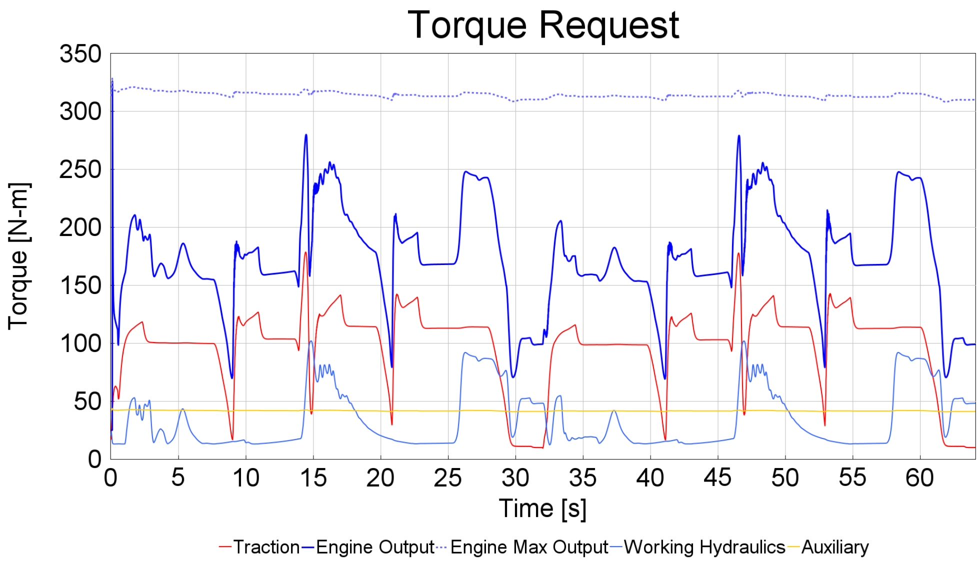

Figure 3.21: Engine Torque output and Torque Requirement of each subsystem

The engine torque produced peaks at about 275 N-m, with an average of 173 N-m, making it a feasible value for the downsized engine to keep constantly, and only use the electric motor in certain sections. These values include all the subsystems of the Skid Steer as well as losses and auxiliary loads.

Power Requirement

Similarly, for the power requirement, the values are shown on figure 3.22. In this case, the values of auxiliary loads such as the hydraulic cooling fan ($P_{absorbed} = 5$ kW), as well as idle losses of the HST and Open Centre Circuits. These all add up to about 10 kW.

In acceleration sections (like for example, t = 14.9 sec) the power requirement is around 40 kW just for the HST, and in total, with the Working Hydraulics circuit, around 54 kW. With losses included, these power peaks goes to about 65 kW, which is above the SCR implementation power limit (shown in table 1.1). In total, the average power demand at the crankshaft is about 41.6 kW. These values are in-line with what most manufacturers rate similar-sized machines' engines for [41] [42] [43] [44].

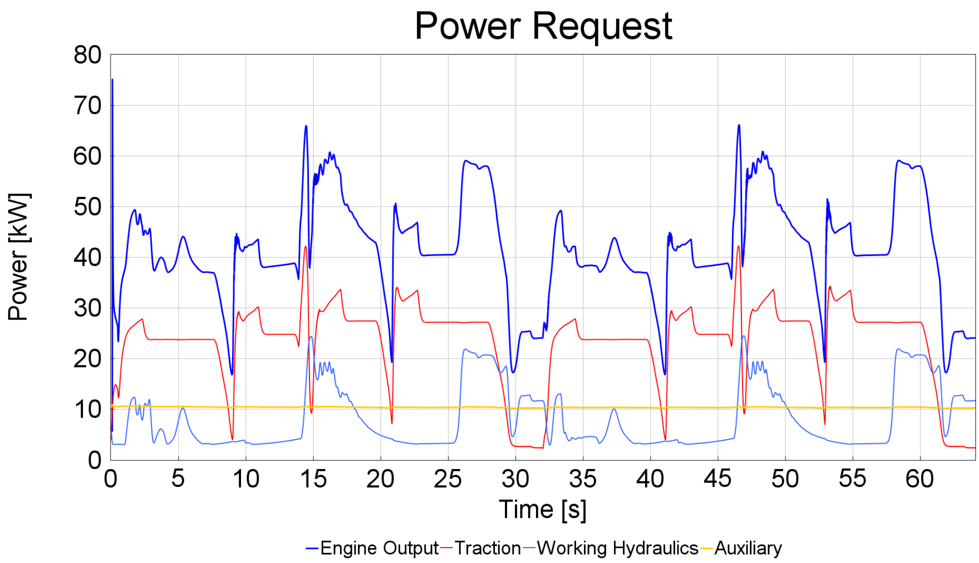

Figure 3.22: Engine Power output and Power Requirement of each subsystem

From these simulations, using a conventional much more powerful powertrain, its clear that the average torque and power request is well within the means of the downsized engines.

3.5.2 Hybrid Powertrain Modelling

Another element to be considered when considering hybridization of a vehicle is the Hybrid Architecture. In this model, different architectures and layouts that will be implemented. These architectures will be looked more in depth in Chapter 4.

Component sizing is another important element of hybridization. Depending on the architecture and engine displacement, components will have different characteristics.

Control strategy and EMS is also crucial in Hybrid powertrain, especially the limits and thresholds set in the Rule Based Algorithm to be implemented in this model. Again, this is dependent on architecture, as between different architectures, different power will be required from each component, based on the power-split that will be implemented. Appropriate component sizing ensures that the EMS can allocate the required power among the different units and that all components can operate together to meet the cycle demands.

Downsized Internal Combustion Engine

Downsizing an 80 kW engine to one with significantly lower power is not practical, because the resulting deficit would have to be compensated by a much larger battery pack and a considerably more powerful electric motor. A consistent and well-balanced sizing of the downsized engines is essential both to build a realistic model and to obtain valid, usable simulation results.

These engines were chosen because their rated power is below the 56 kW threshold, so SCR is not required to meet current non-road emission standards. As explained in Chapter 2, avoiding SCR simplifies the after-treatment system, reduces cost and packaging constraints, and better reflects the target machine segment. Yet, they are powerful enough to provide the average torque and power requests of the load without electric assist, while at the same time having a good fuel consumption and low emissions. The engine maps are shown in figure 3.23.

Their main characteristics are summarized in table 3.1.

V_d	$i_{Cylinder}$	P_{MAX}	T_{MAX}
1.9 L	3	50 kW	250 N-m
2.5 L	4	56 kW	310 N-m

Table 3.1: Engine specifications used for hybridization

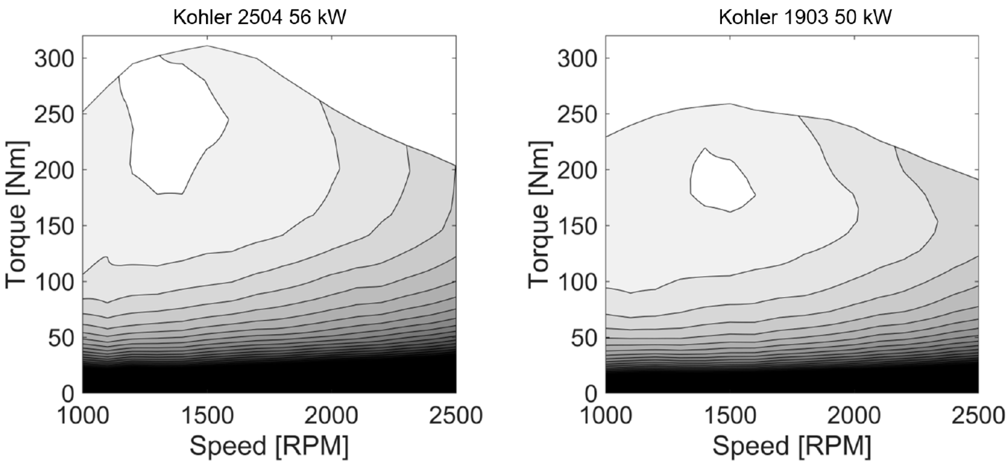

Figure 3.23: Reduced displacement ICE considered in the hybrid models [7]

Electric Machines

For this model, two electric motors were available. The first one, EM_1, is offered in both 48 and 96 V configurations. The 96 V version has a Wider constant-torque and therefore a higher power compared to the 48 V version. Both configurations are shown in figure 3.24.

The second one, EM_2, is a Synchronous Wound Field Machine, used for traction applications in passenger vehicles. These are shown in figure 3.25.

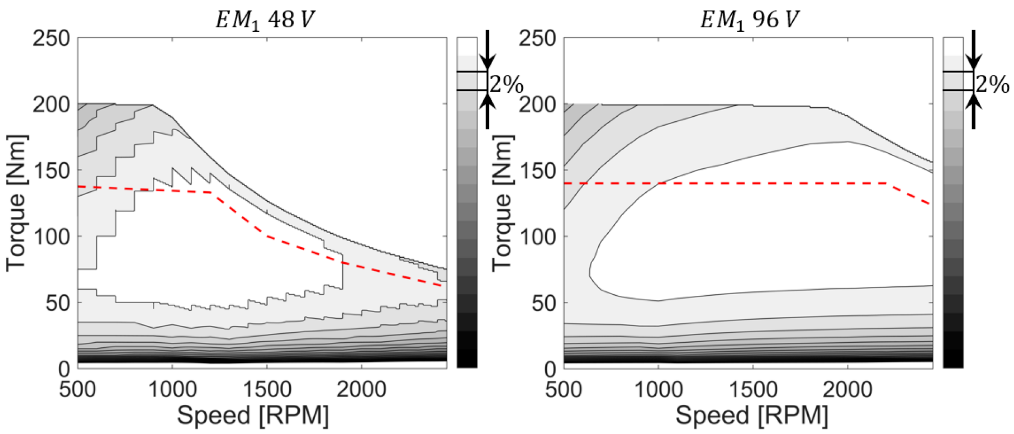

Figure 3.24: EM1 configurations considered in the Model. [7]

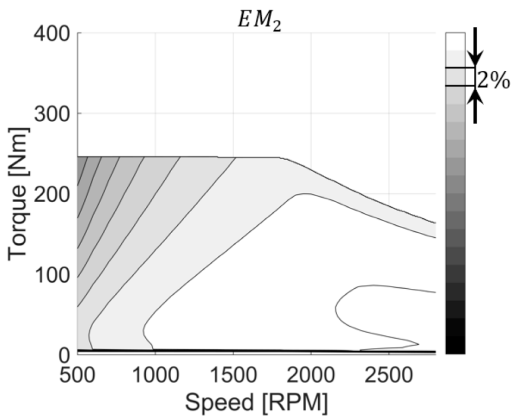

Figure 3.25: EM2 [7]

A recap of the available electric motors is shown in table 3.2.

Electric Motor	Peak Torque	MaxTorqueSpeed
$EM_1(a)$ - 48 V	200 N-m	0 - 1000 RPM
$EM_1(b)$ - 96 V	200 N-m	0 - 2000 RPM
EM_2 - 360 V	250 N-m	0 - 1800 RPM

Table 3.2: Electric motors specifications considered for hybridization

Battery Cells and Pack

A single battery cell is considered for each different battery pack architectures.

From the cell's parameters the battery pack layout was scaled, based on the necessary power, pack capacity, and other system parameters. The equations for the number of cells in series and parallel are shown below:

$$n_{series} = \frac{V_{nominal}^{system}}{V_{cell}} \tag{3.14}$$

The number of cells in parallel is given by equation 3.15.

$$n_{parallel} = \frac{P_{MAX}/V_{nominal}}{C_{Cell} \times C - Rate} \tag{3.15}$$

To calculate each of the characteristics of the battery pack, the equations shown in equation group 3.16. These will be used for different configurations to calculate the necessary characteristics of the pack.

$$V_{\text{nom,pack}} = N_s \, V_{\text{nom,cell}} \tag{3.16a}$$

$$C_{\text{pack}} = N_p \, C_{\text{cell}} \tag{3.16b}$$

$$E_{\text{nom}} = C_{\text{pack}} \, V_{\text{nom,pack}} \tag{3.16c}$$

$$I_{\text{max,pack}} = C_{\text{rate}} \, C_{\text{cell}} \, N_p \tag{3.16d}$$

$$P_{\text{max,dis}} = V_{\text{nom,pack}} \, I_{\text{max,pack}} \tag{3.16e}$$

The battery call considered is shown in table 3.3.

Cell Batteries	Capacity	C-Rate	Chemistry	R_i @ SoC $= 0.5$
Samsung [45]	3 Ah	12	LiPo	13 mΩ

Table 3.3: Battery cell main data

Chapter 4

Hybrid Layouts Analysis

When dealing with hybridization for all types of vehicles in general, it is important to remember that most often, it's not the case of "one size fits all". Depending on the use and work that the vehicle, road going or otherwise, different architectures may be more beneficial than others.

However, developing different prototypes, each with different architectures, component size, control strategies, and applying the same exact parameters, work/drive cycles, etc., especially in the Off-Highway sector, is not only very expensive, but also sometimes impossible.

