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Design and Experimental Analysis of an Adjustable Geometry Vehicle with Independent Drive-Steering for Energy Efficiency Optimisation



THE UNIVERSITY OF
WAIKATO
Te Whare Wānanga o Waikato

A thesis submitted
in fulfilment of the requirements
for the degree of

Master of Engineering with Endorsement in Robotics

At

The University of Waikato

Ray (Rui) Shang

2026

Declaration

In submitting this thesis to The University of Waikato School of Engineering, I affirm my
Awareness of the university's standards and honour code.

Name: Ray Shang

Signature:

The image shows a handwritten signature in Chinese characters, which appear to be '尚锐' (Shang Rui), written in a cursive style.

Abstract

Agricultural vehicles operating in orchard and pasture environments are required to perform frequent low-speed turning manoeuvres on uneven, grass-covered terrain - conditions under which steering strategy and vehicle geometry are major determinants of energy consumption. Despite a well-developed body of mathematical models predicting these effects, experimental validation has remained limited due to the absence of physical platforms that combine adjustable wheelbase, adjustable track width, and switchable steering strategies within a single vehicle.

This thesis addresses that gap through the design, construction, and experimental evaluation of a fully electric, adjustable-geometry agricultural robot platform. The vehicle employs four independent in-wheel drive-steer modules, each integrating a brushless DC drive motor and a dedicated steering motor, mounted at the corners of a reconfigurable semi-monocoque chassis. Both wheelbase and track width can be independently adjusted across three values - 400 mm, 600 mm, and 800 mm - yielding a full 3×3 factorial matrix of nine geometric configurations. Six distinct steering modes are supported: two-wheel steering arc turn ($R = 1.5$ m), four-wheel steering arc turn ($R = 1.5$ m), skid-steer arc turn ($R = 1.5$ m), two-wheel point turn ($R = 0$), four-wheel point turn ($R = 0$), and skid-steer point turn ($R = 0$). Motor current is continuously logged at all eight drive and steer motor channels via a dedicated data acquisition system, enabling direct electrical energy quantification as $E = \sum |I| \cdot V \cdot \Delta t$.

Controlled field experiments were conducted on two terrain conditions representative of New Zealand orchard and pasture environments: near-horizontal grass (mean inclination 0.216°) and a steep grass slope (mean inclination 17.618°). On flat terrain, all nine geometric configurations were tested across all six steering modes. On the slope, three full-width-track configurations ($B = 800$ mm) were evaluated for safety.

Experimental results demonstrate that wheelbase L is the dominant geometric parameter governing turning energy consumption, with strong positive correlation across most steering modes (Pearson $r = 0.80$ - 0.90). Short-wheelbase configurations ($L = 400$ mm) consistently achieve 25-35% lower turning energy than long-wheelbase equivalents ($L = 800$ mm). The globally optimal configuration - B600 $\times$ L400 and B800 $\times$ L400 in four-wheel point-turn mode - achieves a minimum turning energy of approximately 707-710 J, representing a 31% reduction

relative to the worst-performing configuration. Among all tested platforms, B800×L400 (track width 800 mm, wheelbase 400 mm, B/L = 2.00) produced the most outstanding overall performance across both flat and sloped terrain, exhibiting superior agility, smooth and rapid manoeuvre execution, and excellent directional stability, including on the 18° incline. Skid-steering modes uniformly incur the highest energy penalties across all geometric configurations (3-25× relative to the optimal equivalent mode), and a predictive L^2/B parameter is shown to capture skid-steer energy with high accuracy ($r = 0.86-0.98$). Four-wheel steering consistently outperforms two-wheel and skid-steering strategies, and the arc four-wheel steering mode exhibits near-constant energy (~1,016 J, CV = 14%) across all nine configurations, making it the most geometrically robust steering choice.

On slope terrain, arc-path manoeuvres incur substantially higher energy (+38% to +98%) relative to flat-ground equivalents due to gravitational work along the arc path, while pivot-type skid-steering energy is reduced by up to 22% due to decreased wheel-ground normal force. A systematic rearward shift in axle load distribution is identified during arc manoeuvres on slope, with rear-axle drive motors carrying up to 2.7× the energy of front-axle motors - a mechanically significant finding for rear-axle motor sizing in slope-capable platforms. Bivariate regression models of the form $E = f(B, L)$ are derived for all six steering modes, achieving $R^2 = 0.70-0.98$ and providing practical engineering formulae for early-stage energy estimation from geometric parameters.

The platform, its instrumentation methodology, and the experimental dataset collectively constitute a validated experimental foundation for steering-related energy research in electric agricultural vehicles. The findings directly inform design guidelines for energy-efficient small-scale agricultural robots, with particular relevance to orchard and vineyard applications in which turning energy represents a critical fraction of total operational energy.

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1 Introduction

1.1 Overview

Global population growth continues to intensify demand for food production, placing unprecedented pressure on agricultural systems worldwide. As shown in Figure 1-1, according to the United Nations, the global population is projected to approach 10 billion by 2050 (United Nations, 2019). Global food demand is projected to increase by approximately 50 percent by 2050 relative to 2012 levels (Alexandratos & Bruinsma, 2012), placing intensifying pressure on agricultural systems to raise productivity per unit of land. This growth is accompanied by increasing expectations for food quality, supply reliability, and sustainability, requiring agricultural systems to improve productivity while simultaneously reducing environmental impact and energy consumption.

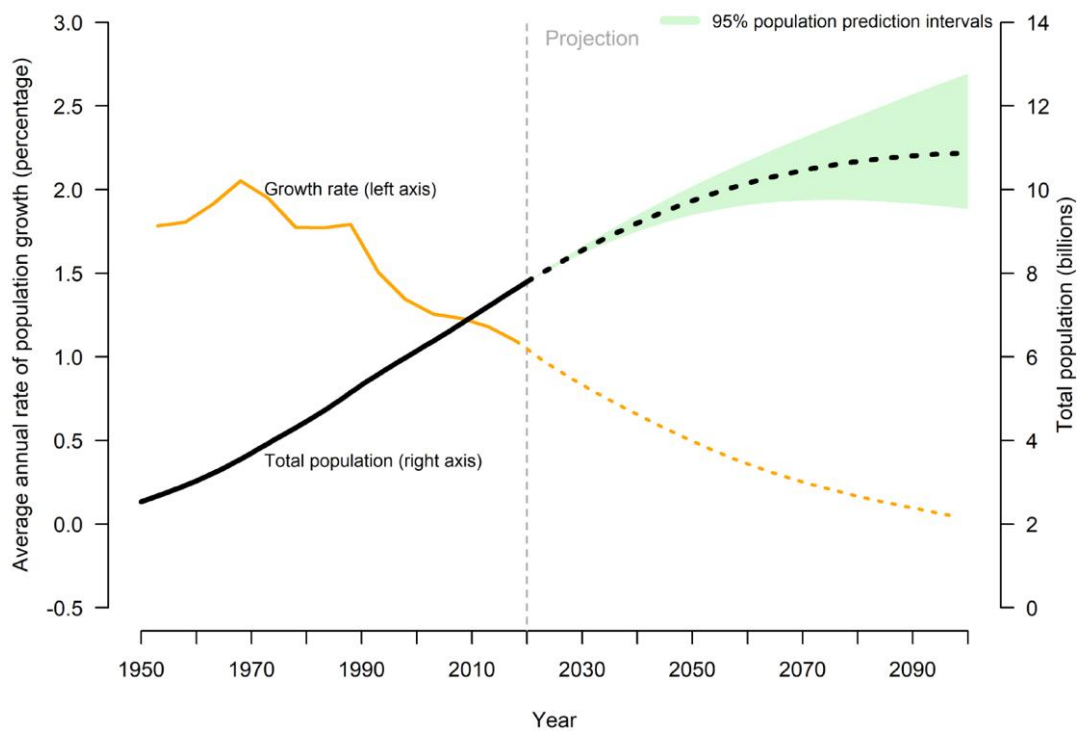


Figure 0-1 Global population growth projections toward 2050 (United Nations, 2019)

Expanding agricultural output through conventional means alone is increasingly constrained by limited arable land and growing pressure on soil health, water resources, and ecosystems. At the same time, agriculture is under scrutiny for its contribution to greenhouse gas emissions and fossil fuel consumption. Consequently, modern agricultural systems face a dual challenge: achieving higher productivity while reducing reliance on carbon-intensive energy sources.

Mechanisation has historically driven productivity improvements, with diesel-powered tractors and field vehicles playing a central role (Renius, 2020). While these machines have improved labour efficiency and output, they have also increased fuel consumption, emissions, and operational costs. These limitations motivate the exploration of alternative propulsion technologies that can maintain performance while improving energy efficiency and sustainability.

This challenge is particularly relevant in countries with strong agricultural sectors such as New Zealand, where orchard, vineyard, and pasture-based systems are widely used. These environments often require compact vehicles capable of frequent manoeuvring on grass-covered or uneven terrain. Under such operating conditions, vehicle energy efficiency directly affects operating range, productivity, and cost.

Electrification has emerged as a promising pathway to address these challenges. Electric agricultural vehicles offer high drivetrain efficiency, reduced emissions, and precise control of vehicle motion. Importantly, electrification enables new vehicle architectures and control strategies that are difficult to achieve with conventional drivetrains. As a result, research into electric agricultural mobility has expanded across drivetrain design, vehicle dynamics, and energy optimisation.

In this context, understanding and minimising vehicle energy consumption-particularly during low-speed manoeuvring and turning-has become a critical research objective. Agricultural vehicles operate under conditions that differ significantly from on-road vehicles, and their energy consumption is strongly influenced by steering strategy, vehicle geometry, and wheel-ground interaction. Addressing these factors requires experimental investigation under representative operating conditions.

1.2 Agricultural Growth Demand and Environmental Pressure

Sustainable growth in agricultural production requires fundamental changes in machinery technology and operational practice. Conventional agricultural vehicles powered by internal combustion engines consume large quantities of fossil fuel and generate emissions of carbon dioxide, nitrogen oxides, and particulate matter (Lajunen et al., 2018). These impacts, combined with noise and rising fuel costs, have increased pressure on the sector to adopt cleaner and more efficient technologies.

International climate commitments and national emissions targets further reinforce this transition. In New Zealand, sustainability goals and economic considerations have accelerated interest in low-emission agricultural machinery (Reguyal et al., 2023). From an operational perspective, fuel price volatility and long operating hours make energy efficiency a key determinant of machinery cost and viability.

Electric powertrains offer several advantages for agricultural applications, including higher efficiency, reduced maintenance, and compatibility with renewable energy sources (Mocera et al., 2023). These benefits are particularly attractive for orchard and vineyard vehicles, autonomous field robots, and other platforms with predictable duty cycles and relatively low operating speeds.

However, vehicle energy consumption is determined by more than the power source alone. In off-road environments, energy losses are strongly influenced by wheel-ground interaction, steering strategy, and vehicle geometry. Slip, soil deformation, and rolling resistance can dominate energy usage, particularly during turning manoeuvres (Wong, 2010; Bekker, 1969). Consequently, optimising electric agricultural vehicles requires an integrated approach that considers mobility strategy alongside drivetrain efficiency.

1.3 Electrification of Agricultural Machinery

The electrification of agricultural machinery has progressed from conceptual research to early commercial deployment, supported by advances in battery technology, electric motors, and power electronics (Troncon et al., 2021). Several manufacturers have introduced electric or hybrid-electric tractors and autonomous platforms, demonstrating the feasibility of electrification in real agricultural operations.

Electric drivetrains are well suited to agricultural tasks due to their ability to deliver high torque at low speeds (Renius, 2020; Lajunen et al., 2024). In addition, electric systems provide precise, rapid torque control, enabling advanced traction and motion control strategies. These capabilities can reduce excessive slip, improve manoeuvrability, and enhance energy efficiency on variable terrain (Kutzbach & Schreiber, 2016).

Electrification also enables new vehicle architectures. Distributed drive and modular layouts allow designers to decouple propulsion from traditional mechanical constraints (Nam et al., 2012), supporting improved packaging and control flexibility. In orchard and vineyard

environments, where narrow rows and frequent turning are common, such flexibility can significantly influence energy consumption and vehicle performance (Deng et al., 2025).

Despite these advantages, many current electric agricultural vehicles retain fixed geometric configurations and conventional steering architectures. While suitable for early deployment, these designs limit the ability to optimise energy efficiency across diverse operating conditions. Vehicle dynamics research indicates that geometry and steering strategy play a critical role in determining energy consumption, motivating systematic investigation in agricultural contexts (Jerrelind, 2021).

1.4 In-Wheel Motors and Steering-Related Energy Consumption

In-wheel motors have attracted increasing interest due to their potential to simplify drivetrain architecture and improve controllability. By integrating the motor within the wheel hub, mechanical transmission components such as gearboxes and differentials can be eliminated, reducing losses and enabling independent torque control at each wheel (Miller, 2010; Zhai et al., 2020).

Independent wheel torque control supports advanced steering and traction strategies, including differential steering and torque vectoring. For agricultural vehicles operating at low speeds with frequent turning, steering-related energy losses can represent a significant portion of total energy consumption. These losses arise from tyre slip, soil deformation, and increased rolling resistance, particularly on soft or grass-covered terrain (Wong, 2010; Farhadi et al., 2020).

Skid steering, commonly used in compact agricultural vehicles and mobile robots, offers mechanical simplicity and tight turning capability but often incurs high energy losses due to wheel slip and ground shear. Experimental and modelling studies have shown that turning manoeuvres in skid-steer vehicles significantly increase power demand (Morales et al., 2010; Mandow et al., 2007; Dogru & Marques, 2018). Alternative steering strategies, such as articulated or Ackermann steering, can reduce lateral slip and energy loss, but introduce additional mechanical complexity and design constraints (Renius, 2020; Vu et al., 2023).

Vehicle dynamics research demonstrates that wheelbase and track width influence turning kinematics, stability, and load transfer, all of which affect energy consumption (Gillespie, 1992; Jerrelind, 2021; Wong, 2001). While recent studies suggest that distributed drive systems can improve energy management through precise torque control (Deng et al., 2025), most

investigations rely on simulations or vehicles with fixed geometry, limiting experimental validation.