Therefore, vehicle simulations carried out in a virtual environment such as GT - SUITE are not only far less expensive than building multiple physical prototypes, but they also allow a clearer comparison between architectures, free from variable external factors, and provide insight into the mechanisms behind the fuel-consumption gains of each configuration.

Simulation Setup

All simulations were run for an initial 64.1 seconds Y-cycle to check the power split, a repeated 641 second sequence of the Y-Cycle to check the fuel consumption, and then also extended 3000 seconds to test the Charge Sustaining. The initial 4 seconds of the simulations were set to idling. The initial SoC was set to 0.5 in all the models. The first two architectures, P1 and P1 + Electro-Hydraulic Actuators, were run at an engine target speed of 2300 RPM, and the last two were run at 2000 RPM for the e-Traction and 1500 RPM for the Full Series.

4.1 Hybrid Powertrain Architectures

There are many different levels and layouts of hybridization in all types of powertrains, however, the main ones that will be considered in this thesis are (in increasing order of electrification factor):

1. **Parallel P1**: This layout is commonly used in automotive applications. The Internal Combustion Engine and Electric Motor both work together, joined mechanically.
2. **Parallel P1 with Electro-hydraulic Actuators**: This layout splits the working hydraulics by using a dedicated electric motor driving the working hydraulics circuit pump.
3. **Parallel P1 with e-Traction**: In this layout, the entire hydrostatic transmission is replaced with traction electric motors on each side.
4. **Series**: Another very common Hybrid architecture in passenger vehicles, where the internal combustion engine works mostly as a generator, and the loads are driven by the electric motor.

4.2 Parallel Architecture - P1

Parallel P1 is a particularly attractive hybridization solution as it offers hybridization benefits, while requiring minimal changes to the machine's layout. This configuration consists of the electric motor being fixed between the engine and transmission. The electric machine is directly connected to the engine flywheel of the engine.

In this setting, the engine will cover most of the torque request during the duty cycle, while the electric motor will only be engaged during transients in the cycle, where the power and torque request is the highest. The downsized engine cannot cover that, so an electric machine is engaged. As seen in chapter 3, the average load request for the engine was much lower than the peaks during high acceleration, or bulldozing situations in the cycle.

A further advantage that P1 offers to the powertrain is that the engine can do what's called Load Point Shifting. This means that the engine will operate at a higher efficiency region, and move to a more efficient area of the BSFC map. However, the benefits of Load Point Shifting are relatively minimal in Off-Highway applications utilizing a hydrostatic transmission, as the engine can work only in a narrow band of speed, and has less freedom to move to more efficient areas.

For the Skid Steer Loader P1 model analysed in this thesis, a simplified schematic is shown in figure 4.1. As described, both the engine and electric motor will drive the HST and Working Hydraulics pumps.

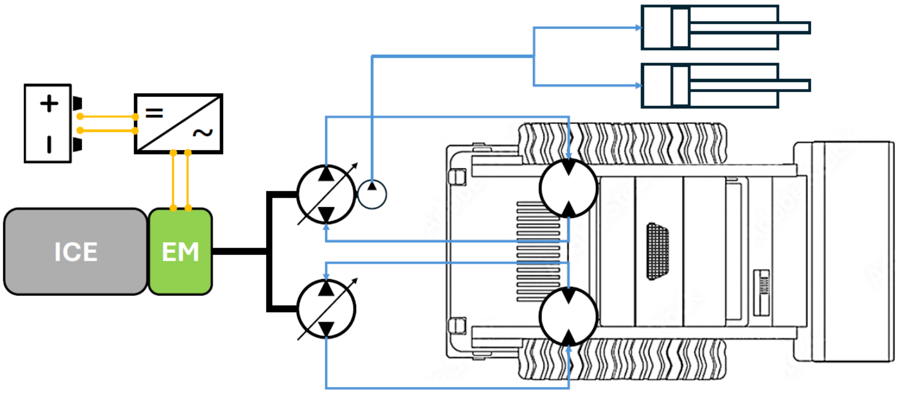

Figure 4.1: Parallel P1 Simplified Schematic

4.2.1 Component Sizing

A few configurations were simulated for this setup 4.2. With a reduced engine displacement and power output, a larger degree of electrification was attempted for the powertrain to cope with the system's demands.

Both downsized engines were considered, as described in the previous sections, and shown in figure 3.23.

The electric motor implemented is EM_1, 48 Volt version, which is shown in figure 3.24a. This motor is able to provide 200 N-m, however, the torque drops off at the speed which the powertrain will operate in.

Regarding the control of the Power split, the P1 operates in 4 main modes:

- **Normal**: When the engine is working in a generally efficient region the BSFC map, the battery SoC is inside its limits, and the load request is less than what the engine can provide, the engine will operate as it would in a conventional machine.
- **e-Boost**: The electric motor will provide an assist torque when the engine cannot cover it.
- **Charging**: When the SoC drops below a certain limit, the engine will produce extra torque to increase it to it's upper limit. If the engine needs e-Boost while the battery is within these two limits, the mode will switch back to e-Boost.
- **Load Point Shift**: When the engine is working in a BSFC map region which is not efficient (such as high speed, low load), and the battery SoC can absorb surplus power, the engine increases the

torque, and generates power, while avoiding low efficiency regions. Simulations showed that using LPS increased the engine efficiency by around 1.5%, compared to not implementing LPS.

The selected battery pack configuration consists of 14 cells in series and 13 cells in parallel (14s-13p). The cell's parameters are described in the table 3.3.

Based on the layout selected for this application, the battery cell specs specified in table 3.3, as well as the equations 3.16, the following results are calculated (table 4.1).

Quantity	**Symbol**	**Value**
Nominal pack voltage	$V_{\text{nom,pack}}$	52.08 V
Minimum pack voltage	$V_{\text{min,pack}}$	47.04 V
Maximum pack voltage	$V_{\text{max,pack}}$	56.98 V
Pack capacity	C_{pack}	39 Ah
Nominal stored energy	E_{nom}	2.0 kWh
Max. continuous pack current	$I_{\text{max,pack}}$	468 A
Max. discharge power	$P_{\text{max,dis}}$	24.37 kW

Table 4.1: Selected pack configuration parameters

ICE	**Electric Machine**	**Layout**	**Batt. Size**
2500 KDI	EM_1 (48 V)	14s 13p	2.03 kWh
1900 KDI	EM_1 (48 V)	14s 13p	2.03 kWh

Table 4.2: Main combinations for P1 Architecture

4.2.2 Results

2500 KDI P1 Architecture

In P1, where the engine will handle most of the mechanical load, the torque split will be discussed to gain some more insight on the EMS and it's operating modes. The torque split for the 2.5 L engine configuration is shown in figure 4.2.

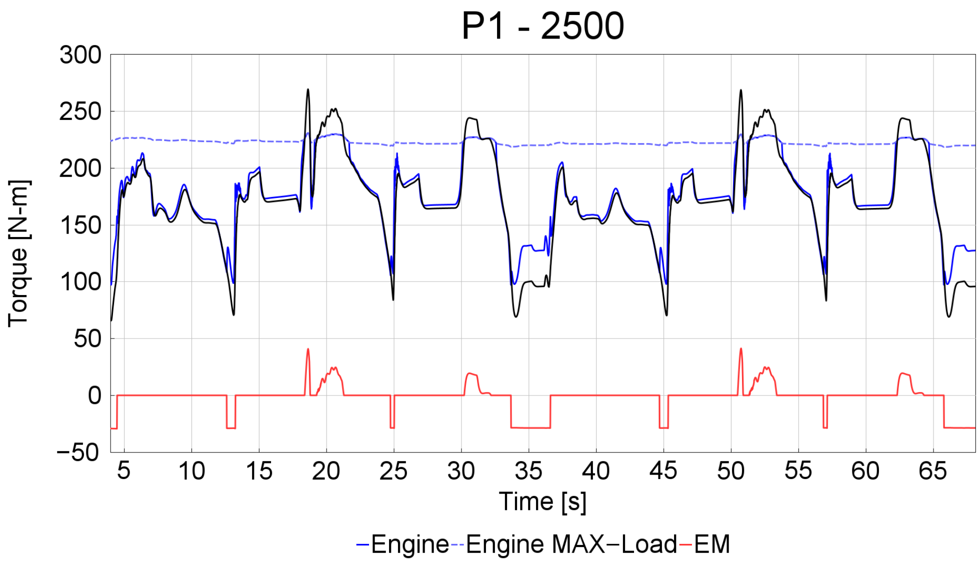

Figure 4.2: Torque split for 2500KDI - P1 Configuration

The operating modes in terms of torque can be observed over several time intervals. Initially, the engine operates in a purely mechanical mode, without battery charging or e-Boost, for example between 5 and 12 s. In this interval the engine simply follows the mechanical load demand.

The e-Boost function is activated during torque peaks, such as the bulldozing event at t = 18 s, and again during the acceleration phase between 18 and 22 s.

Load point shifting can be observed at time instants such as t = 13 s. At this point in the cycle, if the engine simply followed the load, it would operate at a low-torque point with poor efficiency. Instead, the EMS deliberately commands the engine to produce a higher torque, around 100 N-m, where the BSFC is lower. The extra torque is absorbed by the electric machine, which operates as a generator and charges the battery.

The same thing can be seen in the power split graph, in figure 4.3.

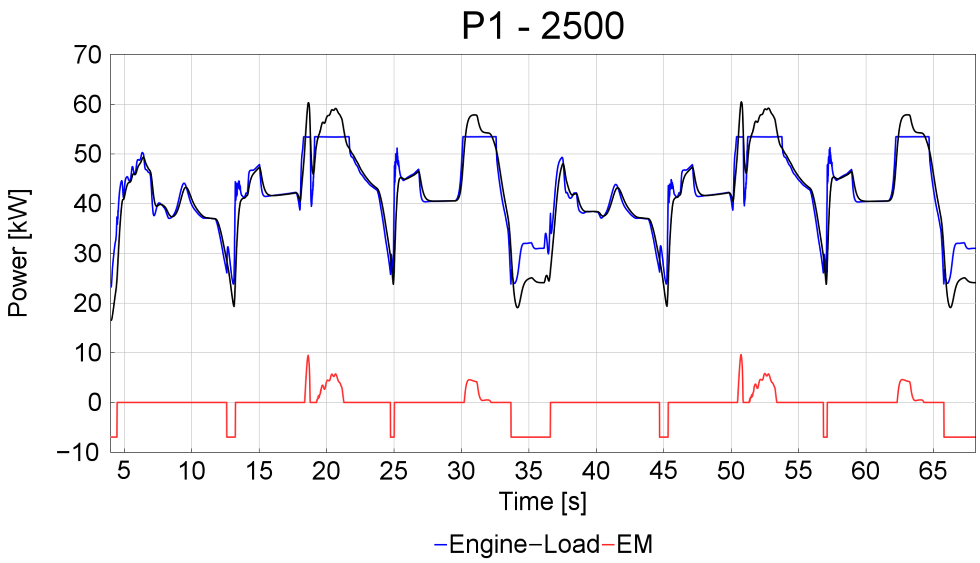

Figure 4.3: Power split for the P1 architecture with the 2.5 L engine

1900 KDI P1 Architecture

Similar results can be seen in the P1 architecture employing the 1.9 L 50 kW engine.

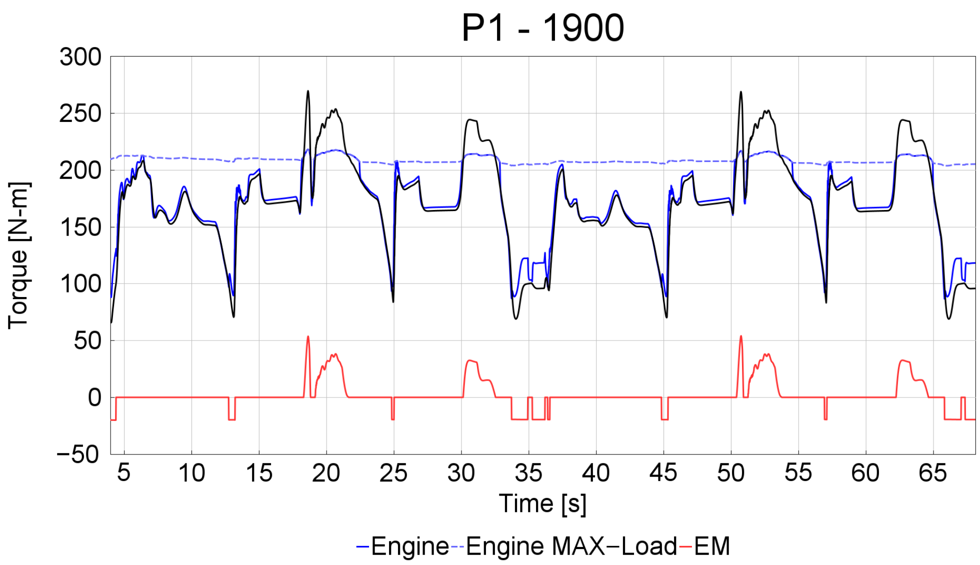

Figure 4.4: Torque split for 1900KDI - P1 Configuration

In the Torque Split graphs it can be seen that the electric machine will work mostly as a boost when the engine cannot provide the full torque, and occasionally as a generator when certain conditions, such as State of

Charge or Load Point Shifting torque level, are met. Compared to the 2.5 L engine, the 1.9 will work much closer to the maximum torque curve at the working speed.

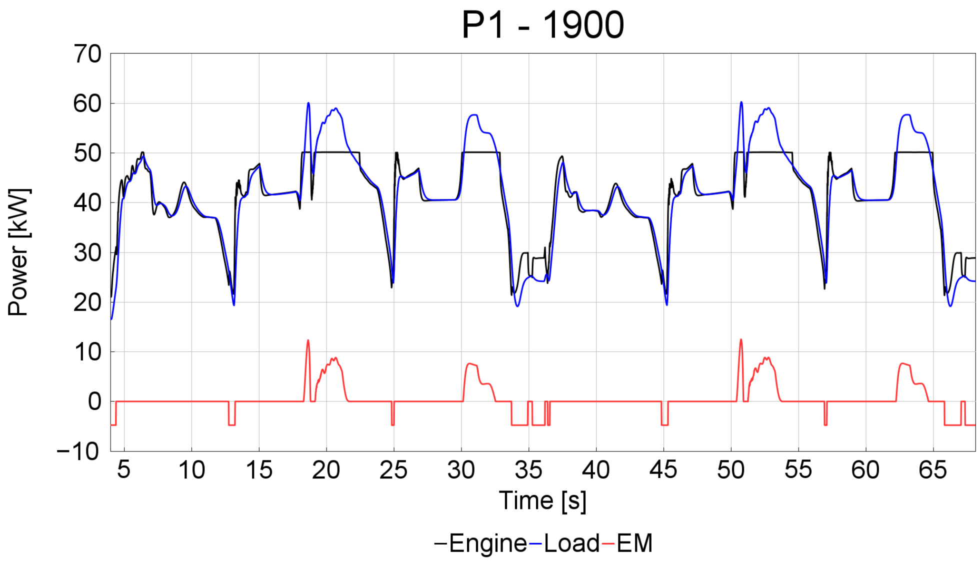

Figure 4.5: Power split for the P1 architecture with the 1.9 L engine

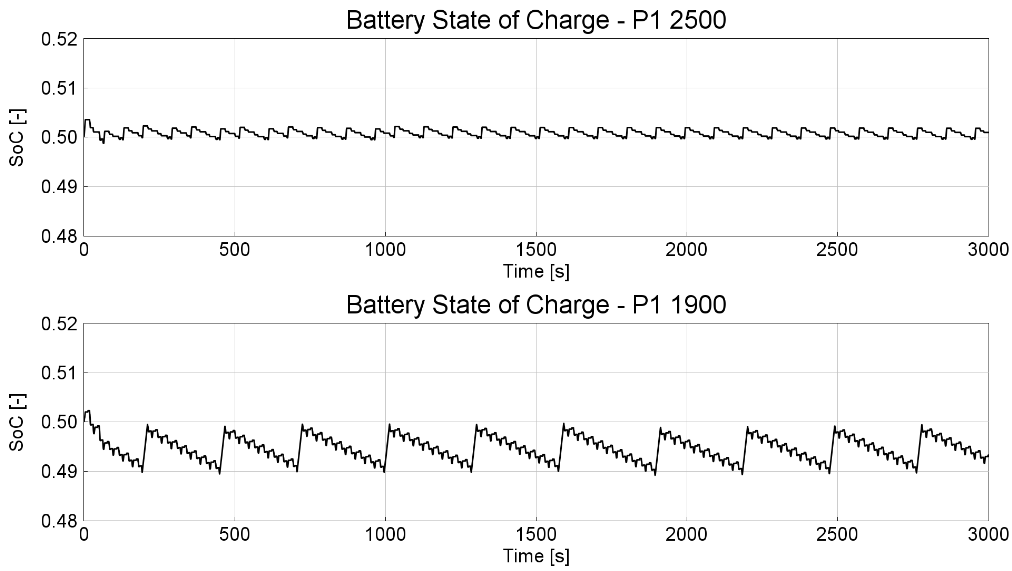

Figure 4.6: SoC Comparison between the two P1 configurations - Extended Cycle

Since the control is Rule Based, the charging will dependent on the load and e-Boost usage. As seen in the previous graphs, the 1.9 L version will need more e-Boost usage, and therefore more charging required for the battery to keep the desired State of Charge, where as the 2.5 L will need less, and thus be able to charge the battery mostly due to Load Point Shifting events.

Comparison

The final results indicate a significant fuel consumption reduction when implementing the P1 architecture. Compared to the 3.4 L engine conventional configuration, the reduction in fuel consumption, as well as model parameters are shown in table 4.3. The 1.9 L engine relies more on the e-Boost Mode, and thus spend more fuel on charge sustaining, while the 2.5 L less, since the engine can cover most of the cycle on its own.

However, since the 1.9 L works at a higher torque region compared to the 2.5 L during normal operations, it will have a lower BSFC, and thus it will have a minor improvement on fuel consumption of 1.04% compared to the 2.5 L version. While the 1.9 L P1 configuration is the most fuel efficient one, both configurations show a good improvement on fuel economy, while keeping the system very close to the original one. Depending on more, or less demanding mission profiles, both engines may yield better results.

ICE Displ.	**Electric Motor**	**SoC**$_{Initial}$	**Battery Config.**	**Avg Fuel Consumption**	**FC Reduct.**
3.4 L	-	-	-	10.53 kg/h	-
2.5 L	EM_1	0.5	14s13p	9.3 kg/h	11.49%
1.9 L	EM_1	0.5	14s13p	9.21 kg/h	12.54%

Table 4.3: Comparison between final Fuel Consumption figures of different P1 Configurations

To improve on these figures even more, a higher level EMS strategy can be implemented, such as ECMS, or even Rule Based Control derived from Dynamic Programming, such as Sub-Optimal Rule based, which extracts rules from DP results to obtain implementable rule-based control algorithms.

4.3 P1 with Electro-Hydraulic Actuators

Electro-Hydraulic Actuator systems is an architecture used in Off-Highway machinery that decouples the working hydraulic circuit from the engine and flywheel motor and implements an electric motor to drive the hydraulic pump. The HST continues to be powered by a P1 architecture, and the engine will also provide with the duties of charge charging as well.

Depending on the hydraulic system being utilized, and more specifically, working hydraulics pump, a number of benefits are seen from this. The biggest benefit is speed reduction. In a conventional or P1 system, the working hydraulics pump will rotate at a constant speed, since that's the required speed for the HST to work correctly. However, utilizing an independent electric motor, we can reduce the speed during idle periods, and thus reduce the inherent losses of the hydraulic system. One other benefit is regenerative action can be taken during the lowering phase of the arm. Much like in passenger vehicles utilizing regenerative braking, this applies a similar concept to Off-Highway machinery.

When arm is being lowered after the dumping phase is completed, it is in what's called "Overrunning conditions". This happens when the direction of force and speed are the same, and the system needs an overcentre (OVC) or even pressure relief valve set to a high pressure level to not have over-acceleration of the arm. However, with an independent electric machine driving the pump, designed and controlled to operate as a pump/motor pair and feed energy back to the DC bus, it will generate the necessary resistance and recuperate some of the energy which would be lost otherwise.

When this type of architecture has been implemented in a wheel loader, the proposed system could save up to 30% energy during a single boom cycle, 20% of input energy can be recovered and stored in the battery and hydraulic efficiency can reach up to 84% [46].

In the present implementation, the hydraulic architecture was kept unchanged, and only the pump drive was electrified to allow lowering the speed operation during idle periods.

A simplified schematic of the model built for this architecture can be seen in figure 4.7. In the figure it can be seen that another electric motor, labelled EM2, is added, and driving only the Working Hydraulics system's pump.

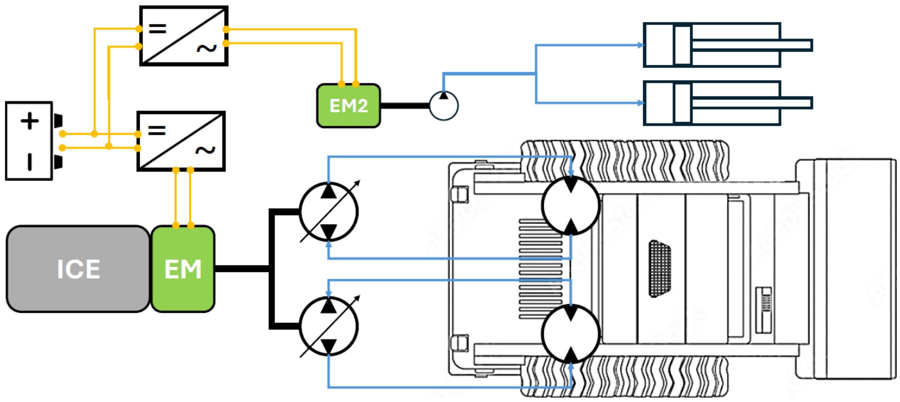

Figure 4.7: P1 + Electro-Hydraulic Actuators Simplified Schematic

4.3.1 Component Sizing

In comparison to the P1 Architecture, the addition of EHA system some components need changing. The flywheel electric motor was changed to the 96 V EM_1, and an additional 96 V EM_1 was added, driving the working hydraulics pump. This encompases the EHA system.

Batteries need to be changed to fit the new electric motor configuration in terms of voltage. The new configuration that will be used for this layout is 28s-11p. Although the pack capacity is reduced by about 15%, the higher number of cells in series (28s vs 14s) increases the total stored energy by about 69%. The difference between the two packs, based in the equations 3.16, is shown in table 4.4.

The engine considered will be the same described in 3.23.

N_{series}	$N_{parallel}$	**Total Capacity**	**Battery Size**	**Power Capacity**
14	13	39 Ah	2.03 kWh	24.37 kW
28	11	33 Ah	3.43 kWh	41.24 kW

Table 4.4: Comparison of the battery pack setup between P1 and P1 + EHA

EM Speed Relation to Pump requested Power

Since the working hydraulics pump is now decoupled from the powertrain that drives also the HST, which requires a constant speed of $n_{HST} \approx$ 2300 RPM, a new working speed will be set. To find the optimal speed for the pump and electric motor with the EHA component, three identical models were built, with different operating speeds:

- **Baseline**: The Pump rotates at a roughly constant speed of $n_{pump} = n_{ICE} = 2300$ RPM. This is the case considered for a non-EHA equipped architecture, like the conventional, or simple P1.
- **Case 1**: Idle speed was set to $n_{pump,idle} = 1300$ RPM, and the working speed was set to $n_{pump,work,1} = 2000$ RPM.
- **Case 2**: Idle speed was the same as in case 1, $n_{pump,idle} = 1300$ RPM, but the working speed was set to the normal $n_{pump,work,2} = 2300$ RPM.
- **Case 3**: The idle speed was set the same as the working speed, but the speed is reduced: $n_{idle} = n_{working} = 2000$ RPM.

Cases 1 and 2 highlight the effect of reducing idle speed on the power consumption of the working hydraulics pump. Since the conventional constraint of 2300 rpm is no longer present, they also show how the chosen working speed influences the pump power demand. Case 3 illustrates that power consumption can still be reduced even when a single constant speed is used. The baseline case is retained as a reference to quantify the benefits of each configuration. The results are shown in figure 4.8.

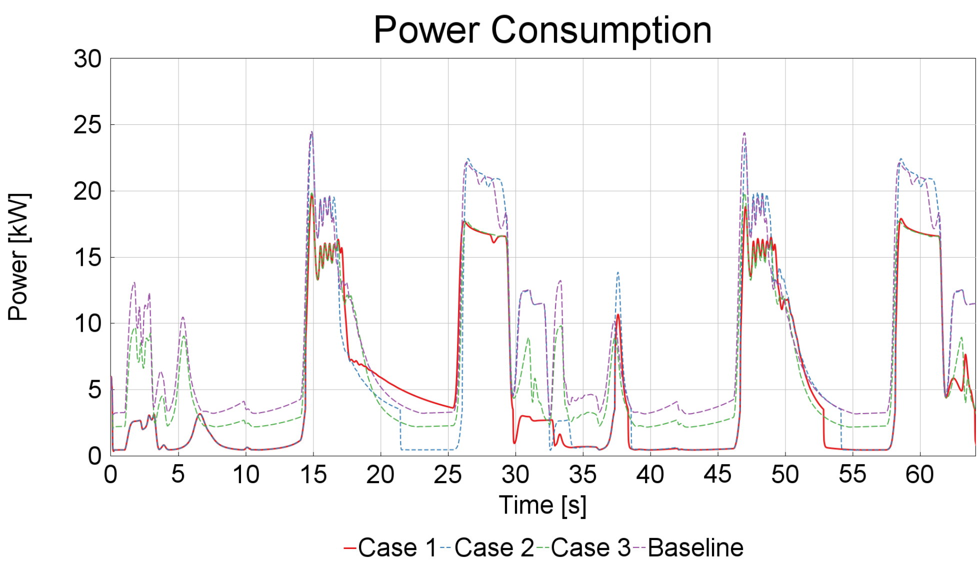

Figure 4.8: Comparison of Power Consumption of WH Pump

The speed plays a huge influence on the power consumption of the pump, not only in idle conditions, but also in working conditions. Lowering the speed reduces the delivered flow and, consequently, the amount of extra flow that is passed to tank in the open-centre circuit. For a given pressure level, the pump torque remains almost constant, while the mechanical power $P \approx \Delta p Q$ scales with speed n, so operating at lower speed directly reduces power consumption.