1.5 Existing Literature and Research Gaps

Extensive research has examined energy consumption in off-road and agricultural vehicles, encompassing vehicle dynamics, terramechanics, and control. Models describing wheel-ground interaction highlight the dominant role of soil deformation and slip in traction losses (Wong, 2010). Agricultural engineering studies further demonstrate that traction efficiency is a major determinant of vehicle energy performance (Kutzbach & Schreiber, 2016).

Steering strategy has been identified as a key factor influencing energy consumption. Skid-steer vehicles, while compact and manoeuvrable, exhibit high energy loss during turning due to wheel slip (Morales et al., 2010). Alternative steering strategies can reduce these losses but are often evaluated independently or on platforms with fixed geometry.

Despite the depth of modelling and simulation-based research, experimental validation remains limited. Most studies use vehicles with fixed wheelbase and track width, preventing systematic investigation of geometric effects. Moreover, few platforms allow direct comparison of multiple steering strategies under consistent conditions.

A critical research gap therefore exists: the lack of a physically existing experimental platform that combines adjustable wheelbase, adjustable track width, and switchable steering strategies, capable of operating under realistic agricultural terrain conditions. This gap limits the validation of theoretical models and the identification of optimal configurations for real-world applications.

1.6 Research Objective

The primary objective of this research is to design, construct, and experimentally evaluate a fully electric agricultural vehicle platform with adjustable wheelbase and track width, capable of measuring steering-related energy consumption under realistic operating conditions. "Specifically, the platform must support two independent categories of design variables: steering strategy and vehicle geometry. This distinction underpins the experimental design adopted throughout this research. "The platform will employ in-wheel motors to enable independent wheel torque control and support multiple steering strategies.

Through controlled experiments on grass-covered terrain representative of orchard and farm environments, the research aims to quantify how vehicle geometry and steering strategy influence energy consumption during straight-line motion, turning manoeuvres, and slope traversal. Experimental results will be compared with existing theoretical predictions to assess model validity and identify opportunities for refinement. Ultimately, the study seeks to identify configurations that minimise energy consumption while maintaining practical manoeuvrability for small-scale electric agricultural vehicles.

1.7 Research Approach

A structured engineering methodology is adopted, comprising literature review, system design, instrumentation, experimentation, and analysis. A modular electric vehicle with adjustable geometry was designed and constructed, incorporating in-wheel motors and multiple steering modes. The vehicle is instrumented to measure electrical power consumption, vehicle motion, steering input, and terrain inclination.

Controlled field experiments were conducted across a range of geometric configurations and steering strategies. Collected data are analysed to identify trends in energy consumption and compared with existing models. Findings are used to inform design recommendations for energy-efficient electric agricultural vehicles.

1.8 Contribution

This research contributes an experimentally validated platform for investigating steering-related energy consumption in electric agricultural vehicles. It provides comparative energy data across multiple steering strategies and geometric configurations under realistic terrain conditions, supports validation of existing models, and offers practical insights for improving energy efficiency in small-scale agricultural vehicles, particularly in orchard and pasture-based systems.

1.9 Thesis Overview

Chapter 2 presents a comprehensive review of the existing literature on mathematical energy modelling for wheeled agricultural vehicles, terramechanics-based energy loss frameworks, steering-related power consumption models, vehicle geometry effects, and distributed in-wheel motor systems. A survey of existing electric experimental platforms for agricultural and off-

road energy research is also provided, and the key research gap motivating this work is identified.

Chapter 3 describes the vehicle design methodology, beginning with the research motivation and design challenges arising from the requirement for adjustable geometry and switchable steering strategies. Four conceptual system architectures are developed and evaluated through a weighted decision matrix, leading to the selection of the final design concept. The structural architecture, load path analysis, and material selection strategy are also presented.

Chapter 4 presents the mechanical design of the experimental platform in detail, covering the drive-steering system design including tyre selection, drive motor sizing and validation, steering motor sizing and validation, L-arm structural design and finite element analysis, and adapter assembly design. The structural assembly design is then presented, including the wheel support beam, semi-monocoque chassis, reinforcement rail, electronics mounting plate, articulated suspension, and protective cover.

Chapter 5 describes the mechatronic design of the platform, encompassing the electrical system architecture, power supply circuit design, signal transmission and CAN bus communication, core electronic component selection, and the embedded control software. The vehicle kinematic models implemented for all six steering configurations are derived and presented.

Chapter 6 presents the experimental results and data analysis. Following an introduction to the experimental setup and terrain characterisation, individual energy analyses are provided for each of the nine geometric configurations tested on flat terrain and three configurations tested on the 18° grass slope. Cross-configuration analyses are then conducted, including energy trend analysis, Pearson correlation analysis, and bivariate regression modelling, for both terrain conditions. A comparative analysis of flat versus slope energy consumption is also presented.

Chapter 7 presents the conclusion of the thesis, summarising the principal findings, research contributions, and design recommendations arising from the experimental campaign, and outlines six directions for future research.

2 Literature Review

The purpose of this literature review is to examine how energy consumption in wheeled agricultural vehicles has been studied to date, and to identify gaps in current knowledge. The review has two aims: to establish the mathematical foundations explaining energy use during vehicle motion and turning, and to assess how far existing experimental platforms have validated these predictions under real agricultural conditions.

Section 2.1 reviews mathematical models of energy consumption, from longitudinal and terramechanics-based formulations to models of steering-induced power loss, geometry effects, and electric drivetrains, showing that steering strategy and geometric configuration are key determinants of energy efficiency off-road. Section 2.2 reviews existing electric test platforms, noting that most rely on fixed geometry and a single steering configuration, limiting their ability to validate these effects systematically. Section 2.3 synthesises this literature to identify a research gap-the lack of a reconfigurable platform for benchmarking multiple steering strategies and geometries under consistent outdoor conditions-while Section 2.4 outlines how the test vehicle developed in this research addresses that gap.

2.1 Mathematical Modelling of Energy Consumption in Wheeled Agricultural Vehicles

A substantial, multidisciplinary body of literature has focused on the mathematical modelling of energy consumption in wheeled vehicles, particularly in off-road, agricultural, and mobile robotic applications. These models aim to predict power demand as a function of vehicle motion, steering strategy, wheel-ground interaction, drivetrain characteristics, and terrain properties. Mathematical modelling plays a critical role in early-stage vehicle design, enabling comparison of alternative configurations and control strategies before costly prototyping and field testing.

Unlike on-road vehicles, agricultural vehicles typically operate at low speeds, perform frequent turning manoeuvres, and interact with deformable terrain such as grass, soil, or crop residue. These operating conditions introduce additional energy loss mechanisms-such as wheel slip, soil deformation, and steering-induced shear-that are not adequately captured by conventional road vehicle models. As a result, researchers have progressively extended classical vehicle

dynamics formulations by integrating terramechanics, steering kinematics, and electric drivetrain modelling.

2.1.1 Longitudinal Energy Consumption Models

Early modelling approaches focused on longitudinal vehicle motion, where energy consumption is expressed as the product of traction force and vehicle velocity. In classical vehicle dynamics, the total traction force is modelled as the sum of rolling resistance, gravitational resistance due to slope, aerodynamic drag, and inertial forces during acceleration (Gillespie, 1992; Wong, 2001). The instantaneous mechanical power demand is therefore given by Equation (2.1), as shown below:

$$P = F_{\text{traction}} \cdot v \quad (2-1)$$

where F_{traction} is the total resistive force and v is vehicle velocity.

For agricultural and off-road vehicles, aerodynamic drag is typically negligible due to low operating speeds. Instead, rolling resistance and terrain-induced losses dominate energy consumption. Terramechanics-based models incorporate soil deformation and sinkage effects, often using Bekker pressure-sinkage relationships and shear stress formulations to estimate rolling resistance and traction efficiency as functions of wheel load, soil cohesion, internal friction angle, and slip ratio (Wong, 2010).

These models have been widely applied to tractors and agricultural robots to estimate tractive efficiency and energy consumption under different soil conditions (Renius, 2020; Taghavifar & Mardani, 2017). More recent work integrates slip-dependent traction efficiency and drivetrain efficiency into longitudinal energy models, enabling prediction of electrical energy consumption for battery-powered vehicles (Lajunen et al., 2024). More recent work integrates slip-dependent traction efficiency and drivetrain efficiency into longitudinal energy models, enabling prediction of electrical energy consumption for battery-powered vehicles.

However, longitudinal models are fundamentally limited to straight-line motion or quasi-steady conditions. They do not capture the additional energy losses introduced during turning manoeuvres, which are particularly frequent in orchard and vineyard operations where vehicles must repeatedly reverse direction at row ends.

2.1.2 Terramechanics-Based Energy Loss Models

Terramechanics provides a foundational framework for modelling wheel-ground interaction in off-road vehicles. Wong (2010) demonstrated that soil deformation, sinkage, and shear displacement can account for a substantial proportion of total energy loss in agricultural vehicles. The original pressure-sinkage relationship underpinning these models was formulated by Bekker (1969), whose classical framework remains the standard reference for off-road vehicle terramechanics. Bekker-based pressure-sinkage models describe how normal stress distribution beneath a wheel leads to energy dissipation through soil compaction.

Energy loss due to terrain interaction is often modelled as the integral of shear stress over slip displacement. This formulation has been applied to both wheeled and tracked vehicles, showing that energy consumption increases rapidly with slip ratio, particularly on soft or grass-covered terrain (Shafaei et al., 2023). Rolling resistance on grass surfaces is significantly higher than on hard terrain and is strongly influenced by tyre load, inflation pressure, and contact patch geometry (Pettersen & Gooch, 2020; Farhadi et al., 2020). These insights are directly relevant to agricultural vehicles, which often operate under conditions where slip cannot be avoided.

Terramechanics-based energy models have been incorporated into simulation tools for predicting vehicle range and efficiency. However, accurate parameterisation of soil properties remains challenging due to spatial variability in moisture content, vegetation cover, and compaction. As a result, terramechanics models often require experimental calibration, limiting their predictive accuracy when applied in isolation.

2.1.3 Steering-Related Power Consumption Models

To address turning-related energy losses, researchers have developed mathematical models that explicitly incorporate steering kinematics and wheel slip. One of the most influential contributions is the simplified power consumption model proposed by Morales et al. (2010) for wheeled skid-steer vehicles. In this formulation, total power consumption is decomposed into rolling resistance power and steering-induced slip power, with the latter modelled as a function of lateral slip velocity and shear forces at the tyre-ground interface.

Morales et al. (2010) demonstrated that skid-steer vehicles experience substantial increases in power demand during turning, even on rigid terrain (Dogru & Marques, 2018; Yu et al., 2010). The model shows that turning losses scale with vehicle mass, wheel separation, turning radius,

and surface friction properties. Earlier work on tracked skid-steer robots similarly showed that turning resistance can dominate total energy consumption (Morales et al., 2009). On loose or compressible terrain such as soil and grass, these losses are further amplified by wheel sinkage and bulldozing effects (Fiset et al., 2023).

Mandow et al. (2007) further refined skid-steer kinematic models by incorporating experimentally measured slip ratios. Their work demonstrated that ideal no-slip kinematic assumptions significantly underestimate both motion error and energy consumption. Yi et al. (2009) formalised skid-steer kinematics using instantaneous centre of rotation analysis, building on earlier foundational work by Maclaurin (2011) who applied the Magic Formula tyre model to characterise skid-steering moment on deformable ground., providing a mathematically rigorous framework that has since been widely adopted in mobile robotics research.

These models collectively highlight that skid steering, while mechanically simple and highly manoeuvrable, incurs substantial energy penalties due to unavoidable slip and shear forces.

2.1.4 Energy Models for Alternative Steering Strategies

In contrast to skid steering, alternative steering strategies aim to reduce lateral slip by aligning wheel trajectories with the vehicle's turning radius. Ackermann steering, widely used in road vehicles, minimises tyre scrub during turning and is often assumed to be more energy-efficient on firm terrain (Gillespie, 1992; Vu et al., 2023). Recent energy-aware path planning studies confirm that Ackermann-type arc steering achieves substantially lower energy consumption than skid steering at equivalent turning radii (Effati et al., 2020).

Articulated or centre steering, commonly used in agricultural tractors, allows the vehicle to pivot about a central joint, reducing turning radius while maintaining rolling contact at the wheels. Renius (2020) reported that articulated steering can reduce soil disturbance and energy loss under certain operating conditions. Wong and Chiang (2001) provided a general analytical framework for skid-steering on firm ground, demonstrating that the turning resistance moment is strongly dependent on the ratio of track width to wheelbase.

Mathematical models comparing steering strategies often focus on kinematic differences and theoretical slip reduction. Each steering mode-skid steering, centre-based steering, front-wheel steering with fixed rear wheels, and four-wheel steering-exhibits distinct kinematic characteristics and wheel slip behaviour, resulting in different energy consumption profiles

during turning manoeuvres. Direct experimental comparison of these strategies is not feasible using conventional fixed-architecture vehicles, reinforcing the need for a reconfigurable platform. However, most comparisons are conducted analytically or through simulation, with limited experimental validation under identical vehicle mass, geometry, and terrain conditions. As a result, while models suggest that steering strategy strongly influences energy consumption, quantitative comparisons remain uncertain.

2.1.5 Vehicle Geometry Effects in Energy Models

Vehicle geometric parameters, particularly wheelbase and track width, play a significant role in steering kinematics, load transfer, and traction distribution. Classical vehicle dynamics shows that wheelbase influences turning radius and longitudinal load transfer, while track width affects lateral stability and normal force distribution (Gillespie, 1992).

Jerrelind (2021) demonstrated that geometric configuration affects both steady-state and transient energy consumption, even in on-road vehicles. These effects are amplified in off-road environments, where changes in normal load influence sinkage, rolling resistance, and slip (Wong, 2010). Li et al. (2023) specifically investigated how wheelbase and track width affect the turning energy consumption of wheeled vehicles through wheel-ground interaction modelling, confirming the geometric sensitivity predicted by classical theory.

Several modelling studies have investigated geometry effects parametrically, showing that changes in wheelbase-to-track-width ratio can alter skid-steer energy losses and turning resistance (Maclaurin, 2011; Pentzer et al., 2014). However, such studies typically assume fixed terrain properties and idealised kinematics, and are rarely validated experimentally on a single adjustable platform.