After the simulations, an approximately 39% reduction in the mechanical power drawn by the pump was observed.

Case	n_{idle} [rpm]	n_{work} [rpm]	P_{avg} [kW]	P_{max} [kW]	Reduction [%]
Baseline	2300	2300	8.19	24.48	-
Case 1	1300	2000	5.02	19.74	39
Case 2	1300	2300	5.92	24.50	28
Case 3	2000	2000	6.32	19.84	23

Table 4.5: Pump speed relation to power consumption

The behaviour of the system with the chosen parameters is illustrated in figure 4.9. The behaviour remains largely unchanged, with a slight initial dip in the performance of the tilt cylinder, but an improved performance on the lift cylinder as compared to the baseline.

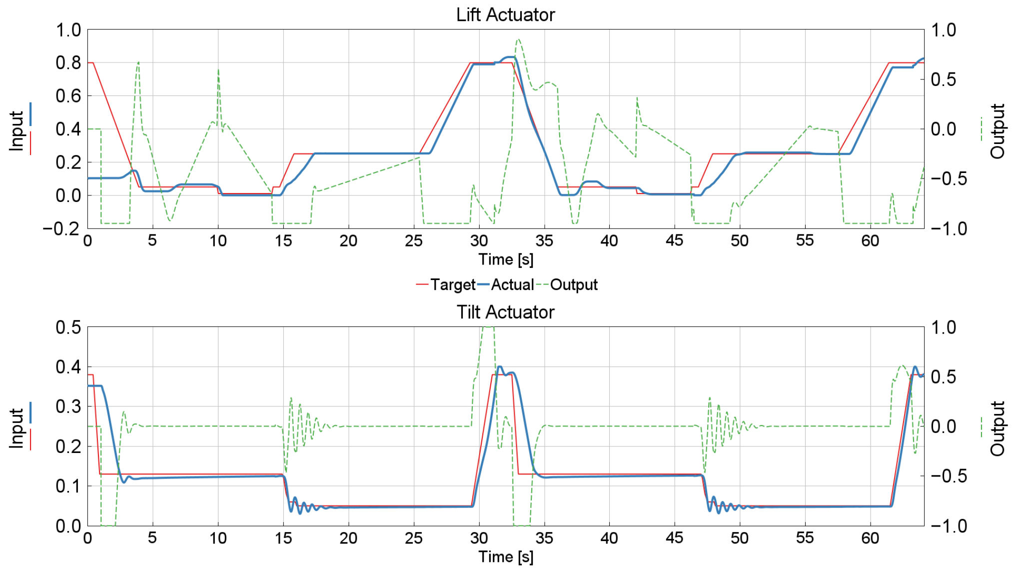

Figure 4.9: Tilt and Lift Behaviour with EHA

The comparison between the conventional and EHA behaviour is illustrated in figure 4.10. In the case of the Lift Actuators, the behaviour remains largely the same, except a slight dip at the beginning of the cycle, shown at around 3 seconds. However, this behaviour doesn't repeat in subsequent cycles. On the other hand, for the Tilt actuator, it's clear that the the behaviour has slightly improved, especially when the actuators are performing the emptying of the bucket, in the time period from 30 to 34 seconds.

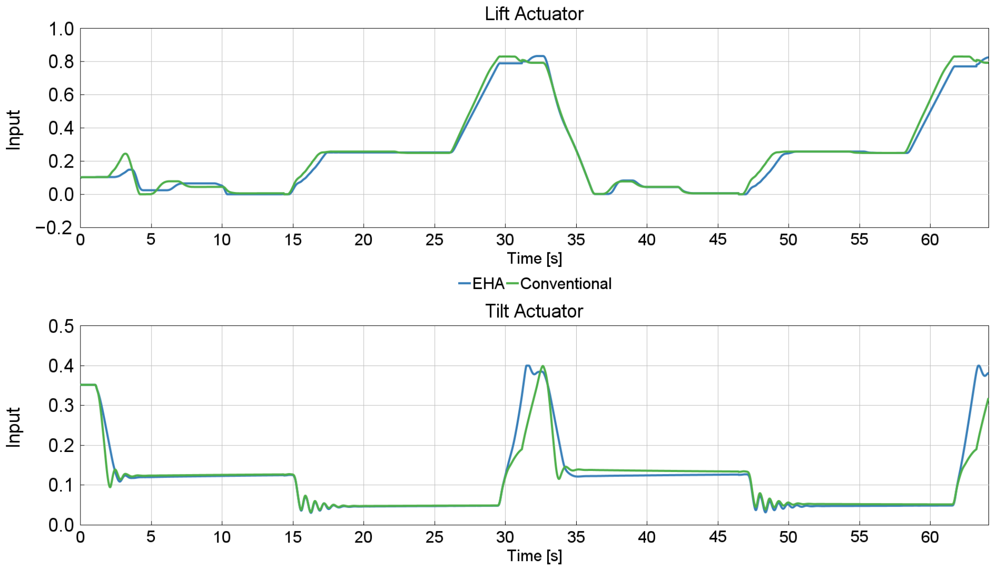

Figure 4.10: Comparison of system behaviour between conventional and EHA systems

Case 1, with $n_{\text{idle}} = 1300$ rpm and $n_{\text{work}} = 2000$ rpm, provides the lowest average and peak power while still satisfying the flow demand of the working hydraulics. Therefore, these conditions were set as final conditions for both configurations of the system, as the benefits in terms of power consumption are clear, and with no drawbacks in terms of performance.

4.3.2 Results

A similar Rule Based EMS to the P1 was implemented in here as well. The engine will not only charge when the SoC drops, but also when the torque demand is in a lower efficiency region. This yielded a minor improvement of fuel consumption of about 1% extra fuel consumption reduction.

2500KDI P1 + EHA Architecture

The benefits of EHA can be seen clearly in the case of the 2.5 L engine version. When the HST and working hydraulics are decoupled, the engine is able to cover fully the HST Power request, and in this case, the electric motor connected to the flywheel of the engine will only work as a generator.

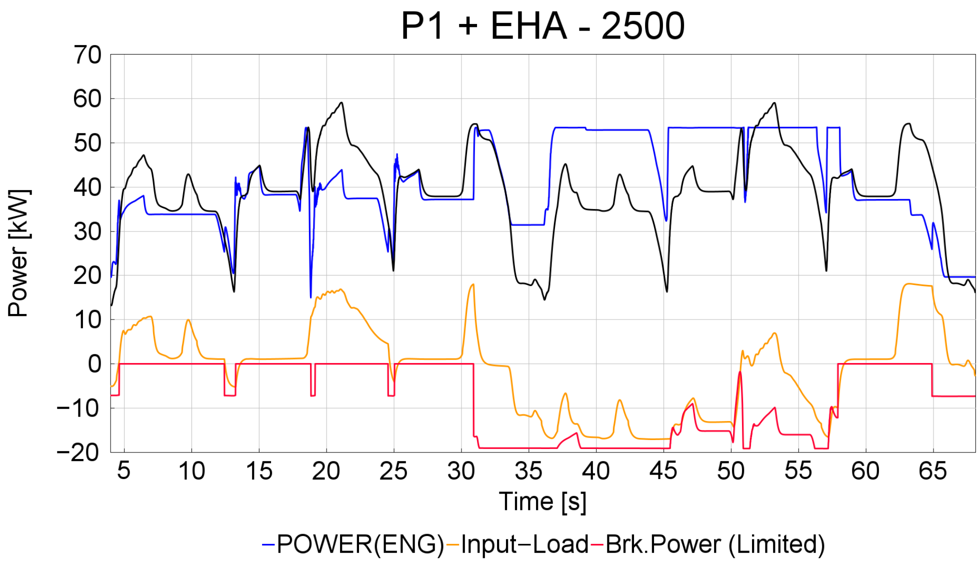

Figure 4.11: Power split in the 2500 KDI P1 + EHA Configuration

1900 KDI + EHA Architecture

Similarly to the 2.5 L engine system, the 1.9 L engine is also able to cover most of the HST torque requests on its own. The electric motor at the flywheel will work mostly as a generator, with occasional, minor (a few N-m) e-Boosts.

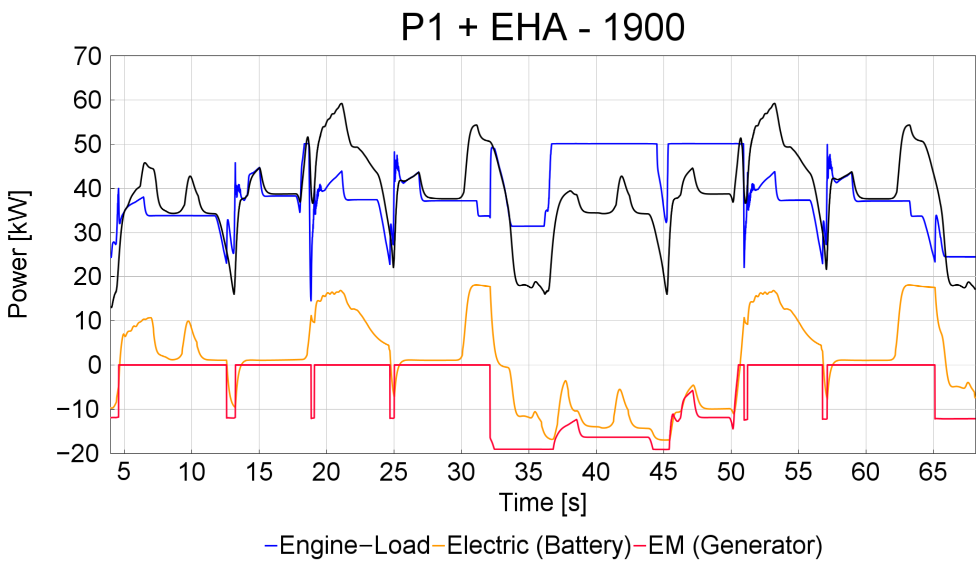

Figure 4.12: Power split in the 1900 KDI P1 + EHA Configuration

A similar control strategy to the P1 architecture was implemented here as well, with the the charging being done under specific conditions. The Rule Based has a priority in Charge Sustaining, while also employing a mild form of Load Point Shifting by increasing the engines torque output to a more favourable spot in the BSFC map when certain load conditions are met.

The State of Charge for the extended 3000 seconds cycle is shown in figure 4.13.

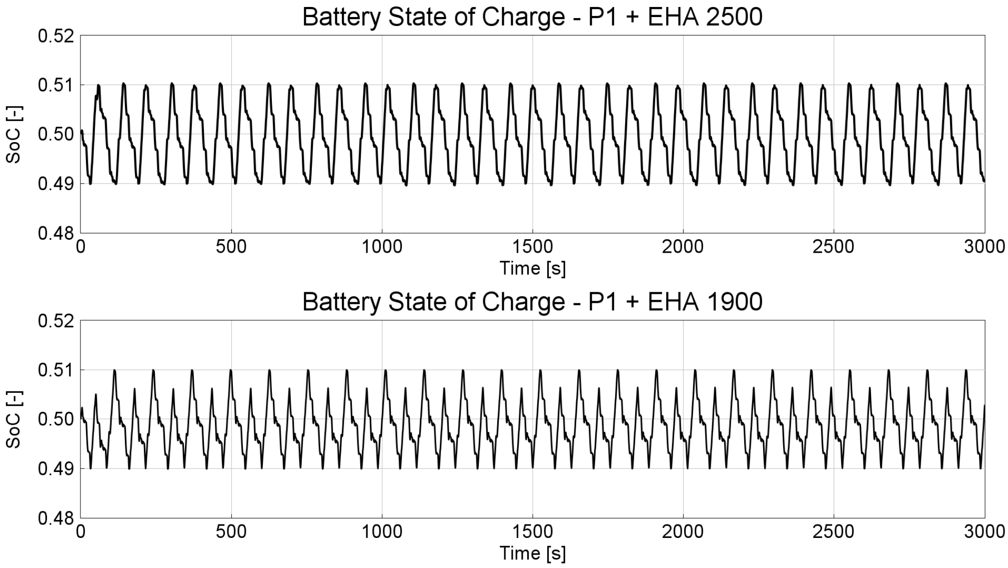

Figure 4.13: SoC comparison for the P1 + EHA architecture - Extended Cycle

Comparison

The final fuel consumptions figure show a good improvement over the previous P1 only architecture, and, of course, from the conventional 3.4L system.

ICE Displ.	EM P1	EM EHA	Battery Config.	Avg Fuel Consumption	FC Reduct.
3.4 L	-	-	-	10.53 kg/h	-
2.5 L	EM_1 - 96 V	EM_1 - 96 V	28s11p	8.91 kg/h	15.38%
1.9 L	EM_1 - 96 V	EM_1 - 96 V	28s11p	8.85 kg/h	15.94%

Table 4.6: Comparison between final Fuel Consumption figures of different P1 + EHA Configurations

Compared to their equivalently-sized engine architectures, the Electro-Hydraulic Actuator system shows a 3.89% fuel consumption reduction in the case of 2.5 L engine P1, and a 3.42% in the case of the 1.9 L engine P1. In contrast to the results in the P1 architecture, the 1.9 L engine model will provide only about 0.57% less fuel consumption than the 2.5 L engine system.

4.4 Electric Traction

Electric Traction is a hybrid architecture getting closer to the full series vehicle. In the e-Traction model, the HST is then replaced entirely with two electric motor for each sprocket. One proposition for a higher electrification on these machinery could be to simply implement an electric motor that drives the HST, like in the case of Volvo ECR25 Electric (mini excavator), which utilizes a 18 kW electric motor to drive the HST [47]. Another is swapping out the entire hydrostatic transmission for traction motors, called Electric Traction, or e-Traction.

e-Traction was chosen because this method is already implemented, in all electric loaders such as the Bobcat T7X and S7X. In these vehicles, the entire hydraulics are replaced by electric motors and ball screw actuators [39]. More similarly, other hybrid Off-Highway machinery such as the Hitachi ZH210LC-5B crawler excavator shown in figure 2.13, and the Hitachi ZW220HYB-5 Wheel Loader shown in figure 2.14 use a similar hybrid architecture, utilizing a series connection for the traction and a parallel connection for the working hydraulics.

To achieve the necessary torque output to cover the traction needs, the electric motors will need a reduction gearbox. While in a real vehicle, the gearbox will take some place in the device, it will still be a relative size compared to the HST system which it's replacing. Specifically, in loaders and other all electric NRMMs, a planetary gearbox with one or two stages is fit for this purpose. Some real world examples are the Bonfiglioli 600WE/700CE [48] and Dana track-drive data for 2–3 stage planetaries [49], with ratios 20:1 to 200:1, specifically designed for use in all electric, tracked, Off-Highway vehicles.

The engine will operate more as a generator, as well as provide some of the power for the working hydraulics, and auxiliary loads such as the cooling fan.

In a vehicle such as this, using electric motors instead of the hydrostatic transmission offers several advantages. The electric motors have a higher efficiency compared to the hydraulic components of the HST. On average, the electric drive system, considered in this thesis, will have a peak efficiency of about 92%, where as the HST, with only the pump and the hydraulic motor not considered ideal, will have a peak efficiency of about

80%. In a real system, the valves, and other smaller components will add to those inefficiencies. Of course, the electric traction circuit will have also more losses in a real vehicle compared to here, however, minor compared to hydrostatic transmission. Electric motors also increases responsiveness and accuracy of the machine.

Another benefit of the e-Traction is the engine operating mode and conditions. Since the engine is decoupled from the HST, the largest power consumer in the system, it will have a lower load request and it will also run at a lower speed, higher efficiency region. However, the engine still needs to maintain a certain speed, and not drop to low as it will be still used to power the working hydraulics.

Therefore engine speed was reduced to $n_{ICE} = 2000$ RPM. Simulations of the previous EHA architecture showed that the pump and working hydraulics system could work well at 2000 RPM, not only requiring less power 4.8, but also having a better responsiveness, as shown in figure 4.10.

A simplified schematic of the e-Traction model built for this study is shown in figure 4.14.

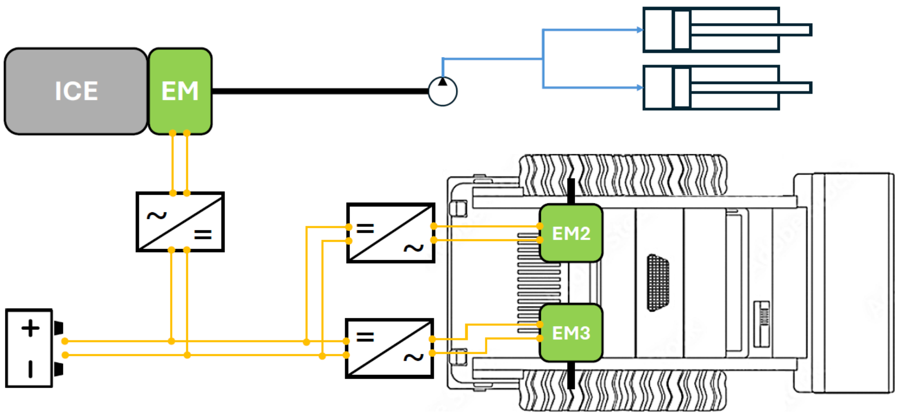

Figure 4.14: e-Traction Simplified Schematic

4.4.1 Component Sizing

One major consideration for the e-Traction model is the gearing issue. Since the most powerful electric motor available in this thesis can produce up to 250 N-m, a reduction is necessary to be able to provide the torque request at the sprocket, which can go as high as 7000 N-m. The reduction ratio needs to also be compatible with the speed. The sprocket needs to rotate about 70 RPM to do the cycle properly, while the motor rotates at a much higher speed. To provide the torque required at

$n_{sprocket} = 70$ RPM, the electric motor was fitted with a 30:1 reduction ratio.

The EM_2 (shown in figure 3.25) was chosen because it can provide the necessary torque and speed for the system. While a EM_1 96 V can also be used, it would require a higher reduction ratio.

The electric motor working connected to the flywheel of the engine was also changed to the EM_2, as shown in figure 3.25. This is because it can receive more torque at the speed which the engine will be set to work at. Changing the Flywheel motor is also necessary to fit the new battery configuration voltage limitations.

To supply the system, the battery was changed to a 97s-4p configuration. Details of the new battery used in this architecture is shown below, in table 4.7.

N_{series}	$N_{parallel}$	**Total Capacity**	**Battery Size**	**Power Capacity**
97	4	12 Ah	4.3 kWh	51.9 kW

Table 4.7: Battery Configuration of the e-Traction Architecture

4.4.2 Results

2500KDI P1 + e-Traction Architecture

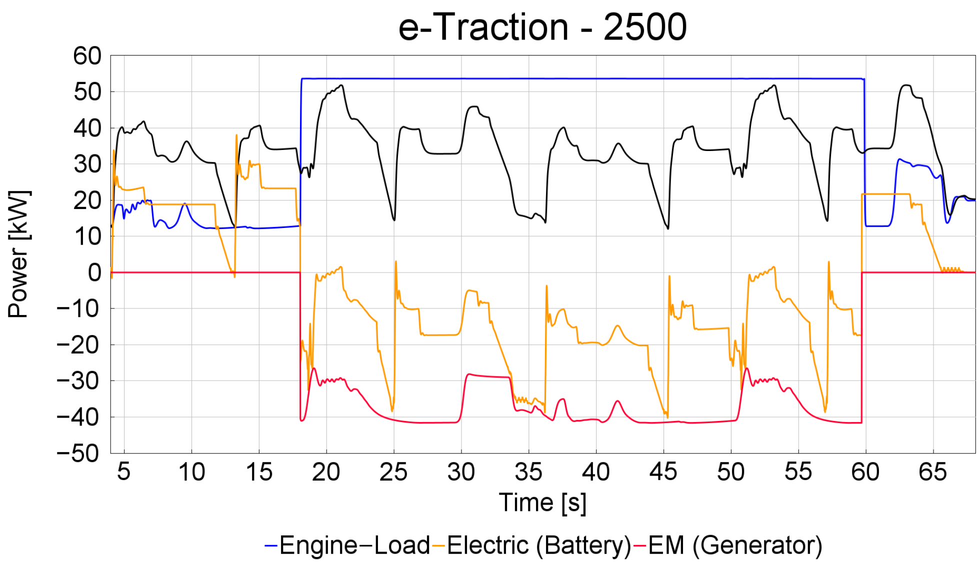

Figure 4.15: Power split in the P1 + e-Traction 2.5 L configuration

Since the largest power consumer of the system is disconnected from the engine shows a substantial change in the engine operating mode. The engine is now spending most of the power for the charging process, while also completing the auxiliary loads and working hydraulics power requests. In the initial time period from 4 to 18 seconds, the battery is inside the initial SoC and minimum SoC to allow for the engine to drive only the working hydraulics pump and auxiliary loads. Once the low SoC limit is hit, the engine increases its mechanical output to not only cover the working hydraulics and auxiliaries, but also provide additional torque to the flywheel-mounted electric machine operating as a generator, thereby charging the battery.

1900KDI P1 + e-Traction Architecture

Since the engine is mostly decoupled from the largest power consumer, these two configurations will be much more similar.The engine will stay around the 40 kW power output, with the occasional drop when no battery charging is required.

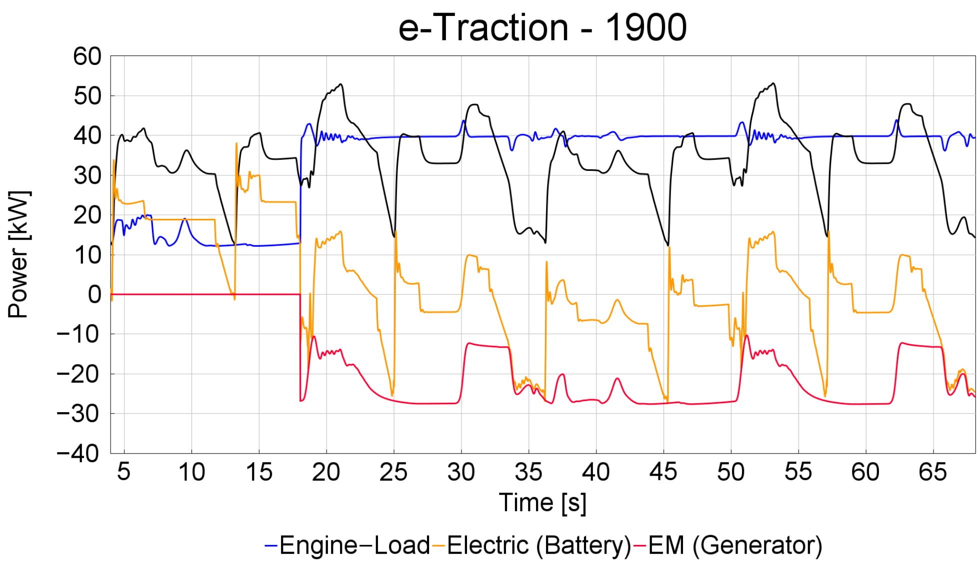

Figure 4.16: Power split in the P1 + e-Traction 1.9 L configuration

Comparison

One difference compared to the previous architectures was to increase the State of Charge operating range in the Rule Based algorithm.

Since in the previous architectures such as P1 the majority of the mechanical power for the load is provided by the engine, and a single 48 V, 200 N-m machine is used mainly for boost and regeneration, a relatively narrow SoC control window (0.49–0.51) is sufficient for proper functioning, while protecting the battery.

In the case of e-Traction, two higher-voltage traction motors (360 V, 250 N-m each) are the ones providing the most power to the mechanical load. This leads to larger and more frequent energy exchanges with the battery. Therefore, a larger SoC control window is adopted (0.48 - 0.52), providing additional buffer for traction while still remaining within a within the optimal operating range for the cells, and reducing undesired mode switching in the Rule Based energy management.

The comparison of the SoC of the two engine configuration is shown in figure 4.17. Due to the power limitations of the 1.9 L configuration, the charging periods will be slightly longer than in the 2.5 L configuration case, which is able to fully cover the charging demands and working hydraulic power request.

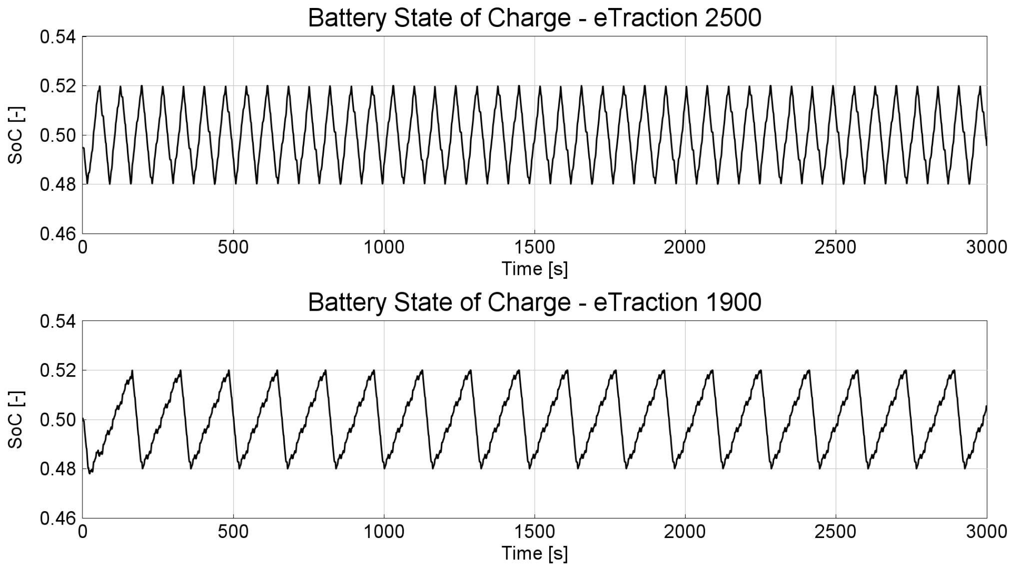

Figure 4.17: SoC comparison between the two e-Traction configurations - Extended Cycle

In the case of e-Traction, the majority of the fuel consumption reduction will come from the fact that the engine is working in a more efficient spot in its speed range. The engine speed cannot be reduced too much due to the working hydraulics operating speed, which needs to be maintained at 2000 RPM to avoid any drops in performance. In real vehicle usage, the engine may be dropped to even a lower speed, if the working hydraulics are not a priority (for example, the vehicle is only moving) or if a different working hydraulic circuit is available in that machine.