2.1.6 Distributed Drive and In-Wheel Motor Energy Models

The transition to electric drivetrains has enabled more detailed modelling of energy consumption at the wheel level. Distributed drive and in-wheel motor systems allow vehicle power consumption to be expressed as the sum of individual wheel motor outputs, incorporating motor efficiency maps, inverter losses, and torque distribution strategies (Miller, 2010; Nam et al., 2012). BLDC motors, which are widely used in in-wheel drive systems, offer high power density, precise controllability, and efficient torque delivery across a wide speed range (Mohanraj et al., 2022).

Recent work on electric agricultural vehicles integrates drivetrain efficiency and control-oriented models to predict energy consumption under variable operating conditions. Deng et al. (2025) proposed a real-time energy-efficient control strategy for a distributed-drive electric tractor, linking predicted operating speed and torque demand to energy consumption. Koca et al. (2025) incorporated traction efficiency and power conservation models into an optimisation framework for agricultural mobile robots.

While these models demonstrate the potential for energy optimisation through control, they typically assume fixed vehicle geometry and a single steering configuration. Pentzer et al. (2022) demonstrated that online power model estimation for skid-steer robots can significantly improve mission energy prediction accuracy, but noted that the approach remains limited to a single fixed vehicle geometry. Consequently, the interaction between distributed drive control, steering strategy, and adjustable geometry remains underexplored.

2.2 Existing Electric Test Platforms for Agricultural and Off-Road Energy Research

While mathematical models provide valuable insight into the mechanisms governing energy consumption in wheeled agricultural vehicles, experimental validation remains indispensable. Real-world agricultural environments introduce complexities-including heterogeneous terrain, vegetation, soil deformation, and transient operating conditions-that are difficult to capture accurately using purely analytical or simulation-based approaches. Consequently, a growing body of literature has focused on the development of physical electric or electrified test platforms to support agricultural vehicle research, energy measurement, and control validation.

However, although numerous experimental platforms have been reported, their mechanical architectures and research objectives vary widely. Most existing platforms prioritise autonomy, task execution, or robustness rather than systematic investigation of steering strategy and vehicle geometry effects on energy consumption. This section reviews representative electric test platforms reported in the literature and identifies their strengths and limitations in the context of steering- and geometry-related energy analysis.

2.2.1 Modular Agricultural Robot Platforms

One major category of experimental platforms comprises modular agricultural robots designed to operate across a range of field environments and tasks. The Thorvald II agricultural robotic system is a prominent example of this approach (Grimstad & From, 2017). A comprehensive

review of advances in agricultural robotics by Oliveira et al. (2021) confirmed that most modular platforms of this type prioritise versatility and task execution rather than energy characterisation. Thorvald II was developed as a flexible research platform capable of operating in orchards, greenhouses, and open fields, with modular hardware and software components that support different payloads and missions.

From an energy research perspective, Thorvald II demonstrates the feasibility of electric propulsion and autonomous operation under realistic agricultural conditions. Field trials have highlighted challenges related to terrain variability, traction, and operational efficiency. However, the platform employs a fixed chassis geometry and a predefined steering architecture. As a result, while it enables valuable application-level experimentation, it does not support controlled investigation of how changes in wheelbase, track width, or steering strategy influence energy consumption.

Similarly, the BoniRob platform has been widely cited as an autonomous field robot designed for plant phenotyping and precision agriculture (Ruckelshausen et al., 2009; Bangert et al., 2013). Wirkus et al. (2024) subsequently compared the soil interaction and driving characteristics of multiple agricultural and space robots, confirming that energy consumption profiles differ substantially between platforms even under nominally identical conditions. BoniRob provides robust outdoor operation and high-quality sensing, making it well suited for crop-level analysis. However, like Thorvald II, its mechanical design is fixed, and steering is typically implemented using a single configuration. Energy-related measurements reported using BoniRob therefore reflect the characteristics of a specific vehicle layout rather than providing generalisable insights into steering or geometry effects.

2.2.2 Electric Platforms Instrumented for Energy Measurement

A second group of platforms focuses explicitly on measuring electrical energy consumption during agricultural or off-road operation. These platforms are often lightweight electric vehicles or mobile robots equipped with current, voltage, and power sensors to log energy usage during field tasks.

Loukatos et al. (2024) presented a prototype autonomous electric cargo robot designed for agricultural field operations, with detailed analysis of power consumption under different task scenarios. Their results showed that energy usage varied significantly with terrain conditions and operational mode, underscoring the limitations of purely model-based predictions.

However, the vehicle geometry and steering system were fixed, and experiments did not explore alternative configurations.

Reiser et al. (2023) reported the development of an agricultural robotic platform optimised for high traction and low soil compaction. Gonzalez-de-Santos et al. (2017) similarly demonstrated instrumented autonomous spraying robots operating in orchard environments, providing energy and performance data under representative agricultural conditions. The platform was instrumented to measure traction forces and energy consumption during field operation. While the study provided valuable insight into the relationship between traction performance and energy efficiency, the mechanical design constrained the vehicle to a single geometry and steering approach. Consequently, the experimental results cannot be directly extrapolated to vehicles with different wheelbase or track-width ratios.

These studies demonstrate that physical testing is essential for understanding real energy consumption, but they also highlight a critical limitation: energy measurements are inherently tied to the specific platform on which they are obtained.

2.2.3 Platforms with Advanced or Multi-Mode Steering

A smaller subset of experimental platforms incorporates more advanced steering mechanisms, including multi-mode steering or omnidirectional capability. These designs are often motivated by the need for improved manoeuvrability in confined agricultural environments such as orchards and vineyards.

Cui et al. (2024) reported an agricultural field management robot platform capable of supporting multiple steering modes, including Ackermann steering and zero-radius turning. Yasuda (2023) further demonstrated an agricultural field robot with four-wheel steering capability, confirming the mechanical feasibility of independent four-wheel steering in precision agriculture applications. This work demonstrates that multi-steering architectures are technically feasible and can enhance operational flexibility. However, the primary focus of the study was on kinematic feasibility and task execution rather than systematic energy benchmarking. Moreover, geometric parameters such as wheelbase and track width were not varied independently during experimentation.

In the broader mobile robotics literature, omnidirectional and differential-drive platforms have been used extensively to study motion control and navigation (Bakker et al., 2011). The energy-efficient path planning studies of Gupta et al. (2017) and Effati et al. (2020) using simulated

differential-drive and skid-steer robots highlight that energy-optimal behaviour can differ substantially from kinematically feasible behaviour. While these platforms provide insight into steering mechanics and control, they typically operate on rigid indoor surfaces and at small scales. As a result, their energy consumption characteristics differ substantially from those of agricultural vehicles operating on grass or soil, limiting their relevance to full-scale agricultural vehicle design.

2.2.4 Structural and Experimental Limitations of Existing Platforms

Despite the diversity of reported electric test platforms, several common limitations emerge when evaluated from the perspective of steering- and geometry-related energy analysis.

First, most platforms employ fixed wheelbase and track width. While fixed geometry simplifies mechanical design and improves reliability, it prevents systematic experimental investigation of geometric effects predicted by mathematical models. Consequently, relationships between wheelbase-to-track-width ratio and energy consumption-particularly for skid steering-remain largely unvalidated experimentally.

Second, steering strategies are typically singular. Platforms are often designed exclusively for skid steering, Ackermann steering, or caster-based steering. This design choice prevents direct, controlled comparison of steering strategies under identical vehicle mass, drivetrain, and terrain conditions. Energy differences between steering modes must therefore be inferred from separate studies rather than measured directly.

Third, experimental campaigns are rarely factorial in nature. Even when platforms are instrumented for energy measurement, experiments typically evaluate a limited set of operating conditions without systematically varying geometry and steering parameters. This limits the usefulness of the resulting data for informing full-scale vehicle design.

Finally, scalability remains uncertain. Many reported platforms are small robots or task-specific vehicles. Without experimental datasets that explicitly link geometry and steering strategy to energy consumption, designers of full-scale agricultural vehicles must rely heavily on mathematical models or partial experimental evidence.

2.3 Synthesis of Literature and Identification of Research Gap

The reviewed literature reveals a clear dichotomy between theoretical modelling and experimental validation. On the one hand, a wide range of mathematical models has been

developed to analyse energy consumption in wheeled vehicles. These models address longitudinal motion, terramechanics, steering-induced slip, vehicle geometry, and electric drivetrain efficiency. Collectively, they demonstrate that steering strategy and geometric configuration can significantly influence energy consumption, particularly in off-road environments.

On the other hand, a growing number of electric experimental platforms have demonstrated the feasibility of field operation and energy measurement. These platforms confirm the importance of real-world testing but are generally constrained by fixed geometry and limited steering configurability.

As a result, a critical gap exists in the literature: there is no widely reported experimental platform that enables systematic, outdoor energy benchmarking across multiple steering strategies and adjustable geometric configurations within a single vehicle. This gap limits the ability to validate mathematical models and to inform design decisions for full-scale agricultural vehicles with confidence. The reviewed literature therefore highlights the need for a configurable experimental platform capable of systematically validating model predictions under realistic agricultural operating conditions.

The adjustable electric test vehicle developed in this research is designed to fulfil this need and to provide a robust experimental foundation for subsequent chapters.

2.4 Contribution of This Research Relative to Existing Literature

The research presented in this thesis directly addresses the identified gap through the development of a physically existing, adjustable-geometry electric test vehicle designed specifically for steering-related energy analysis. Unlike existing platforms, the proposed vehicle enables:

1. Independent adjustment of wheelbase and track width, allowing controlled investigation of geometric effects;
2. Implementation of multiple steering strategies, including skid steering, centre-based turning, and front-wheel (Ackermann-type) steering;
3. Outdoor operation on representative agricultural terrain, including grass-covered and sloped surfaces;

4. High-resolution energy measurement, enabling direct comparison with mathematical model predictions.

By supporting combinatorial experimentation across geometry and steering configurations, the platform provides experimentally validated data that bridge the gap between theory and practice. This approach enables more informed design decisions for full-scale electric agricultural vehicles and contributes to the development of energy-efficient, sustainable agricultural machinery.

3 System Overview

3.1 Specification of the machine

Table 3.1 outlines the specifications of the test vehicle.

Table 3.1 Specifications of the test vehicle

Parameter	Specification
Vehicle weight	63.9 kg
Maximum payload	150 kg
Maximum speed	15 km/h
Drive type	4-wheel independent in-wheel motors
Steering type	Independent steering motor at each wheel
Suspension type	Center-pivot oscillating axle
Braking type	Electronic braking
Emergency stop (E-STOP)	Yes
Remote control	Bluetooth
Steering configurability	Configurable to skid steer, 2WS (two-wheel steering), and 4WS (four-wheel steering)
Track width adjustability	400–800 mm, adjustable in increments of 50 mm
Wheelbase adjustability	400–800 mm, adjustable in increments of 50 mm

3.2 Innovation-Oriented Development Methodology

To manage the complexity inherent in the proposed experimental platform, an innovation-oriented development methodology was adopted. Design decisions were guided primarily by experimental requirements rather than by production-oriented optimisation criteria.

First, overall vehicle size and mass were constrained to ensure safe operation and experimental practicality. The total vehicle mass was limited to less than 100 kg, with overall length and width each constrained to approximately 1000 mm. This size and mass envelope is consistent with the category of compact agricultural robots reviewed by Oliveira et al. (2021) and Raji & Lal (2023).

Second, propulsion and steering functions were mechanically integrated into modular drive-steer units. Each module combines one traction motor and one steering motor into a compact sub-assembly. Four identical modules are mounted at the corners of the chassis, providing a symmetric and modular mechanical layout. This modular architecture is consistent with the drive-by-wire approach adopted in recent distributed-drive electric vehicle designs, in which each wheel is controlled independently to improve energy efficiency and traction (Nam et al., 2012; Zhai et al., 2020). This modularity simplifies assembly, enhances structural consistency, and supports repeatable experimental comparison across different configurations.

Third, the chassis was designed to allow independent adjustment of wheelbase and track width. Both parameters can be varied without modification of other subsystems, and arbitrary combinations within the design range can be realised. This capability is essential for isolating the effects of vehicle geometry on steering behaviour and energy consumption.

Fourth, the vehicle employs a fully distributed actuation and control architecture. Traction and steering motors are actuated independently and coordinated through a central control unit. This architecture enables precise motion coordination during driving and turning manoeuvres, supports the programmable implementation of multiple steering strategies, and ensures smooth and stable vehicle motion across different operating conditions.

Fifth, vehicle motion and steering mode switching are achieved via a Bluetooth-based remote-control interface (Naik & Srividya, 2021). An emergency stop function and signal-loss protection logic are implemented to ensure operational safety during experimental testing.

Finally, electrical current sensors are used to measure energy consumption during driving and steering. This direct electrical energy measurement approach - computing $E = \sum |I| \cdot V \cdot \Delta t$ from

sampled current and voltage - is consistent with the methodology employed in recent agricultural robot energy studies (Loukatos et al., 2024; Reiser et al., 2023). By recording motor current under controlled conditions, the platform enables quantitative comparison of energy usage across steering strategies and geometric configurations, forming the basis for subsequent experimental analysis.

This development methodology ensures that the vehicle functions as a dedicated experimental platform rather than a task-specific prototype, providing the configurability and measurement capability required to address the research objectives.

3.3 System Architecture and Conceptual Design

Based on the overall vehicle design objectives and functional requirements defined in the previous sections, four alternative system architectures were developed during the conceptual design stage. These concepts were formulated to explore different approaches to achieving adjustable wheelbase and track width while integrating distributed drive and steering functionality. Although all four concepts satisfy the core functional requirements, they differ significantly in structural layout, manufacturing approach, and mechanical performance.

3.3.1 General Concepts

Four overall concept designs were considered to achieve the requirements from Section 3.1, as pictured in Figure 3.1.