ICE Displ.	EM P1	EM Traction	Battery Config.	Avg Fuel Consum.	FC Reduct.
3.4 L	-	-	-	10.53 kg/h	-
2.5 L	EM_2 - 360 V	EM_2 - 360 V	97s4p	8.05 kg/h	23.52%
1.9 L	EM_2 - 360 V	EM_2 - 360 V	97s4p	7.89 kg/h	25.07%

Table 4.8: Comparison between final Fuel consumption figures of different P1 + e-Traction Configurations

The 1.9 L configuration shows about 1.52% reduction in fuel consumption compared to the 2.5 L engine configuration.

4.5 Full Series Hybrid

Finally, a Full Series Hybrid system is simulated. In this case, the engine will work only as a generator, as well as maintaining the auxiliary loads such as the hydraulic fan mentioned earlier, in Chapter 3. In contrast to the e-Traction model, the full series will also utilize an electric motor for the working hydraulics circuit.

The biggest benefit of the Series architecture, much like in passenger vehicles that utilize this type of powertrain, is that the engine will be operating in its highest efficiency region, or the lowest BSFC region for the system.

In figure 4.18, a simplified schematic of the model is shown.

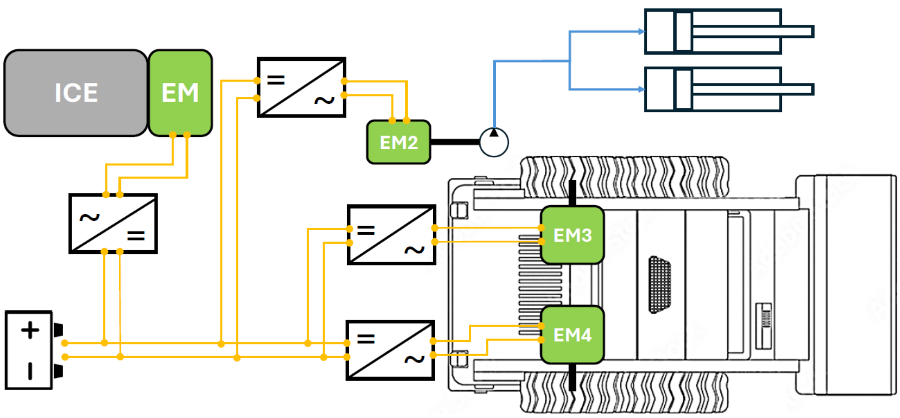

Figure 4.18: Series Simplified Schematic

4.5.1 Component Sizing

All the electric motors were kept the same as in the previous configuration, with the addition of the EHA motor, which was also changed to the EM_2 - 360 V version to avoid any mismatches with the battery pack.

The battery is kept the same as in the e-Traction model, shown in table 4.7, and the engine speed was set at 1500 RPM. This is a major benefit of the Full Series, as it allows the engine to work in a more efficient region, and can produce more torque with respect to the 2000 RPM or 2300 RPM working points.

4.5.2 Results

Here it can be seen how the EMS operates. Generally, the engines will work in a load following mode, while also directly driving the the auxiliary loads. The engine was made to stay at a medium load region, despite the load request dropping, to ensure it's operating at the most efficient region. When the SoC drops below its minimum of 0.48, the engine will increase it's mechanical output to charge the battery, while also driving the auxiliary loads.

2500KDI Full Series Architecture

The 2.5 L engine is better at maintaining the Load following for a longer time without draining the battery, compared to the 1.9 L.

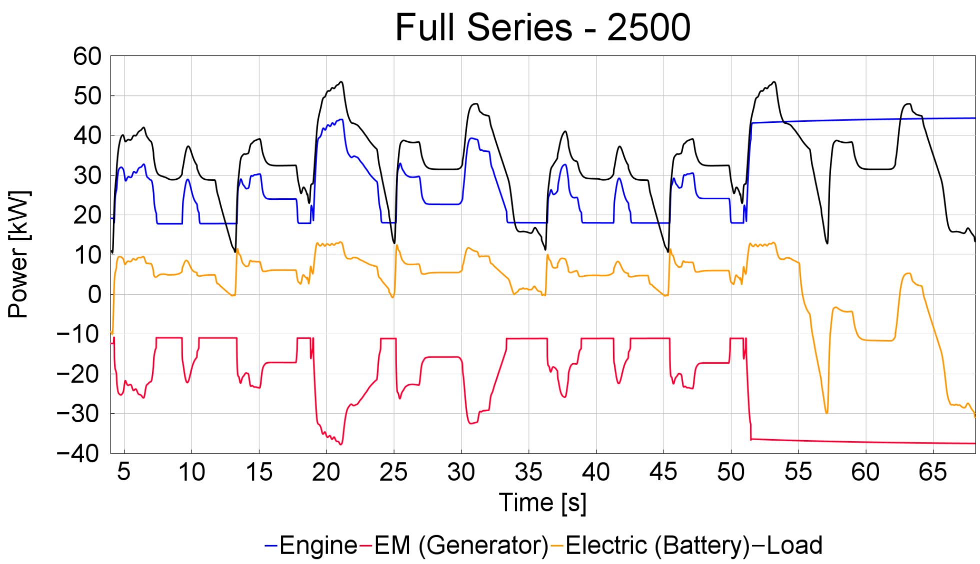

Figure 4.19: Power split in the Full Series 2.5 L configuration

1900KDI Full Series Architecture

A similar power split can be seen in the 1.9 L configuration. However, since this engine has a lower torque rating at that operating speed compared to the 2.5 L configuration, it's unable to keep the load following and auxiliary loads at the same time for long, and thus will resort to drawing more battery power. This leads to the engine defaulting to more high torque regions of the map more often than the 2.5 L configuration.

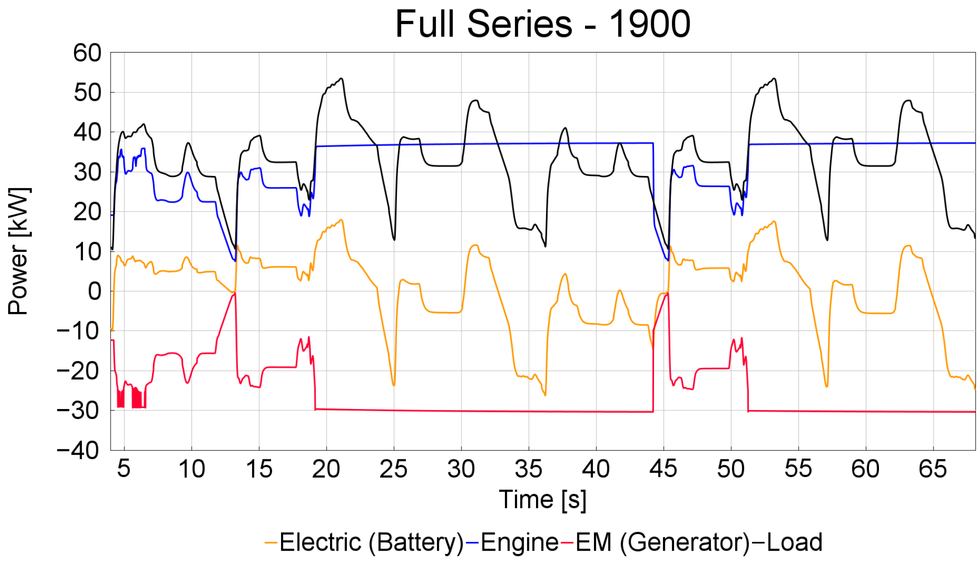

Figure 4.20: Power split in the Full Series 1.9 L configuration

Comparison

The CI engines considered in this study have their highest efficiency regions in the 1200 to 1500 RPM, as shown in figure 3.23. The working speed was set to 1500 RPM, since there are no longer restrictions in terms of engine speed like in the previous architectures. This yielded a large reduction in fuel consumption.

The engine runs in two main modes: a charging mode, activated when the SoC drops below the lower threshold, and a reduced load-following mode when the SoC lies between the two limits. Owing to its mechanical characteristics, the 2.5 L engine can supply a higher continuous load-following torque to the generator and therefore requires fewer dedicated charging events.

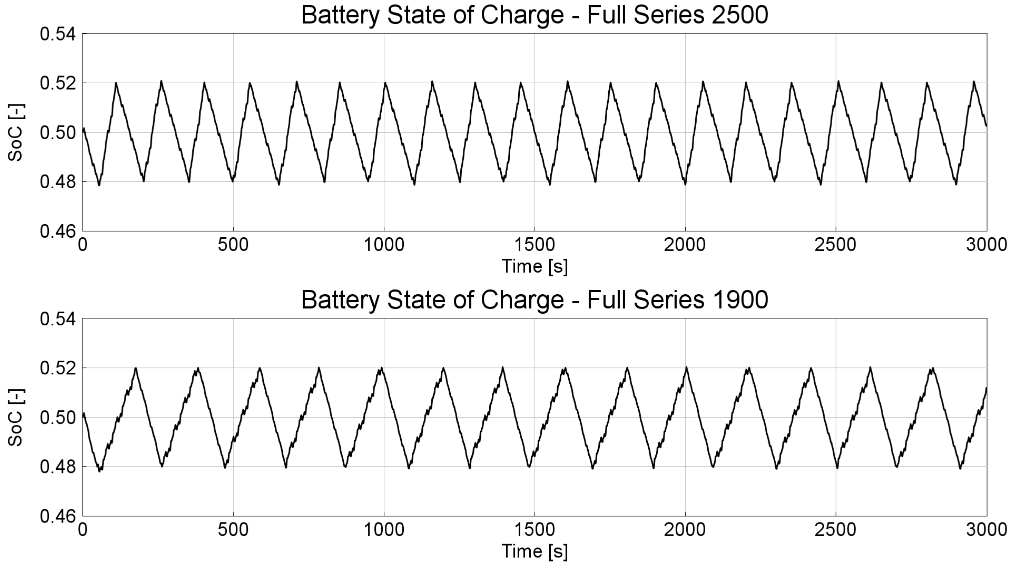

Figure 4.21: SoC comparison between the two Full Series configurations - Extended Cycle

ICE Displ.	**EM P1**	**EM Traction**	**Battery Config.**	**Avg Fuel Consum.**	**FC Reduct.**
3.4 L	-	-	-	10.53 kg/h	-
2.5 L	EM_2 - 360 V	EM_2 - 360 V	97s4p	6.64 kg/h	36.94%
1.9 L	EM_2 - 360 V	EM_2 - 360 V	97s4p	6.59 kg/h	34.42%

Table 4.9: Comparison between final Fuel consumption figures of different Full Series Configurations

The 1.9 L configuration shows about 0.47% reduction in fuel consumption compared to the 2.5 L engine configuration.

4.6 Comparison of Different Architectures

Among the aspects that influence the choice of hybrid architecture are not only fuel consumption reduction, and performance related reasons, but also the typical use and application, availability of electrification components, and other technical, economic and practical considerations. Depending on whether the producer and the buyer prioritise reduced fuel consumption, higher performance, or a compromise between development and operating costs, different architectures can be more suitable. Figure 4.22 shows the fuel reduction for each architecture and engine configuration, and table 4.10, offers a more comprehensive overview of each architecture with the configuration with the highest reduction is presented.

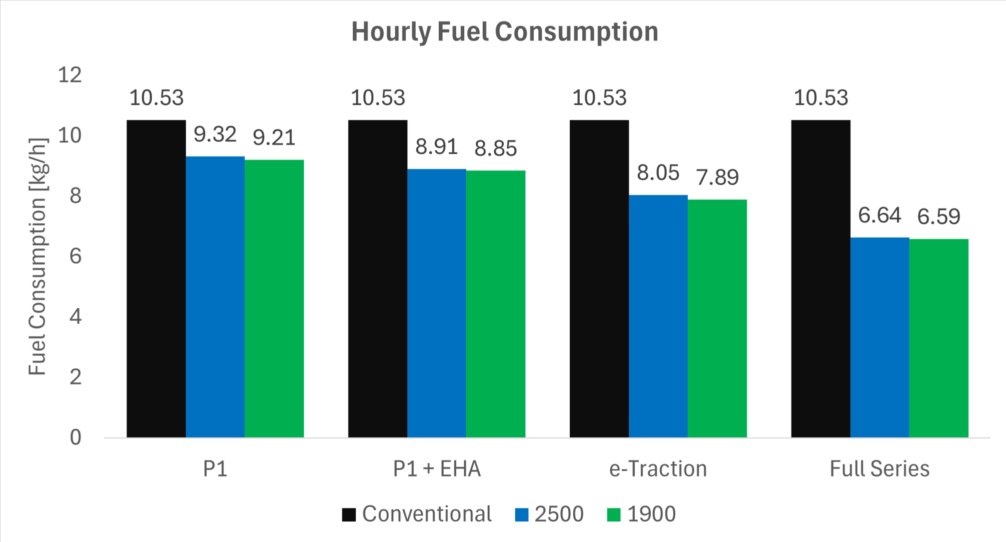

Figure 4.22: Fuel Consumption reduction figures, categorized by type of Architecture and engine configuration

Powertrain	**Consumption** (kg/h)	Δ_{FC} (%)	**Battery** (kWh)	**EM** [-]	**Voltage** (V)
3.4 L 80 kW (Conv.)	10.53	-	-	-	-
P1 (1.9 L)	9.21	-12.5	2.03	1	48
P1 + EHA (1.9 L)	8.85	-16.0	3.4	2	96
e-Traction	7.89	-25.1	4.3	3	360
Full Series (1.9 L)	6.59	-37.4	4.3	4	360

Table 4.10: Overview of the most relevant hybrid configurations (relative the conventional baseline). All hybrid configurations listed utilize the 1.9 L downsized engine

The results show that with an increased level of electrification, fuel consumption reduction will be higher.

This can be attributed partly to downsizing, and partly to the engine being progressively decoupled from the main mechanical loads. In addition, implementation of simple EMS control strategies are used to improve the engine's operating point, such as increasing the engine torque in low load regions (similar to Load Point Shifting, which pulls the engine away from non-efficient regions, but without changing the speed), and using the electric machines for charging and boost. However, depending on the architecture, for this machine the potential of Load Point Shifting is very limited. At the engine's operating speed the BSFC curve is monotonically decreasing with load, so there is no clear efficiency "sweet spot" that the EMS can reach by redistributing torque between the engine and the electric machine. Regardless, the implementation of these strategies showed in the P1 and P1 + EHA configurations minor 1% to 1.5% improvements compared to a fixed Rule Based EMS with no Load Point Shifting.

The biggest jump in fuel consumption reduction occurs when moving from architectures with a hydrostatic transmission such as P1 or P1 + EHA to the e-Traction configuration. This indicates that for this type of mission profile, the hydrostatic transmission contributes heavily to the fuel consumption, not only due to the inefficiencies, but also the high load. Therefore, decoupling the engine from vehicle speed by means of electric traction is a particularly effective hybridisation step.

Another benefit of decoupling the engine from the biggest load, is also that we are allowed to lower the engine speed. This helps the efficiency and fuel consumption, because the engine will work at a speed where the BSFC is lower. This point is further illustrated, since in the case of the Full Series configuration, the engine speed was set to 1500, which is in the high efficiency speed band of both considered downsized engines, the fuel consumption decreases more with respect to e-Traction than P1 with Electro-hydraulic Actuators increased with respect to the P1 architecture, even though the added electrical load is the same in both cases.

Different configurations will have different benefits, despite the lower or higher levels of fuel consumption reduction. For example, the P1 architecture yields good fuel reduction, yet can be achieved with a simple 48 V mild hybrid system. The 48 V system is also particularly attractive, since its a low voltage system and not safety features need to be implemented.

On the other hand, architectures like the e-Traction and Full Series, while they offer significantly more fuel consumption reduction, require a total redesign of the system, with HST being replaced with Traction motors, implementation of a 360 V system, larger batteries and implementation of a one speed, high ratio gearbox. Their benefit relies in the engine working in a slightly more efficient speed region (in the case of e-Traction), and the optimal operating range of the engine in the case of Full Series.

4.7 Off-Highway Considerations

As summarized above, the hybridization of Skid-Steer Loaders is a viable option when considering the reduction of fuel consumption and exhaust gas emissions, and possible reduce the total ownership and operating costs with no performance downgrades.

Powertrain

The primary advantage of these machines can exploit is engine downsizing. The electric motor can kick in when the downsized engine needs more mechanical output. However, this requires that the engine can comfortably cover the average mechanical load. Especially in the 56 kW to 90 kW power output bracket, the sector can employ minimal hybridization as an alternative to expensive and complicated after-treatment systems required by current emissions regulations.

Despite the fact that in this specific case, where the engine speed is connected to the hydrostatic transmission speed Load Point Shifting implementation has minimal effect, utilizing a similar EMS strategy may yield better results in machines that do not require a fixed engine speed, or generally use a mechanical or hydro-mechanical transmission such as wheel loaders and telehandlers with powershift or CVT drivelines, agricultural tractors with CVT, or on-road trucks with automated manual gearboxes. The engine speed in these applications can be varied over a wider range, so both speed and mechanical output can be varied, allowing for proper LPS.

Machine Layout

As the results showed, decoupling the engine from the mechanical loads, and especially the largest one which is traction, yielded much better results in terms of fuel consumption. As the results showed, decoupling the engine from the mechanical loads, and especially from the largest one, traction, yields much better results in terms of fuel consumption. However, achieving this requires replacing the hydrostatic transmission, which is an essential subsystem in off-highway machinery such as skid-steer loaders and crawler excavators, with electric traction motors. These motors must be able to deliver comparable torque and dynamic response while still remaining decoupled from the engine. Even without accounting for the fact that the HST will generally have a lower efficiency than an electric traction system, changing the HST for the traction motors enables the operating speed of the engine to sit at a lower BSFC region, and this enables savings in fuel consumption. In the architectures considered here, replacing the hydrostatic transmission with traction motors required the introduction of a mechanical gearbox. Although similar solutions have already been adopted on comparable machines, they are not yet widespread in this sector and may still be perceived as less familiar by potential users.

Another benefit of the traction motor can be taken from automotive applications. This is regenerative braking, which allows some of the energy that would otherwise be lost to be recovered during vehicle slowing down periods. While it remains to be seen if it can be applied to Skid-Steer Loaders, it enables even more saving that aren't possible with the HST.

Similarly, the simple implementation of EHA systems, as shown in the models in this thesis, can enable good power consumption reduction from the working hydraulics (for this application ΔP = -39%), and, with a redesign of the system, can enable energy regeneration when the load is going down, or when the boom is going down. This can be applied in all Off-Highway machinery that utilize working hydraulics circuit, and yield good results. Specifically, Wheel Loaders, Skid-Steer Loaders, Telehandlers, and Forklifts can benefit from this, as they are used to lift heavy weights, and can regenerate energy during the down manoeuvre.

Improving efficiency of the components and different subsystems of a Off-Highway machinery therefore is a necessary step, that will lead to lower fuel consumption, and sometimes better performance.

Electric Motors

As it was seen in the architectures employed above, different architectures will require different levels of electrification and component sizing for the electrical components.

Starting from the P1 which employed a mild hybrid 48 V system, this was enough to provide the necessary additional power and torque to cover for downsizing the engine from the 80 kW 3.4 L conventional powertrain, to the 56 kW 2.5 L and 50 kW 1.9 L engines, despite the torque drop off at a speed lower than the engine operating speed.

Going to higher power requirements from the electrical components, such as in the case of e-Traction and Full Series, larger voltage motors and larger batteries are required.

The operating speed of each electric machine must be compatible with the speed imposed by the architecture. In the P1 layout, the flywheel-mounted motor rotates at the engine speed (2300 rpm), and at this operating point the 48 V machine can still deliver sufficient torque to cover both the charging phases and the short e-Boost events. Therefore, a larger or higher-voltage motor is not required in this case.

Developing an electric machine with a higher efficiency in the relevant speed–torque region of the engine could yield better results. Such a motor should be designed to operate on the flywheel side and to provide the required power for both charging and e-Boost phases.

However, when more power needs to be absorbed, like in the cases where electric traction was employed, a higher torque motor, with a voltage matching that used in the battery pack or electric motors may be necessary.

For traction motor applications, the electric machine must satisfy both the speed and torque requirements at the sprocket. In particular, it needs sufficient low-speed torque to meet the maximum traction demand. These requirements vary between applications, depending on the vehicle duty cycle and on the selected reduction ratio of the gearbox between the motor and the sprocket.

Batteries

Battery sizing in these applications is not only a matter of total energy capacity, but also of power delivery capability. The smaller pack used in the P1 configuration are sized as power buffers, that can deliver a relatively high power over short bursts, to support the operations of the downsized engines, and then recover the lost SoC during low load phases.

In the P1 + EHA configuration, the battery is expanded a bit due to two reasons: the battery must provide more power for larger periods of time, as well as regenerate that over low load conditions, or if the SoC targets are met. They must be also able to handle more frequent charge/discharge cycles, while keeping the SoC inside said limits.

For architectures in which the electrical system becomes the main source of mechanical power, namely e-Traction and Full Series, the batteries were made larger and configured to deliver the voltage level needed by the higher-power traction motors. In the cases of P1 + EHA, e-Traction and Full Series therefore the role of the batteries is less and less that of the support, temporary peak power delivery setup.

From a practical point of view, it is convenient if the battery pack, inverters and electric machines work at the same nominal DC voltage. If the battery voltage and the machine DC bus voltage are different, additional DC–DC converters are needed to step the voltage up or down, which introduces extra cost and losses. In this work such converters are not considered: for each hybrid architecture it is assumed that all electrical components are connected to a single DC bus and therefore share the same nominal voltage. The voltage level, and therefore the number of cells in series, are dictated by the needs of the system. Initially in the P1, the 48 V system was enough. For higher electrification architectures, especially the ones utilizing traction motors, a higher voltage system was necessary.

Chapter 5

Conclusions

This thesis focused on the development of a virtual test bench for evaluating the hybridization of a Off-Highway machinery, in particular a Skid-Steer Loader. The model represents both the hydrostatic transmission and the conventional working hydraulics circuit. The model was then tested in a characteristic mission profile, called the Y-Cycle. On this basis several hybrid powertrain architectures were implemented and compared against a conventional diesel model, with a focus on the impact on fuel consumption, operating strategy and component utilisation. The overall objective was to understand which configurations of electrification and control can contribute to decarbonising this type of machine without compromising its performance.

The work started by presenting a review of the available research on the topic, as well as exploring the different, currently available, hybridized models of Off-Highway machinery. The review highlighted the main used architectures in these applications and typical control strategies. It also underlines the challenges in the hybridization process of Off-Highway. A particular focus was given to the high power and torque request, the interactions between traction and working hydraulics, and the demanding duty cycle these machines need to complete. These reasons make the prototype based development particularly costly, and open room for model based design using a forward simulation approach to evaluate and optimize this hybridization process.

On this technical background, a detailed 1D forward facing model was implemented in GT - SUITE, modeling the hydrostatic transmission, the working hydraulic circuit, the internal combustion engine, and the vehicle dynamics under a representative Y-Cycle mission profile. The hydraulic side represents a conventional open-centre working hydraulics system with parallel directional control valves, capturing the main throttling and pressure losses over a realistic Y-cycle duty. The internal combustion engine is modelled using performance and BSFC maps, allowing simulation of different operating points on the engine map and the evaluation of possible downsizing. Electric motors and battery packs were introduced as modular elements that allowed for interchange and reconfiguration in different hybrid architectures. Particular attention was paid to the integration of the mechanical, electrical and hydraulic subsystems, to ensure correct model behaviour, and realistic results. This allowed for the same cycle to be run in different configuration and thus a common comparison frame in terms of fuel consumption, performance, charge sustaining capabilities, and power split, was developed.

Starting from the reference conventional model, initially a P1 architecture was analysed. Then, engine downsizing was investigated, which showed good fuel consumption reduction while keeping the functionality and performance of the vehicle. In this layout, the electric motor, bolted to the flywheel, was used to compensate for the engine's power and torque reduction. After that, a configuration combining the previous P1 architecture with the Electro-hydraulic Actuator was developed. This decouples the working hydraulics from the engine and flywheel machine, allowing for lowering the working and idle speeds of the working hydraulics pump. More advanced and hybridized setups, such as e-Traction and P1 architecture in conjuncture, in which the use of traction motors instead of the hydrostatic transmission allowed for decoupling the engine from the largest mechanical load. The final step was to create a Full Series hybrid, in which, the engine is able to work at the optimal operating point. All the models were simulated using the same working cycle, and in charge sustaining mode, to ensure that the fuel consumption of each architecture were compared to a fair basis.