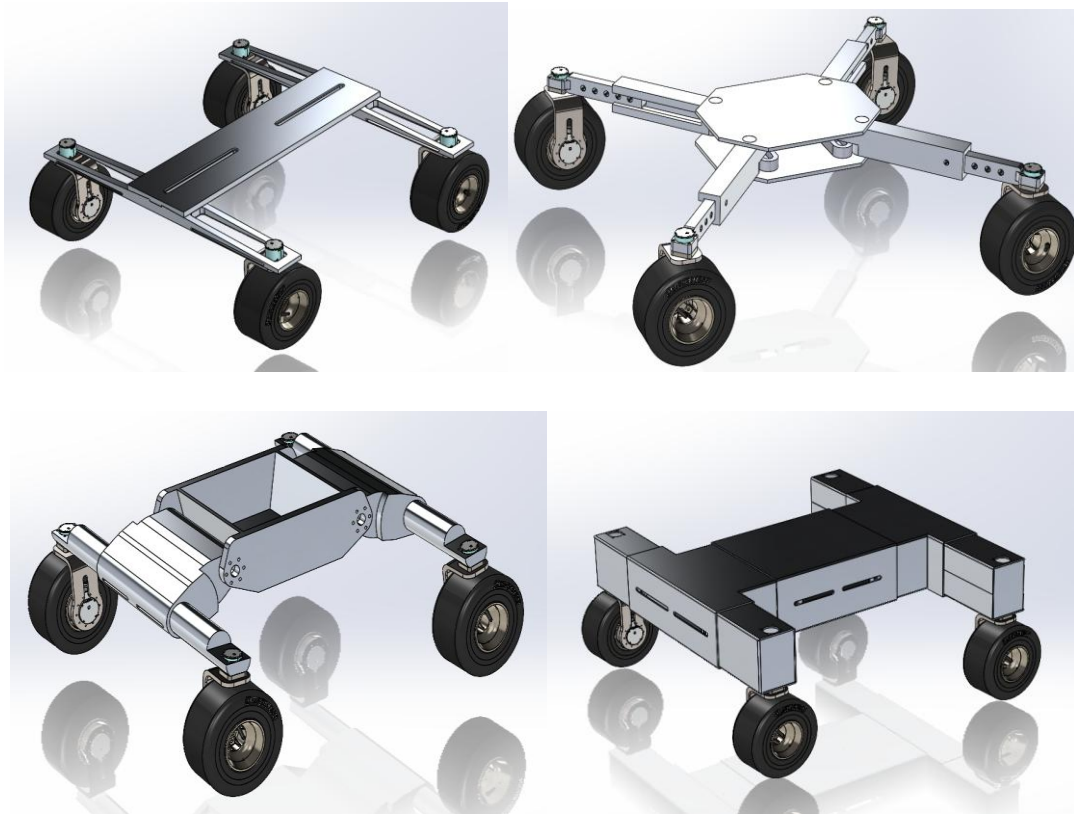


Figure 3-1 Structural layout and layered configuration of Concept 1-4

Four conceptual architectures were developed to achieve adjustable wheelbase and track width, as illustrated in Figure 3-1. Concept 1 adopts a vertically stacked layout, with drive-steer modules forming the lowest layer and front/rear axle structures and the chassis positioned above; geometric adjustment is achieved through sliding interfaces integrated into the axle structures, giving a mechanically simple, load-aligned structure with a high strength-to-weight ratio. Concept 2 instead uses commercially available telescopic linear rails, hinged to a bolted aluminium chassis via steel pins, allowing both linear extension and angular adjustment of the axles; this offers a large adjustment range and high configurational flexibility, but at the cost of increased component count, mechanical complexity, and reduced stiffness due to multiple sliding and rotational joints. Concept 3 achieves track width variation through axles sliding radially within rotating adjustable arms (for wheelbase adjustment), with a laser-cut and welded stainless-steel chassis that doubles as the vehicle body-eliminating a separate body structure and providing good overall rigidity-though at a higher cost due to CNC-machined aluminium axles and arms. Concept 4 similarly uses sliding axles within adjustable arms (track width) and arms sliding within the chassis (wheelbase), but fabricates all structural components from laser-cut, bent, and welded stainless-steel sheets, giving low manufacturing cost and a simple form

suitable for one-off prototyping, at the expense of potential distortion from sheet-metal welding and possible need for additional reinforcement at the hollow, thin-walled sliding interfaces.

3.3.2 Weighted Analysis

Table 3.2 Weighted decision matrix for the four concepts (scoring scale: 1=poor, 5=excellent)

Criterion	Weight	Concept 1	Concept 2	Concept 3	Concept 4
Structural strength	0.3	5	2	5	3
Adjustability flexibility	0.25	4	5	4	3
Mass (weight)	0.2	4	4	3	4
Cost	0.15	4	3	2	5
Manufacturing feasibility	0.1	4	3	2	4
Weighted score	—	4.3	3.4	3.6	3.6

As summarised in Table 3.2, a weighted decision matrix was used to compare the four concepts against structural strength, adjustability flexibility, mass, cost, and manufacturing feasibility. Concept 1 achieved the highest weighted score (4.3), outperforming Concepts 2, 3, and 4 (3.4, 3.6, and 3.6 respectively), driven primarily by its strong structural strength (5/5)-the most heavily weighted criterion-combined with balanced performance across the remaining factors, unlike Concept 2's poor structural strength, Concept 3's high cost, and Concept 4's reduced structural strength. Concept 1 was therefore selected as the final design concept, offering the most favourable overall balance of structural strength, adjustability, mass, cost, and manufacturability for the research platform.

3.4 Design Challenges and Structural Architecture

The mechanical design of the research vehicle is governed by a set of interrelated challenges arising from its intended operating environment, experimental objectives, and practical deployment requirements. Unlike production-oriented agricultural vehicles, the platform developed in this research is conceived as a configurable experimental system, requiring both mechanical robustness and high structural adaptability.

3.4.1 Design Challenges and Constraints

The primary mechanical challenges of the research vehicle can be categorised into four aspects: structural strength and durability, vehicle mass limitation, geometric adjustability, and cost control.

Structural strength and durability constitute the primary mechanical requirement of the vehicle. The platform is intended to operate predominantly in outdoor environments characterised by uneven grassland terrain, soil surfaces, and sloping ground conditions. During experimental testing, the vehicle is required to perform repeated straight-line motion and, in particular, a wide range of steering manoeuvres. As a result, the chassis and axle structures are subjected not only to longitudinal and lateral forces generated during steering, but also to impact loads and dynamic excitation arising from surface irregularities. Compared with vehicles operating exclusively on flat and rigid surfaces, the load-bearing requirements of this research vehicle are therefore significantly higher. In addition, considering the humid and frequently wet climate encountered in many regions of New Zealand, corrosion resistance and basic water protection must be incorporated into the mechanical design to ensure structural reliability and repeatability over extended outdoor testing periods.

A second major constraint is vehicle mass limitation. As a research and experimental platform, the vehicle is expected to be transported between different test locations, potentially including vineyards or orchard trellis environments. To facilitate handling, deployment, and storage, the overall vehicle mass is required to be as low as reasonably achievable. In particular, the design aims to allow manual handling by a small number of operators, ideally enabling two people to lift and reposition the vehicle when necessary. This requirement imposes strict limits on structural mass and directly constrains material selection, section sizing, and overall chassis architecture.

The requirement for adjustable wheelbase and track width represents a further and significant design challenge. While geometric adjustability is central to the experimental objectives of the vehicle, adjustable structures inherently introduce mechanical compromises. Compared with fully integrated and fixed-geometry designs, adjustable mechanisms typically exhibit reduced stiffness, increased local stress concentrations, and a greater number of discrete components. The increased part count not only raises structural complexity but also contributes to additional mass and more demanding manufacturing and assembly processes. Designing adjustable joints

and interfaces that maintain acceptable strength, alignment accuracy, and repeatability under varying load cases is therefore a key mechanical challenge.

Finally, cost control imposes an overarching constraint that is inversely related to the preceding requirements. High structural strength, low mass, and geometric adjustability generally demand more sophisticated materials, tighter manufacturing tolerances, or complex fabrication processes, all of which increase cost. As a one-off research prototype developed with limited resources, the vehicle design must carefully balance performance, robustness, and manufacturability against overall project cost. This trade-off strongly influences decisions regarding structural form, material choice, and the extent of mechanical integration throughout the vehicle.

Beyond the mechanical challenges described above, the mechatronic design of the platform introduced a distinct and equally demanding set of difficulties spanning control architecture, real-time coordination, and embedded software implementation.

A fundamental challenge concerned the integration of a consumer-grade wireless controller - the Sony PS4 DualShock 4 - with eight independent brushless DC motor controllers via a single ESP32 microcontroller. Establishing a stable and low-latency Bluetooth connection between the PS4 controller and the ESP32 required careful protocol implementation, and the CAN bus network connecting the ESP32 to all eight motor controllers had to be configured to handle simultaneous command transmission within the strict timing requirements of real-time vehicle control.

The most demanding control challenge arose from the need to govern sixteen independent actuator parameters - rotational speed and direction for each of the eight drive and steering motors - through a single handheld controller with a limited number of physical inputs. Each of the six implemented steering modes requires a distinct and mode-specific set of wheel speed and steering angle relationships. Furthermore, these relationships are not static: as the operator deflects the joystick continuously, all sixteen parameters must update simultaneously and dynamically in response, with the instantaneous values of each motor command determined by the real-time kinematic state of the vehicle. Mapping the analogue joystick inputs of a PS4 controller onto this sixteen-dimensional, continuously varying actuator space - while maintaining smooth, stable, and physically correct vehicle motion - represents a non-trivial real-time control problem.

The embedded software implementation presented a further layer of complexity. For each steering mode, the vehicle kinematic model - derived from classical vehicle dynamics and Ackermann-type geometry - had to be translated into closed-form expressions that could be evaluated in real time on the ESP32. This required the explicit derivation and programming of per-wheel speed and steering angle formulae for all six configurations, including both arc-path and zero-radius pivot variants. The resulting control programme must simultaneously handle Bluetooth input parsing, kinematic computation, CAN message packaging and transmission to eight motor controllers, and current data logging via the Arduino-based data acquisition system - all within the computational and timing constraints of a single embedded processor operating in an outdoor field environment.

3.4.2 Structural Architecture and Load Paths

To establish a clear mechanical rationale for the overall structural layout of the research vehicle, a free body diagram (FBD)-based analysis was first conducted. By abstracting the vehicle as a system composed of the chassis, axles, drive-steering units, and tyres, the dominant external forces and their transmission paths can be clearly identified. This approach provides a direct link between vehicle operating conditions and the adopted structural architecture.

Vertical Load Distribution

As shown in the Figure 3-2, in the vertical direction, the primary external load acting on the vehicle is the total gravitational force associated with the vehicle mass and onboard components. As illustrated in the free body diagram, this downward force acting on the chassis is balanced by the upward normal reaction forces generated at the tyre-ground interfaces.

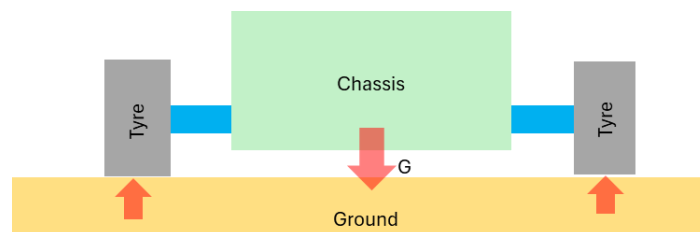


Figure 3-2. Free body diagram of chassis and tyres

The vertical load transfer path can be described as from the chassis to axle to tyre and finally to ground. In this sequence, the tyres serve as the sole interface between the vehicle and the terrain, transmitting ground reaction forces into the mechanical structure. These forces are then

transferred through the axles and ultimately distributed into the chassis. Consequently, the chassis must provide sufficient global bending stiffness to limit vertical deflection and maintain geometric integrity, particularly under uneven terrain conditions.

Unlike vehicles operating exclusively on rigid and level surfaces, the research vehicle is intended to traverse grassland, soil, and sloping outdoor environments. As a result, the vertical loads are not purely static but include dynamic components arising from surface irregularities and local impacts. This places additional demands on the axle-chassis interfaces and necessitates a structural architecture capable of accommodating both static weight and transient vertical excitation without excessive deformation.

Horizontal Load Conditions During Steering

As shown in the Figure 3-3, on the horizontal plane, the dominant external forces arise during steering manoeuvres, particularly under skid-steering conditions. During such manoeuvres, the tyres experience significant lateral interaction forces with the ground due to intentional slip between the tyre and terrain.

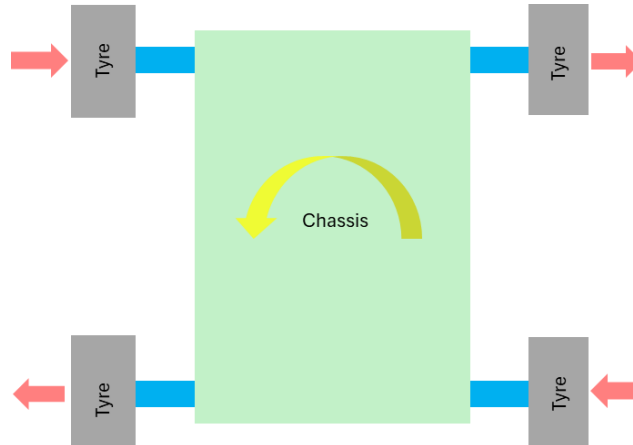


Figure 3-3. Skid steering moment of the vehicle

As shown in the horizontal-plane free body diagrams, the inner and outer tyres are subjected to lateral ground reaction forces that may differ in both magnitude and direction. These forces are transmitted from the tyres to the axles and subsequently to the chassis, generating substantial lateral loads and torsional moment within the vehicle structure.

Compared with conventional front-wheel steering, skid steering imposes two critical structural challenges. First, the magnitude of lateral tyre forces is generally higher due to increased slip.

Second, the asymmetric force distribution between the left and right sides introduces torsional loading that must be resisted by the axle and chassis assemblies. As a result, the load transfer path in the horizontal direction can be summarised as: ground → tyre → axle → chassis

with the axle playing a key role in distributing lateral loads before they reach the chassis. Based on the vertical and horizontal load analyses, a layered structural arrangement was adopted, in which the drive-steering units, axles, and chassis are arranged sequentially from bottom to top. This hierarchy reflects the natural direction of force transmission within the vehicle and ensures that each structural layer primarily carries the loads most relevant to its function.

Within this architecture, the tyres are mounted on axles that allow lateral sliding to enable track-width adjustment, while the axles themselves are designed to slide relative to the chassis to facilitate wheelbase adjustment. Importantly, this configurability is introduced without disrupting the fundamental load paths identified in the free body analysis.

By locating adjustable mechanisms at the axle and chassis interface, the majority of vertical loads remain compressive and well distributed, while lateral forces generated during steering are absorbed primarily within the axle layer before being transmitted into the chassis. This arrangement minimises stress concentration in the chassis and avoids exposing thin or lightly reinforced regions to excessive lateral or torsional loads.

3.4.3 Structural Layering and Load-Oriented Arrangement

Based on the preceding analysis of vertical and horizontal load conditions, the mechanical design of the research vehicle adopts direction-specific structural solutions to address the dominant loads acting on the system. By explicitly distinguishing between vertical load-bearing and horizontal load-resisting mechanisms at the architectural level, the proposed design achieves structural robustness while preserving geometric adjustability and practical manufacturability.

In the vertical direction, a stacked structural layout is adopted to support the vehicle weight and dynamic impact loads arising from uneven terrain. In this configuration, the two ends of each axle are supported directly by the corresponding drive-steering units beneath them, while the chassis is supported by the axles above, forming a continuous load path from the bottom to the top of the vehicle.

This layered, gravity-aligned arrangement ensures that the primary contact interfaces between major structural components remain predominantly under compressive loading. Consequently,

the vertical loads of the vehicle are borne primarily by the compressive strength of the structural materials and cross-sectional geometry, rather than by mechanical fasteners such as bolts or screws. This approach significantly reduces the risk of fastener fatigue, loosening, or failure caused by sustained gravitational loading or repeated vertical impacts, thereby enhancing the long-term structural reliability of the vehicle under outdoor operating conditions.

Structural Solution for Horizontal Loads

In the horizontal direction, the dominant loads are generated during steering manoeuvres, particularly under skid-steering conditions, where substantial lateral forces are transmitted from the ground through the tyres into the vehicle structure. To accommodate these loads, a groove-tongue (male-female) sliding rail interface is employed at the connection between the drive-steering units and the axles, as well as between the axles and the chassis.

In this interface design, one structural element incorporates a protruding feature, while the mating component contains a corresponding groove. This configuration allows relative sliding motion between adjacent units for geometric adjustment, while simultaneously enabling the interface geometry itself to carry horizontal shear and lateral loads. By transferring lateral forces through direct geometric contact rather than through fasteners, the structure maintains simplicity and robustness, and avoids concentrating horizontal or torsional loads in bolts, brackets, or thin connecting plates.

Material Selection and Durability Considerations

To ensure structural integrity, corrosion resistance, and mass efficiency, the vehicle structure is primarily constructed using 304 stainless steel and 6061-T6 aluminium alloy. The use of thin-sheet steel structures with laser-cut and welded construction is consistent with weight-optimised chassis approaches validated in prior lightweight vehicle work (Tsirogiannis et al., 2019). Stainless steel components provide excellent corrosion resistance and durability in humid and wet outdoor environments, while 6061-T6 aluminium alloy offers a high strength-to-weight ratio, contributing to overall mass reduction and improved portability of the research platform.

3.4.4 Mechatronic System Architecture and Control Solutions

The mechatronic design of the research vehicle is governed by a set of challenges that are as demanding as those of the mechanical structure. Where the mechanical solutions described in Section 3.4.3 address load transmission through geometry, the mechatronic solutions must

address the coordination of eight independent actuators through software, all operating under a unified real-time control architecture.

Power and Signal Architecture Solution

The electrical system adopts a three-tier voltage architecture to serve three functionally distinct subsystems: 48 V for the four drive motors, 24 V for the four steering motors, and 5 V for the control and data acquisition electronics. This layered approach isolates high-power drive circuits from sensitive control electronics, reducing electromagnetic interference and improving system reliability. Two 48 V-to-24 V converters are deployed in parallel to distribute the steering motor load, ensuring no single converter is operated near its rated limit. Emergency-stop switches are connected in series at both ends of the vehicle, guaranteeing that a single activation command immediately cuts power to all eight motors regardless of their current state.

The core control challenge - governing sixteen independent actuator parameters (rotational speed and direction for each of the eight drive and steering motors) through a single embedded processor - is resolved through a CAN bus network architecture. All eight motor controllers are connected to a single ESP32 microcontroller via CAN bus at 1 Mbit/s, with 120 Ω termination resistors at both ends of the bus in compliance with the ISO 11898 standard. The ESP32 sequentially broadcasts mode-specific command frames to each motor controller within each control cycle, achieving effective simultaneous coordination of all eight actuators. The use of CAN bus, rather than individual signal wires, dramatically reduces wiring complexity and improves noise immunity in the outdoor operating environment.

Kinematic Computation and Steering Mode Implementation



Figure 3-4 General arrangement for the test robotic



Figure 3-5 Orchard Rover Operating on Grass Terrain

The general arrangement and physical prototype of the test robot are shown in Figures 3-4 and 3-5, respectively.

4 Mechanical Design

This chapter presents the mechanical design of the adjustable-geometry experimental vehicle, focusing on the development, structural implementation, and engineering validation of its key subsystems. The mechanical system is required not only to satisfy fundamental vehicle functions under off-road operating conditions, but also to support the specific requirements of an experimental platform, including geometric adjustability, modularity, and measurement reliability. Therefore, the design approach extends beyond conventional vehicle engineering by incorporating structural optimisation and functional integration tailored to research objectives.

The chapter begins with the design of the drive-steering system, including tyre selection, drive motor sizing, and steering actuator configuration. This section addresses the requirements of propulsion and steering under different operating conditions, such as steady straight-line motion and skid-steering manoeuvres, with particular emphasis on load modelling and performance validation.

Subsequently, the design of critical structural components within the drive-steering modules—such as the L-arm and adapter assemblies—is presented. These components are analysed and optimised to ensure reliable load transmission and structural integrity under combined loading conditions, supported by finite element analysis and engineering verification.

Building on the subsystem design, the chapter then introduces the overall structural assembly of the vehicle, including the wheel support beam, the semi-monocoque chassis, and associated reinforcement structures. Particular attention is given to the implementation of dual-axis sliding mechanisms that enable independent adjustment of wheelbase and track width, while maintaining structural stiffness and well-defined load paths.

In addition, the suspension system is developed based on a centrally articulated configuration to improve terrain adaptability and maintain symmetric load distribution, thereby reducing experimental uncertainty. Finally, the design of protective and auxiliary structures, such as the protection cover and electronics mounting platform, is presented to enhance system usability, maintainability, and environmental robustness.

Through the integrated design of these subsystems, this chapter establishes a mechanically robust, functionally flexible, and experimentally suitable platform, providing a solid foundation for subsequent system integration and experimental investigation.

4.1 Drive-Steering System Design

The drive-steering system adopts an independent modular design, with each module integrating both drive and steering functions. Four modules are distributed at the four corners of the vehicle, operating independently while coordinating through a distributed control system to support energy consumption experiments across different steering strategies and geometric configurations.

From a structural standpoint, each module must simultaneously withstand combined loads in three directions: continuous vertical loading from vehicle weight; longitudinal forces from driving thrust and rolling resistance during straight-line motion; and lateral forces together with torsional moments from tyre-ground interaction during skid steering. The latter are particularly severe during point-turn manoeuvres, imposing stringent strength and stiffness requirements on the module's primary structure and necessitating finite element validation of critical components.

Furthermore, to accommodate the adjustable wheelbase and track width of the platform, each module must remain structurally compact, allowing smooth lateral translation along the axle and precise positioning at discrete locations. The interface between the module and the chassis must maintain sufficient load-bearing stiffness while enabling straightforward geometric adjustment, ensuring that the module envelope does not restrict the available range of configuration changes. A drive-steering system fulfilling the above requirements has been developed, with its overall structure shown in Figure 4-1.



Figure 4-1 Drive-steering system of the test vehicle

4.1.1 Tyre Selection

The tyre-ground contact patch geometry is a key determinant of rolling resistance and traction performance, influenced by tyre load, inflation pressure, diameter, and width (Bekker, 1969; Pacejka, 2012; Petterson & Gooch, 2020; Farhadi et al., 2020). Larger diameters generally reduce rolling resistance and improve obstacle-climbing capability, while wider profiles distribute load over a greater contact area, reducing ground pressure and soil sinkage (Wong, 2010). Since the experimental vehicle is intended primarily for off-road operation in orchard environments - where terrain includes grass, loose soil, and uneven ground - tyres must provide sufficient traction while minimising soil compaction and rolling resistance, making these the primary considerations governing tyre selection.

Increasing tyre radius reduces the required motor speed for a given vehicle velocity but increases the required torque. Tyre diameter must therefore be selected in conjunction with motor torque capability and gearbox ratio to ensure proper drivetrain performance.

Based on the overall vehicle dimensions (1 m × 1 m) and total mass of approximately 100 kg, a tyre diameter of 250 mm - approximately one-quarter of the vehicle length - was selected, balancing obstacle-crossing capability with drivetrain torque requirements. Assuming uniform load distribution, each wheel supports approximately 25 kg statically; tyres with a minimum load capacity of 75 kg per wheel (three times the static load) were specified to ensure adequate safety margin under dynamic conditions.

The selected tyres are pneumatic agricultural-type off-road tyres with the following specifications: diameter 250 mm, width 130 mm, inflation pressure 200 kPa, and maximum load capacity of 390 kg per tyre. Their wide tread profile increases ground contact area, improving traction and reducing soil sinkage, while the pneumatic construction provides passive shock absorption, offering an appropriate balance between load capacity, traction performance, rolling resistance, and terrain adaptability for this experimental platform.

4.1.2 Steering Motor Selection

To minimise steering torque, the output shaft centreline of the steering motor is required to intersect orthogonally with the wheel rotation axis, while simultaneously lying within the wheel mid-plane. According to the tyre-ground contact area, the maximum safe torque for each steering unit can be calculated.

The key design parameters for the steering and drive systems are listed in Table 4.1.

Table 4.1 Design parameters for drive and steering systems

Parameter		Specification
The vehicle mass	m (kg)	100
The vehicle weight	W (N)	981
The Vehicle Maximum Speed	v (m/s)	2.78
The vertical load applied to a steered tire	F_z (N)	245.25
Tyre Diameter	D (mm)	254
Tyre Thickness	B (mm)	114.3
Tyre pressure	P (kPa)	206.84
Tyre Design Parameter	c_t	1.5
Radial tire constant	n	4
Radial tire empirical coefficient 1	s	0.55
Radial tire empirical coefficient 2	t	122.7
Kingpin offset	c (mm)	0
Friction Coefficient	μ	0.69
Roll resistance	f_{RR}	0.1

4.1.2.1 Steering Torque Calculation

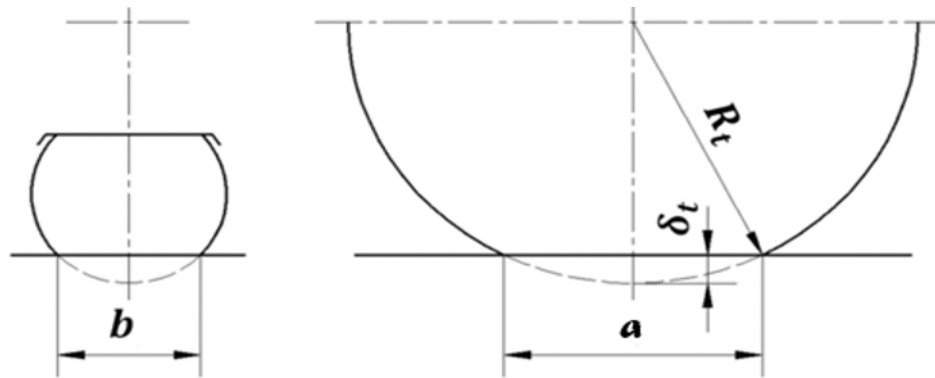


Figure 4-2 Tire-ground contact area

As shown in Figure 4-2, the tyre-ground contact patch geometry can be determined sequentially through the following series of equations (Cao et al.,2019).

$$k_t = 1.5 \times 10^{-3} B + 0.42 = 0.591 \quad (4.1)$$

$$\delta_t = \frac{c_t F_z^{0.85} k_t}{B^{0.7} D^{0.43} p^{0.6}} = 0.013\text{m} \quad (4.2)$$

The contact patch dimension can be calculated as follows (Cao et al.,2019):

$$a = 2D \left(\frac{\delta_t}{D} \right)^S = 0.097m \quad (4.3)$$

$$b = B(1 - e^{-t\delta_t}) = 0.091m \quad (4.4)$$

Accordingly, the pressure distribution within the tyre - road contact patch can be expressed as follows (Cao et al.,2019):

$$p(x, y) = \frac{n+1}{n} \frac{2^n F_z}{a^{n+1} b} \left[\left(\frac{a}{2} \right)^n - y^n \right] \quad (4.5)$$

The steering torque required for each tyre is therefore given by (Cao et al.,2019):

$$M_{f-s} = \iint dM_f = \int_{-\frac{b}{2}+c}^{\frac{b}{2}-c} \int_{-\frac{a}{2}}^{\frac{a}{2}} \mu p(x, y) \sqrt{x^2 + y^2} dy dx = 5.88Nm \quad (4.6)$$

The required motor mechanical power is calculated by dividing the product of the rated output torque and rated angular velocity by the mechanical efficiency of the gearbox.

$$P_m = \frac{\omega T_{output}}{\eta} \cdot SF = 13.84W \quad (4.7)$$

$$T_{output} = SF \cdot M_{f-s} = 8.81Nm \quad (4.8)$$

4.1.2.2 Motor Selection

Based on the calculated motor MG4010E-i36v3 output requirements, a brushless geared motor was selected. The specifications of the selected motor are listed below.

- Rated voltage: 24 V
- Rated current: 3.5 A
- Rated speed: 72 rpm
- Rated torque: 9 N.m
- Max torque: 18 N.m
- Gear ratio: 36
- Rated power: 84 W

The torque and power of the selected motor were verified against the calculated requirements, and the results are presented below.

$$\text{Torque margin} = \frac{T_{\text{rated}} - T_{\text{required}}}{T_{\text{required}}} = \frac{9 - 8.71}{8.71} = 3.33\% \quad (4.9)$$

$$\text{Power margin} = \frac{P_{\text{rated}} - P_{\text{required}}}{P_{\text{required}}} = \frac{84 - 13.68}{13.68} = 514\% \quad (4.10)$$

The selected motor satisfies the calculated torque and power requirements. The torque and rated power margins are 3.33%, 514%, respectively, confirming that the selected motor provides sufficient performance for the intended application.