The simulations showed that fuel consumption decreases as electrification levels increase, with varying degrees of engine decoupling from the mechanical load. In the case of P1, the biggest benefits come from moderate downsizing of the engine, and to a lesser degree to low implementation of load shifting the engine to more efficient working regions. While simple to implement, one drawback of this architecture is that its direct relation of engine speed and HST speed means that more aggressive and efficient Load Point Shifting cannot be fully implemented due to the speed restriction. When the Electro-hydraulic Actuators are implemented, throttling losses in the working hydraulics system are reduced, and the hydraulic power reduces significantly. Electrifying traction and moving towards e-Traction layout increases the flexibility of the system. The engine is used mainly as a generator and operated closer to its best efficiency region, making more advanced EMS strategies more effective.

Then, the engine is decoupled completely from the mechanical loads related to traction and actuation system, and thus work in the most efficient region of the engine map. This yields by far the largest reduction in fuel consumption, but also very little change between the two engines considered. Overall, the systems confirm that largest consumption reduction comes from those architectures that are able to decouple the engine from more of the mechanical loads.

Beyond the numerical values, these results underline the role of high accuracy modeling as a cheap and reliable way to assess the hybridization impact on both fuel consumption and performance before investing in prototypes. The virtual test rig has made it possible to identify that downspeeding associated with EHA systems can be introduced without compromising the ability of the machine to not drop in performance. This kind of insight on specific subsystems is especially valuable in Off-Highway applications, since each machine is comprised of many subsystems, and developing of prototypes, changing components on said prototypes, etc., is very expensive. This also shows, how small changes can have substantial effect on fuel consumption, emissions, performance, and operating costs of the vehicle. More generally, the work confirms that hybridization solutions can be tailored to specific working cycles, and specific machines can contribute in the de-carbonization of the Off-Highway sector, and with the combination of more efficient engines, and better, purpose-made, components can achieve even further improvements, provided that they are supported by a robust modeling and simulation framework.

Different directions for future work emerge from this study. A first step is the optimisation of hybrid architectures considered here, including the sizing of components, such as engines, electric motors, batteries and hydraulic components. Thanks to the forward-facing nature of the model, a productivity analysis could be added, so that fuel consumption and emissions are evaluated together with cycle time, or task completion metrics, enabling truly exhaustive comparisons. Another important development is the experimental verification of the model against data from a test rig or physical prototype, in order to refine key sub-systems and increase confidence in the predictions. Finally, the implementation of more advanced, real time EMS strategies such as Equivalent Consumption Minimization Strategy - ECMS or Model Predictive Control - MPC. This would mean moving away from Rule Based control towards optimisation based control that can be implemented in a real machine. This would guarantee an optimal control strategy, and a more optimised strategy, which can extract further fuel consumption reduction from the system.

Bibliography

[1] U.S. Department of Transportation, "DOT report to congress: Decarbonizing U.S. transportation," tech. rep., U.S. Department of Transportation, Washington, DC, July 2024. Figure 2: "Total 2022 U.S. GHG emissions with transportation and mobile sources breakdown." Available at: `https://rosap.ntl.bts.gov/view/dot/75677`.

[2] Caterpillar Inc., "Backhoe loader maintenance," 2025. Available at: `https://www.cat.com/en_GB/support/maintenance/compact-equipment-maintenance/backhoe-loader-maintenance.html`.

[3] Deere & Company, "John deere italia," 2025. Available at: `https://www.deere.it/it-it`.

[4] U.S. Environmental Protection Agency, "EPA nonregulatory nonroad duty cycles," 2017. Available at: `https://www.epa.gov/moves/epa-nonregulatory-nonroad-duty-cycles`.

[5] S. Vaschetto and L. Rolando, "Hybrid and electric propulsion systems," *Politecnico di Torino*, 2024. Course guide. Available at: `https://didattica.polito.it/pls/portal30/gap.pkg_guide.viewGap?p_cod_ins=01ONKQD&p_a_acc=2026&p_header=S&p_lang=IT&multi=N`.

[6] Hitachi Construction Machinery Co., Ltd., "The New Generation: ZH210-5 Hybrid Hydraulic Excavator (Model Code ZH210LC-5B) — Product Brochure." Product brochure (PDF), 2015. Hosted by Nasta AS; accessed 2025-11-22.

[7] A. Vinogradov, *Comprehensive Methodology for the Development of Electrified Powertrain Solutions for Off-road Applications.* PhD thesis, Politecnico di Torino, 2025.

[8] J. Wang, Z. Yang, S. Liu, Q. Zhang, and Y. Han, "A comprehensive overview of hybrid construction machinery," *Advances in Mechanical Engineering*, vol. 8, no. 3, pp. 1–15, 2016.

[9] F. Mocera and A. Somà, "A review of hybrid electric architectures in construction, handling and agriculture machines." In: *New Perspectives on Electric Vehicles*, 2021.

[10] Compact Equipment Magazine, "Caterpillar compact track loader specs and product line summary for 2025," 2025. Available at: https://compactequip.com/compact-track-loaders/caterpillar-compact-track-loaders-summarized-2022-spec-guide/.

[11] A. Bertini, M. Ceraolo, and G. Lutzemberger, "Systematic approach in the hybridization of a hydraulic skid loader," *Elsevier*, 2015.

[12] M. Rundo, "Mobile and industrial fluid power, politecnico di torino," *Politecnico di Torino*, 2024. Course guide. Available at: https://didattica.polito.it/pls/portal30/gap.pkg_guide.viewGap?p_cod_ins=01HMBQD&p_a_acc=2026&p_header=S&p_lang=IT&multi=N.

[13] Bobcat Company, "T650 compact track loaders spec sheet," 2010. Available at: https://www.bobcat.com/na/en/equipment/loaders/compact-track-loaders/t650.

[14] J. Han, L. Zhang, L. Jiang, and Q. Wei, "A control method for the differential steering of tracked vehicles driven independently by a dual hydraulic motor," *Applied Sciences*, 2022.

[15] DieselNet, "EU: nonroad engines." https://dieselnet.com/standards/eu/nonroad.php#s5.

[16] European Rental Association and EY, "Energy transition in rental: Non-road mobile machinery impact on climate." https://erarental.org/wp-content/uploads/2024/10/Energy-Transition-Project-Presentation.pdf, 2025. Presentation, 4 June 2025.

[17] T. S. Poulsen, "Market analysis for non-road mobile machinery sector," tech. rep., Scandinavian Green Public Procurement (GPP) Alliance, 2017. Report for the Scandinavian Green Public Procurement Alliance. Available at: https://carbonneutralcities.org/wp-content/uploads/2019/06/2.Market-Analysis-for-Non-Road-Mobile-Machinery-Sector_FINAL.pdf.

[18] M. Allam, O. Fernandez, and M. Linjama, "Fuel efficiency analysis and control of a series electric hybrid compact wheel loader," *SAE International*, 2024.

[19] A. Lajunen, P. Sainio, L. Laurila, J. Pippuri-Mäkeläinen, and K. Tammi, "Overview of powertrain electrification and future sce-

narios for non-road mobile machinery," *Energies*, 2018.

[20] Vehicle Certification Agency, "Non-road mobile machinery (NRMM)." https://www.vehicle-certification-agency.gov.uk/vehicle-type-approval/what-is-vehicle-type-approval/non-road-mobile-machinery-nrmm/.

[21] Liebherr-Werk Bischofshofen GmbH, "New liebherr mid-sized wheel loader series," 2022. Available at: https://www.liebherr.com/en-il/n/new-liebherr-mid-sized-wheel-loader-series-22124-5345836.

[22] Volvo Construction Equipment, "Volvo launches new generation of wheel loaders," 2025. Available at: https://www.volvoce.com/united-states/en-us/about-us/news/2025/volvo-launches-new-generation-of-wheel-loaders/.

[23] Liebherr-International Deutschland GmbH, "Crawler excavators," 2025. Available at: https://www.liebherr.com/en-int/earthmoving/products/excavators/crawler-excavators-5376587.

[24] Volvo Construction Equipment, "Ew145b prime wheeled excavator," 2025. Available at: https://www.volvoce.com/middle-east/en-sa/products/excavators/ew145b-prime/.

[25] Komatsu Europe International N.V., "Pc240lc/nlc-11 standard crawler excavator," 2025. Available at: https://www.komatsu.eu/en/excavators/crawler-excavators/standard-crawler-excavators/pc240lcnlc-11.

[26] Liebherr-Mining Equipment GmbH, "R 9600 mining excavator," 2025. Available at: https://www.liebherr.com/en-ie/mining-equipment/equipment/mining-excavators/r-9600/r-9600-5337578.

[27] Caterpillar Inc., "793d mining truck," 2025. Available at: https://www.cat.com/en_IN/products/new/equipment/off-highway-trucks/mining-trucks/13894258.html.

[28] European Union, "Commission regulation (eu) 2017/1151 supplementing regulation (ec) no 715/2007 on type-approval of motor vehicles with respect to emissions (wltp/rde)." Regulation (Official Journal of the European Union), 2017. Accessed 2025-11-22.

[29] UNECE, "Un global technical regulation no. 15: Worldwide harmonized light vehicles test procedures (wltp)." Regulatory document (PDF), 2022. Accessed 2025-11-22.

[30] DieselNet, “Cars and light trucks: Real driving emissions (rde) testing.” Webpage. Accessed 2025-11-22.

[31] DieselNet, “Ftp-75 emission test cycle.” Webpage. Accessed 2025-11-22.

[32] U.S. Environmental Protection Agency, “40 cfr part 600 — fuel economy testing; 5-cycle calculations from ftp, hfet, us06, sc03 and cold ftp.” Code of Federal Regulations (PDF via GovInfo), 2025. Accessed 2025-11-22.

[33] DieselNet, “Epa highway fuel economy test (hwfet) cycle.” Webpage. Accessed 2025-11-22.

[34] Sumitomo Construction Machinery Co., Ltd., “SH200HB/SH220HBL-7 Hybrid Excavator Catalog.” Product catalog (PDF), 2020.

[35] Construction Equipment Guide, “Volvo Unveils L220F Hybrid Wheel Loader at ConExpo.” Webpage, Mar. 2008. Published March 11, 2008; updated April 11, 2008; accessed 2025-11-22.

[36] K. Ishida and M. Higurashi, “Hybrid wheel loaders incorporating power electronics,” *Hitachi Review*, vol. 64, 2015.

[37] Komatsu, “Komatsu WE-series wheel loader with Generation 3 hybrid technology offers sustainable operation with great fuel efficiency, strength and versatility.” Press release (Komatsu Newsroom), 2021.

[38] Global Highways, “Imer’s hybrid technology unveiled in skid steer.” Webpage, Apr. 2015. Accessed 2025-11-22.

[39] Bobcat Company, “T7x & s7x all-electric compact track & skid-steer loaders,” 2024. Concept & Future Products. Available at: `https://www.bobcat.com/na/en/equipment/future-products/t7x-s7x-all-electric-compact-loaders`.

[40] G. Kotiev, B. Padalkin, A. Miroshnichenko, A. Stadukhin, and B. Kositsyn, “A theoretical study on the high-speed electric tracked vehicle mobility,” *IOP Conf. Ser.: Mater. Sci. Eng. 820*, 2020.

[41] B. Company, “T76 compact track loader.” `https://www.bobcat.com/eu/en/equipment/loaders/compact-track-loaders/t76`. Product specifications, accessed 29 Nov 2025.

[42] B. Company, “T86 compact track loader.” `https://www.bobcat.com/eu/en/equipment/loaders/compact-track-loaders/t86`. Product specifications, accessed 29 Nov 2025.

[43] C. Inc., “279d3 compact track loader.” `https://www.miltoncat.com/machinery-equipment/new/skid-steer-loaders/`

279d3-compact-track-loader. Product specifications, accessed 29 Nov 2025.

[44] C. C. Equipment, “Tv380b compact track loader.” https://www.casece.com/en/africamiddleeast/products/b-series-compact-track-loaders/tv380b. Product specifications, accessed 29 Nov 2025.

[45] P. Kollmeyer and M. Skells, “Samsung INR21700 30t 3ah li-ion battery data,” 2020. Available at: https://doi.org/10.17632/9xyvy2njj3.1.

[46] S. Qu, E. Busquets, R. Cardoso, and Z. Liu, “Electrification of a compact skid-steer loader – redesign of the hydraulic functions,” *2024 International Maha Fluid Power Conference*, 2024.

[47] Volvo Construction Equipment, “ECR25 Electric – Electric Machines,” 2024. First in a New Range of Battery-electric Compact Excavators. Available at: https://www.volvoce.com/asia/en-as/products/electric-machines/ecr25-electric/.

[48] Bonfiglioli S.p.A., “700 ce – riduttori epicicloidali per traslazione,” 2025. Riduttore Epicicloidale per Trasmissioni di Cingolati, Disponibile anche in Versione Elettrica. Available at: http://bonfiglioli.com/italy/it/prodotto/700-ce.

[49] Dana Incorporated, “Dana planetary drives for tracked and wheeled vehicles,” 2025. Spicer Torque-Hub Planetary Drives with Ratio Range 26:1–210:1, Configurable with Hydraulic or Electric Motors. Available at: https://www.oemoffhighway.com/drivetrains/axles-driveshafts/product/22017839/dana-incorporated-dana-planetary-drives.