4.1.3 Drive Motor Selection

Drive torque requirements are derived from two operating conditions: continuous output during rated straight-line driving on flat terrain, and peak output during skid-steering manoeuvres.

4.1.3.1 Drive Torque Calculation

For the continuous operating condition, the vehicle is assumed to travel continuously on a 5° incline to account for the resistance caused by the minor undulations typically encountered in off-road terrain. This results in resistive forces acting in two directions. The normal force is given by:

Rolling resistance is:

$$F_{RR} = f_{RR} F_N = f_{RR} G \cos \theta = 97.73N \quad (4.11)$$

The tangential force required to overcome the equivalent slope is:

$$F_T = G \sin \theta = 85.5N \quad (4.12)$$

Hence, the total driving force required is:

$$F_D = F_T + F_{RR} = 183.23N \quad (4.13)$$

After applying the safety factor f_s , the required output torque per wheel becomes:

$$T_{\text{output}} = f_s T_D = f_s \frac{F_D r}{4} = 8.59N \quad (4.14)$$

Using the equations 4.7, the drive motor rated power is 191 W.

For the peak output condition, corresponding to the skid-steering manoeuvre, the resistive moment generated by each wheel can be expressed as (Yang et al., 2020):

$$M_{R4} = \frac{\mu G}{6} \sqrt{a^2 + b^2} + \frac{\mu G a^2}{12b} \ln \frac{\sqrt{a^2 + b^2} + b}{a} + \frac{\mu G b^2}{12a} \ln \frac{\sqrt{a^2 + b^2} + a}{b} + \frac{1}{2} fGR + 2\mu G \frac{cd}{L} \quad (4.15)$$

$$= 287.16 \text{ Nm}$$

The driving force required per wheel is:

$$F_{D,each} = \frac{M_{Q4}}{(0.5B) \cdot 4} = 159.54 \text{ N} \quad (4.16)$$

4.1.3.2 Motor Selection

From formular 4.7 and 4.8, the peak torque is 29.91Nm and the peak power is 664.73W.

The selected MG8016E-i6v2 is validated against both operating conditions as follows:

- Rated torque: $T_{\text{rated, motor}} = 12 \text{ N.m}$
- Peak torque: $T_{\text{max, motor}} = 37 \text{ N.m}$
- Rated power: $P_{\text{rated, motor}} = 403.2 \text{ W}$
- Maximum power: $P_{\text{max, motor}} = 670 \text{ W}$

From formular 4.9 and 4.10, the torque and power of the selected motor were verified against the calculated requirements, and the results are presented below.

$$\text{Torque margin} = 24\%$$

$$\text{Power margin} = 1.2\%$$

The selected motor satisfies the calculated torque and power requirements. The torque and rated power margins are 24%, 1.2%, respectively, confirming that the selected motor provides sufficient performance for the intended skid steer application.

4.1.4 L-Arm Design

The L-arm is a critical structural component of the drive-steering module and represents one of the primary load-bearing elements of the vehicle. When assembled, the L-arm adopts an inverted L-shaped configuration relative to the vehicle body.



Figure 4-3 Assembled drive-steering system integrated into the test vehicle

As shown in Figure 4-3, the L-arm connects the steering motor adapter at its upper end and the drive motor and wheel assembly at its lower end, requiring it to withstand significant driving torque, impact loads from uneven terrain, and the inertial loads generated during steering manoeuvres. Given these demands, material selection with a high strength-to-weight ratio is critical.

Three options were evaluated following the systematic design methodology of Pahl et al. (2007) and Ulrich & Eppinger (2016): 3D-printed titanium alloy, offering the highest specific strength but at prohibitive cost for a one-off prototype; CNC-machined 6061-T6 aluminium alloy, providing good strength-to-weight ratio but with long lead times and complex machining; and laser-cut welded stainless steel plates, offering low cost and a short fabrication cycle, albeit limited to planar geometries.

Considering the balance between structural performance, manufacturing complexity, and cost, Option 3 - laser-cut welded stainless steel - was selected, with the key design challenge being how to exploit these fabrication techniques to realise a geometrically complex and structurally robust L-arm.

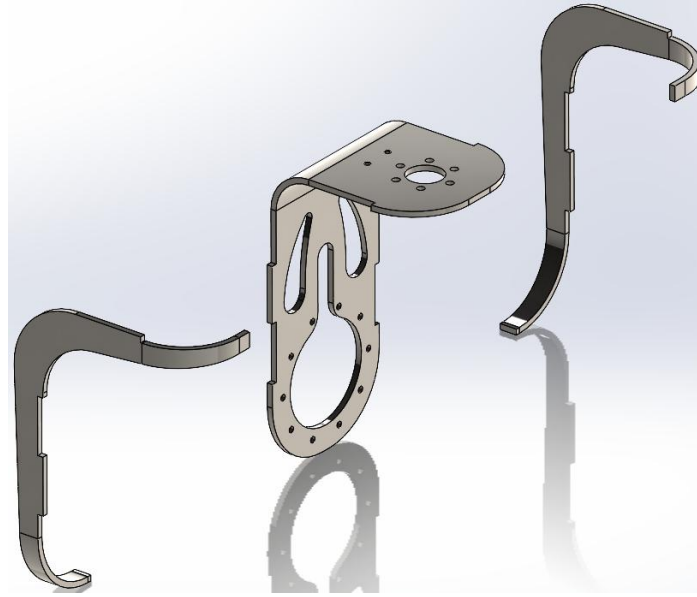


Figure 4-4 3D model of the L-arm before welding

As shown in Figure 4-4, the L-arm is fabricated from three laser-cut 4 mm thick 304 stainless steel plates, bent and welded together using interlocking protrusions and slots with a 0.2 mm clearance for pre-assembly alignment. This approach maximises dimensional accuracy while enabling mass optimisation.

The side plates provide vertical bending resistance to the central plate, with their width maximised near the L-shaped corner where stress concentration is highest and tapering toward both ends. To minimise bending moments on both motor output shafts, the regions closest to the wheel and steering motor are designed as planar surfaces. The resulting reinforced zones are arranged in an interleaved X-shaped configuration relative to the central plate, simultaneously satisfying the conflicting requirements of structural reinforcement and compact geometry. Weight-reduction cut-outs are distributed according to the predicted stress field, reducing steering inertia while providing cable routing clearance.

FEA was conducted under the most critical loading condition - a skid-steering point turn - incorporating three simultaneous load components: vertical loading from vehicle weight, lateral tyre-ground friction forces, and longitudinal traction forces from the in-wheel motor (Tsirogiannis et al., 2019; Dassault Systèmes, 2022).

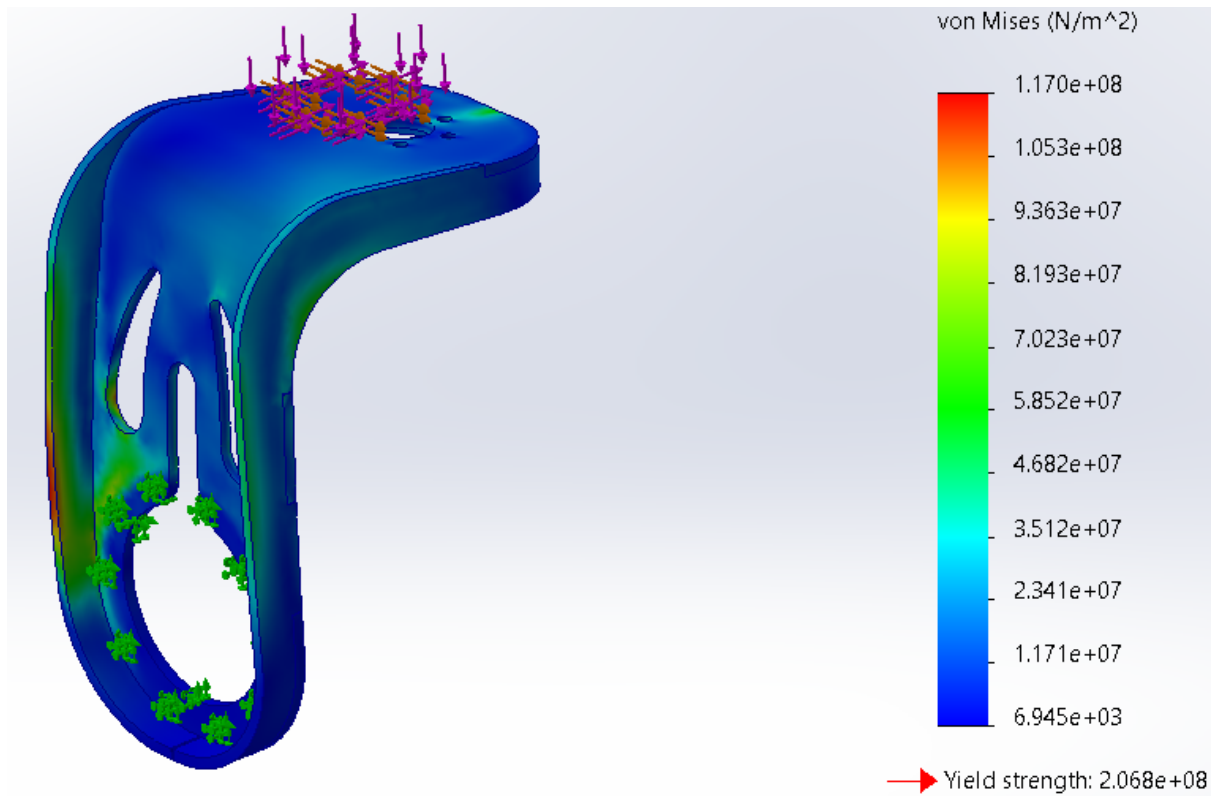


Figure 4-5 Finite element analysis (FEA) of the L-arm

As shown in Figure 4-5, FEA was performed under the combined loading condition of skid-steering point turn, with a safety factor of 1.5 applied for dynamic uncertainties. Vertical, lateral, and longitudinal loads were determined from vehicle weight distribution and tyre-ground interaction, and applied simultaneously in the model. The resulting maximum von Mises stress yields a safety factor of 1.77 against the yield strength of 206.8 MPa, confirming that the welded L-arm assembly satisfies the strength requirement with an adequate safety margin.

4.1.5 Adapter Assemblies Design

The adapter assemblies serve as essential structural transition components within the drive-steering module, resolving geometric incompatibilities between the commercially sourced motors and wheels and the custom-designed L-arm. Their interfaces provide precise alignment and structural rigidity to ensure stable operation. Beyond geometric compatibility, the adapters are designed to redirect lateral bending loads - generated by tyre-ground interaction during skid steering - away from the motor output shafts, which are primarily intended for torsional loading, and into the L-arm and chassis structure. This optimised load transfer path reduces combined torsion-bending stress on the motor shafts, improving fatigue life and overall structural reliability. Aerospace-grade aluminium alloy or stainless steel is used to balance strength,

stiffness, and corrosion resistance. An exploded view of the drive-steering assembly is shown in Figure 4-6.

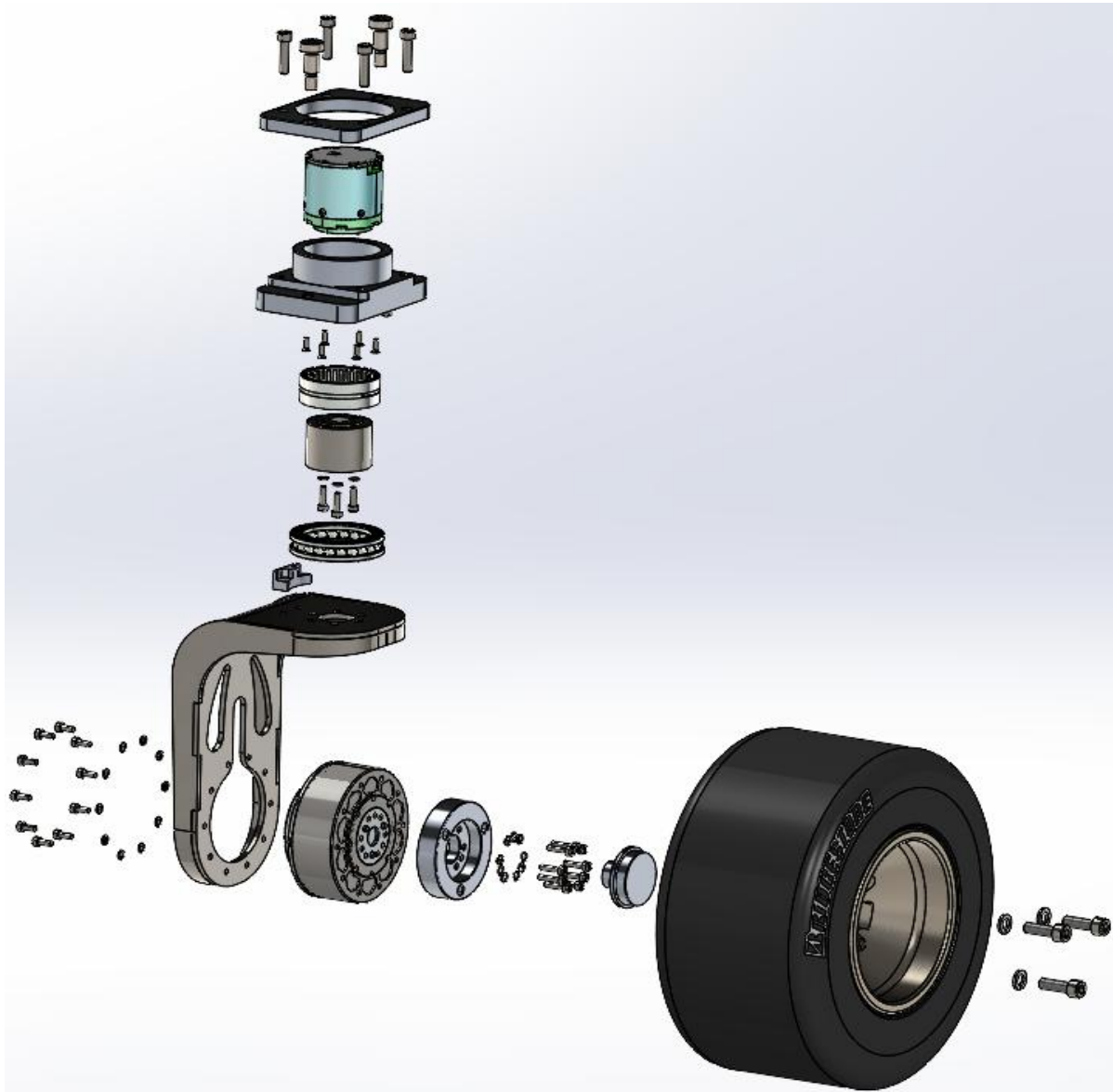


Figure 4-6 Exploded view of the drive-steering system

4.1.5.1 Adapter Assembly for the Drive Motor

The drive-side adapter primarily serves to ensure precise concentric alignment between the motor output shaft and the wheel hub. The motor housing is directly embedded into a pre-machined circular recess on the L-arm, eliminating relative translational and rotational degrees of freedom and minimising assembly-induced misalignment.

The critical structural concern on the drive side is the aluminium threaded interface at the motor output flange, which must resist the bending moment generated by the combined effect of vehicle weight and lateral tyre-ground friction during skid steering. Under the worst-case

loading scenario - maximum skid steering - the tyre-ground friction force reaches its peak value, and together with the gravitational load, generates a bending moment about the motor output shaft centreline that is ultimately resisted by the threaded bolt group. A proof load calculation was performed based on the free-body diagram of the flange bolt group, accounting for both the vertical load per wheel and the friction force at the tyre-ground interface. With the load shared across three bolts, the maximum bolt force was determined to be 783.99 N, which remains well below the typical proof load of 1980 N for M4 bolts, yielding a safety margin of approximately 2.5. This confirms that the threaded interface retains sufficient strength under the most critical loading condition considered in this study.

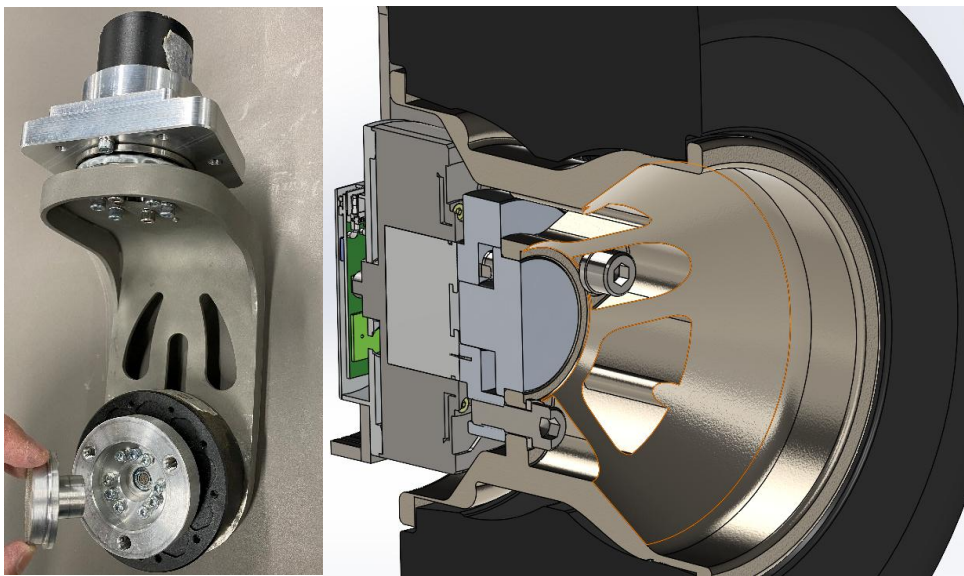


Figure 4-7 Exploded and sectional view of the drive-side adapter in the drive-steering system

As shown in Figure 4-7, two adapter components, CNC-machined from 6061-T6 aerospace-grade aluminium alloy, are installed between the motor output shaft and the wheel assembly. High dimensional precision and tight tolerances ensure accurate concentric alignment, preventing eccentric rotation and premature bearing wear. The adapters are designed to position approximately two-thirds of the motor body within the wheel hub cavity, minimising radial dimension and structural overhang, and reducing the bending moment acting on the motor output shaft.

Spring washers are installed between all bolt heads and fastening surfaces to prevent self-loosening under continuous vibration and cyclic loading (Bickford, 2008). Field testing confirmed their necessity - without this measure, noticeable bolt relaxation occurred after repeated operation over uneven terrain.

4.1.5.2 Adapter Assembly for the Steering Motor

Unlike the drive side, the steering motor output flange uses only three M3 threaded holes machined into aluminium, resulting in a significantly lower structural safety margin.

Verification calculations confirm that under maximum skid-steering lateral loads, the bolt force per screw reaches 2045 N, exceeding the proof load limit of 1130 N - meaning the original threaded interface alone is insufficient to resist the induced bending moment.

To address this, a dedicated adapter assembly is introduced to redistribute lateral bending loads away from the motor output shaft, allowing it to transmit primarily steering torque. The adapter incorporates a needle roller bearing of minimal radial envelope to accommodate the relative rotation between the drive-steering unit and the chassis, while maintaining a compact overall profile. The load path is thereby redirected through the adapter structure rather than the vulnerable threaded interface, effectively protecting the motor output shaft from combined torsion and bending.

To satisfy the conflicting demands of high strength within a highly constrained space and the need for sliding adjustability along the axle for track-width reconfiguration, both aerospace-grade aluminium alloy and stainless steel are employed in the adapter assembly. This material combination achieves the necessary structural rigidity and load capacity while keeping the assembly compact and lightweight.

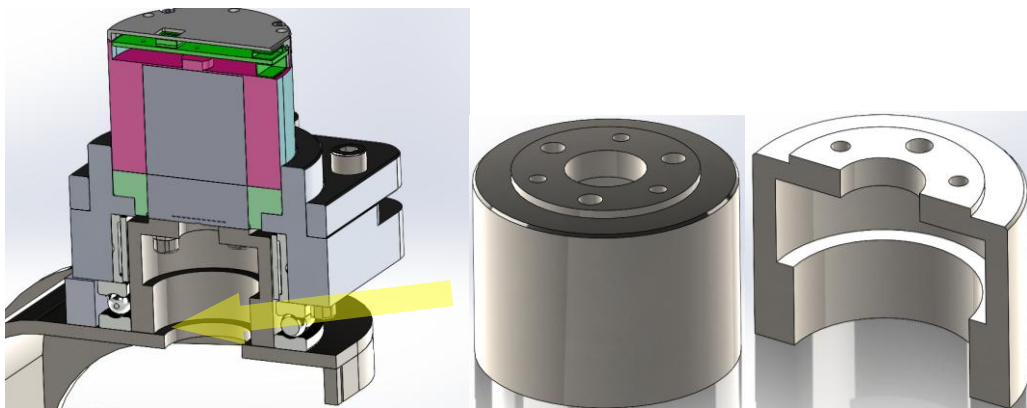


Figure 4-8 Adapter for transmitting steering motor bending moment

As shown in Figure 4-8, a hollow 304 stainless steel shaft is introduced as the key load-bearing element of the steering-side adapter. One end connects to the steering motor output shaft via a fastening screw through the central cavity and a dowel pin for precise radial positioning, while the other end is secured into a pre-machined bore in the L-arm to transmit steering torque. The

hollow shaft is radially supported by a needle roller bearing within the adapter, while a thrust ball bearing between the adapter and L-arm carries the vertical load from vehicle weight. This separation of radial and axial load paths (SKF Group, 2018) prevents gravitational forces from acting on the motor output shaft, and the hollow shaft itself resists the bending moments generated during skid steering, thereby significantly reducing the load transmitted to the vulnerable threaded interface.

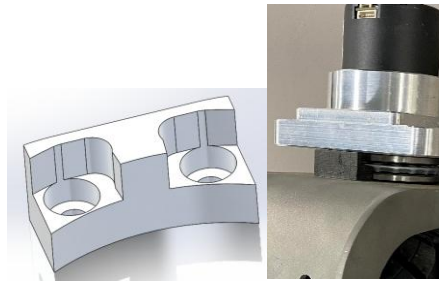


Figure 4-9 Limit stop

To prevent excessive steering rotation, a set of mechanical limit stops is installed between the L-arm and the flange adaptor as shown in Figure 4-9. These limit mechanisms restrict the steering angle to within $\pm 180^\circ$. By physically constraining the rotational range, the system prevents excessive torsional strain on the steering motor and associated electrical wiring. This protective measure enhances operational safety and improves the long-term reliability of the steering assembly.

4.2 Structural Assembly Design

Following the drive-steering system design, this section presents the structural assembly, which provides the primary load-bearing framework and geometric adjustment mechanism of the platform. The structural assembly consists of three principal components arranged from bottom to top: the wheel support beam (WSB), the semi-monocoque chassis (comprising the body shell and reinforcement rails), and the protective sliding cover.

The WSB serves as the transverse structural element connecting the drive-steering assemblies, with each assembly capable of sliding laterally along the beam for track-width adjustment. The WSB itself slides longitudinally along the reinforcement rails of the semi-monocoque chassis, enabling wheelbase adjustment. These two independent orthogonal sliding mechanisms provide two-dimensional geometric configurability while preserving clear load paths. The semi-monocoque chassis acts as the primary structural carrier, with the thin body shell contributing to global stiffness and the thicker reinforcement rails carrying concentrated loads.

The protective sliding cover mounted on top shields internal electronics and maintains accessibility for inspection and maintenance.

4.2.1 Wheel Support Beam Design

The WSB serves as the primary transverse load-bearing member, fabricated from laser-cut 304 stainless steel plates assembled through precision welding, providing a yield strength of 206.8 MPa and corrosion resistance suitable for outdoor agricultural environments (ASM International, 2008).

Structurally, the WSB is subjected to two dominant bending moment systems. Vertically, it behaves analogously to a simply supported beam under central loading, with the SMC applying load at mid-span and the DSAs providing end support, resulting in significant mid-span bending stresses. Horizontally, lateral tyre-ground interaction forces during skid steering generate additional bending and shear stresses. The beam must therefore provide sufficient stiffness and strength in both planes.

Beyond its load-bearing role, the WSB integrates two independent orthogonal sliding mechanisms within a single welded structure: transverse sliding rails for track-width adjustment and longitudinal sliding-slot engagement with the SMC reinforcement rails for wheelbase adjustment.



Figure 4-10 Sliding mechanism of the drive-steering assembly within the wheel support beam

As shown in Figure 4-10, the WSB must integrate two orthogonal sliding interfaces with positioning and locking functionality, requiring a dimensional tolerance of ± 0.1 mm. This presents a considerable challenge for a welded plate assembly, as welding-induced thermal distortion can compromise dimensional accuracy. To resolve this, the WSB is constructed from three laser-cut 6 mm thick 304 stainless steel plates - one bottom plate and two side plates -

whose geometric design and assembly strategy are critical to achieving both structural integrity and dimensional precision.

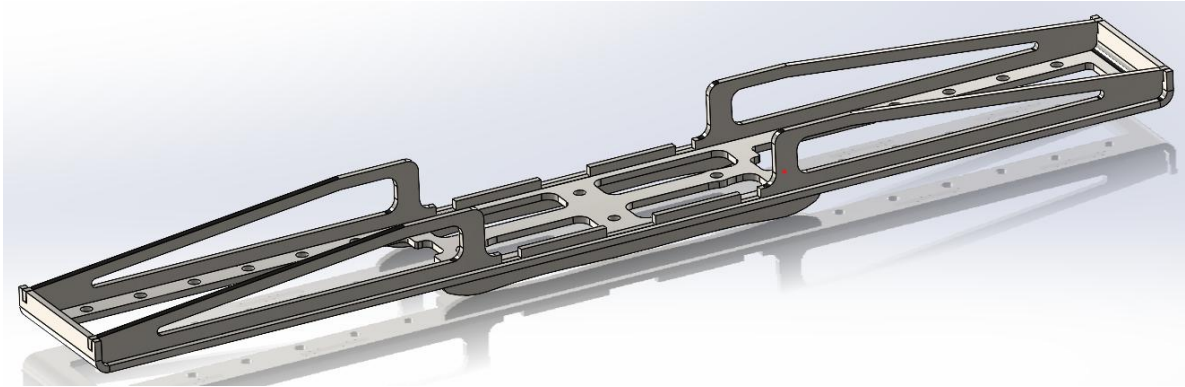


Figure 4-11 3D model of wheel support beam

As shown in Figure 4-11, The bottom plate adopts a "U"-shaped configuration with a flat central region and vertically bent ends. Rectangular slots along its sides serve as transverse sliding rails for the DSAs' lateral track-width adjustment. Elongated slots at the front and rear interlock with tongue-like protrusions on the side plates via a 0.1 - 0.2 mm controlled clearance, acting as a self-jigging mechanism that preserves key geometric dimensions post-welding. The bent ends also form partially closed sections that increase torsional rigidity and vertical bending resistance. High-precision laser-cut circular holes enable dowel-pin positioning of the drive-steering assemblies and chassis alignment.

The side plates are flat stainless-steel plates with a central downward recess and three rectangular slots that engage with the SMC's reinforcement rails for longitudinal wheelbase adjustment. Their variable width - narrower at the ends, wider at centre - reflects the bending moment distribution. A central hollow section reduces weight without compromising integrity, as the 8 mm reinforcement rails provide strong lateral support. The downward extension at the bottom centre increases effective section depth, enhancing vertical bending resistance.

Both components incorporate strategically distributed weight-reduction slots aligned with internal stress patterns: low-stress regions are selectively relieved while high-stress zones remain reinforced, maintaining uniform stress distribution.

FEA was conducted under the critical skid-steering point-turn condition. The WSB experiences three load components: vertical load from the chassis via the L-arm, lateral frictional forces from tire-ground interaction, and opposing longitudinal traction forces at both ends that

generate the yaw moment for point-turn rotation. This combined bending-torsion loading governs the structural design requirements.

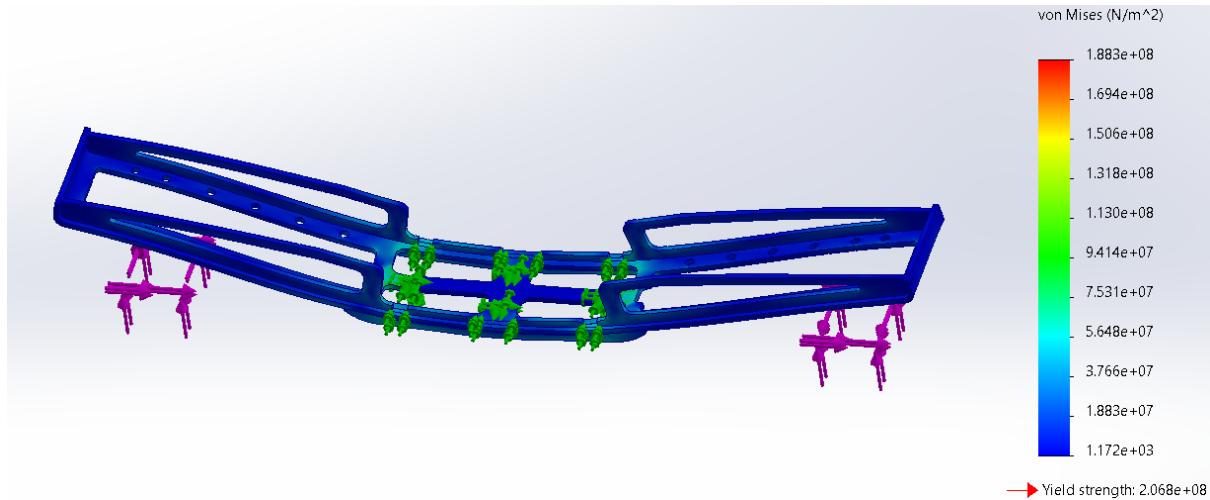


Figure 4-12 FEA for wheel support beam assembly

As shown in Figure 4-12, FEA was performed under the combined loading condition of skid-steering point turn with a safety factor of 1.5. Vertical, lateral, and longitudinal loads were determined from vehicle weight distribution and tyre-ground interaction, with lateral and longitudinal loads applied as equal and opposite force pairs at both ends of the WSB. The maximum von Mises stress remains below the yield strength of 206.8 MPa, confirming that the WSB satisfies the strength requirement with an adequate safety margin.

4.2.2 Semi-Monocoque Chassis Design

The semi-monocoque chassis (SMC) is the primary load-bearing structure of the experimental vehicle, as shown in Figure. The semi-monocoque concept, where the outer skin contributes to load-bearing through membrane stress, delivers an excellent strength-to-weight ratio widely adopted in aerospace, automotive, and ground vehicle applications (Tsirogiannis et al., 2019). It comprises the body shell, reinforcement rails, slewing bearing, electronics mounting plate, antenna mounting plate, and associated fasteners, rigidly bolted together to combine structural strength with geometric adjustability.

As shown in Figure 4-13, the chassis is divided into two nearly identical front and rear halves connected at the vehicle centre by a heavy-duty slewing bearing, providing structural continuity and controlled rotational freedom. This articulated connection enables the structure to function

as an integrated passive suspension system, allowing the vehicle to adapt to uneven terrain and maintain continuous ground contact without a conventional suspension mechanism.

Nearly all onboard electronics are housed within the SMC's internal volume, with layout carefully balanced between both halves to keep the centre of mass near the vehicle's geometric centre. This ensures approximately equal vertical load across all four wheels, eliminating mass-offset as a confounding variable in subsequent energy analysis.

The detailed design and functional roles of each SMC component are described in the following sections.

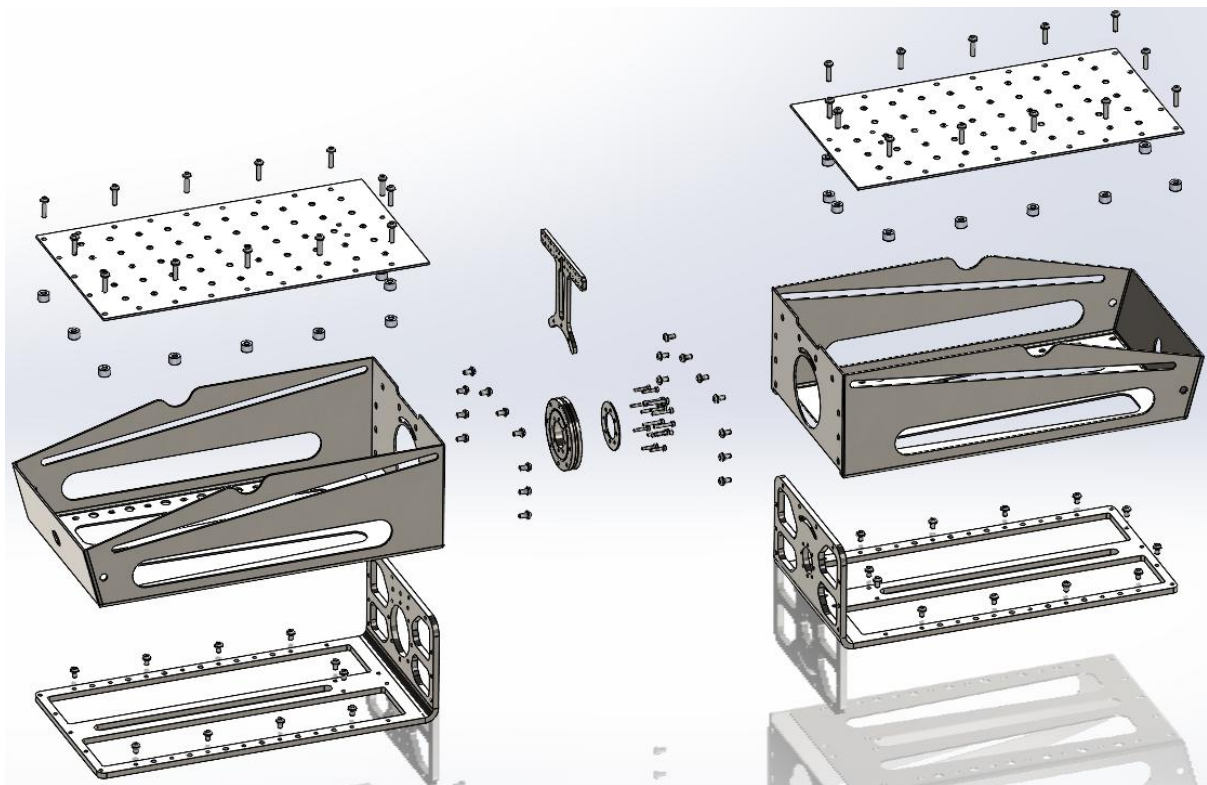


Figure 4-13 Exploded view for semi-monocoque chassis assembly

4.2.2.1 Body Shell Design

The body shell is fabricated from a single sheet of 2.5 mm thick 304 stainless steel, laser cut to shape, press-brake formed, and welded into a closed box-section component. Two identical units produce a fully symmetric chassis with no defined front or rear, simplifying manufacturing and part interchangeability.

The body shell exhibits increasing structural height from the outer end toward the slewing bearing interface, reflecting cantilever beam behaviour: bending moment increases toward the fixed support, so greater section depth at the inner region improves the second moment of area

and reduces bending stress. Weight-reduction slots follow the same principle - larger cut-outs at the low-stress outer end, more material retained near the high-stress inner region.

Two pairs of symmetric elongated slots are incorporated on both sides. The upper slots serve as sliding tracks for the protective cover; the lower slots provide cable routing paths that accommodate relative movement during wheelbase adjustment, allowing cables to slide freely without tension or damage.

4.2.2.2 Reinforcement Rail Design

The reinforcement rail has an L-shaped cross-section, fabricated from 8 mm thick 304 stainless steel via laser cutting and precision bending. As one of the primary structural members of the SMC, it integrates structural load transfer, geometric adjustment functionality, and subsystem mounting interfaces into a single component. It performs three major functions: providing the sliding guide for wheelbase adjustment, reinforcing the body shell, and acting as a mounting platform for key functional modules.

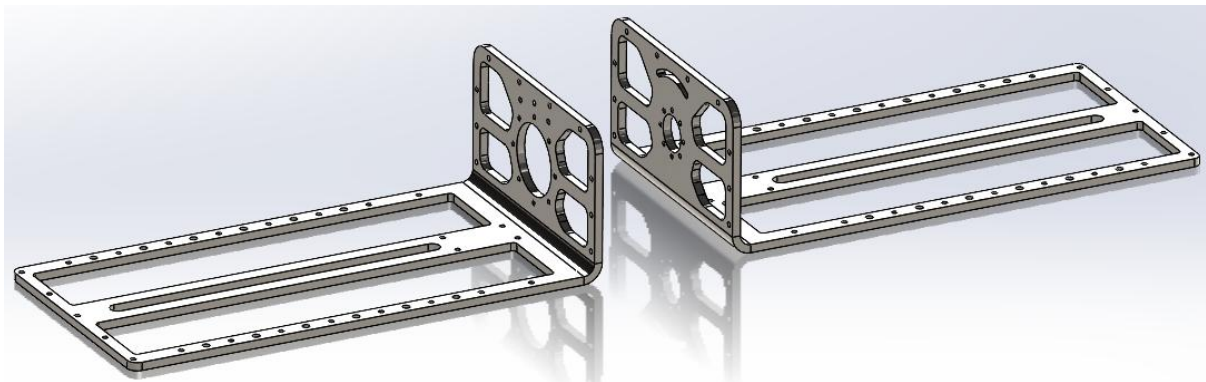


Figure 4-14 3D model of reinforcement rail

As illustrated in Figure 4-14, the reinforcement rail serves as the sliding guide for wheelbase adjustment. The WSB translates longitudinally along the rail, which provides lateral constraint to prevent transverse looseness or misalignment. Sliding blocks on the WSB engage with the rail, preventing vertical separation while allowing controlled longitudinal motion. Once the desired wheelbase position is achieved, two shoulder bolts are tightened into threaded holes on the rail, creating a rigid mechanical interlock that ensures precise positioning under vibration and dynamic loading.

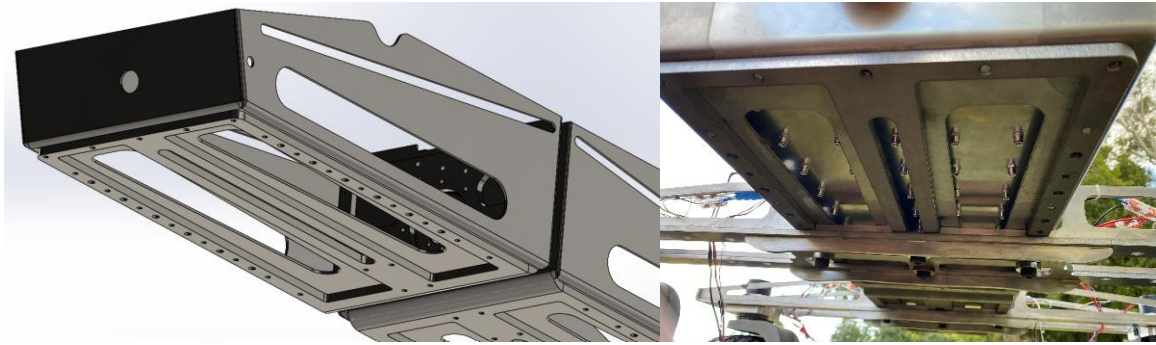


Figure 4-15 Rail system on the bottom of the chassis

As shown in Figure 4-15, the reinforcement rail provides structural reinforcement for the body shell. Although the welded box-section shell has good overall stiffness, its relatively thin sheet construction is susceptible to local deformation under concentrated loads. The reinforcement rail, with its greater thickness and L-shaped geometry, increases stiffness in both vertical and longitudinal directions. Multiple threaded holes along both surfaces allow distributed bolted fastening to the body shell, improving load transfer, reducing stress concentration, and enhancing overall chassis rigidity.

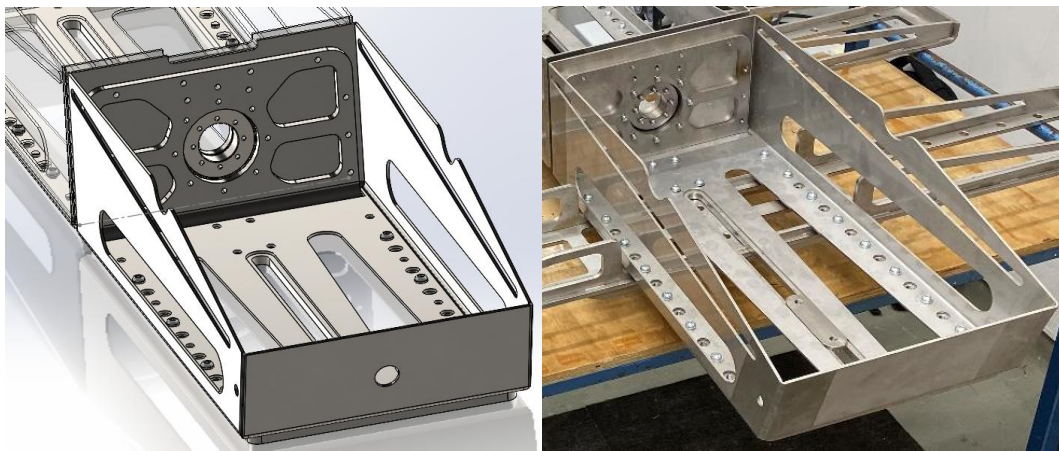


Figure 4-16 Perspective view of the reinforcement rail matched with the body shell

As shown in Figure 4-16, the reinforcement rail serves as the primary mounting platform for several critical subsystems. It provides the main mounting interface for the slewing bearing, which transmits loads between the front and rear chassis halves and experiences the highest local tensile and bending loads in the structure - necessitating the 8 mm thick stainless steel. Additional threaded holes on the horizontal surface mount the electronics mounting plate, while the vertical surface accommodates the antenna bracket and auxiliary equipment. A curved slot

limits relative rotation between the two chassis halves to approximately 30°, preventing excessive deformation and protecting internal wiring.

FEA-guided optimisation introduced weight-reduction cut-outs in low-stress regions while retaining material in critical load-bearing areas, improving the strength-to-weight ratio without compromising structural integrity.

4.2.2.3 FEA of the Semi-Monocoque Chassis

To verify the structural integrity of the semi-monocoque chassis, finite element analysis (FEA) was conducted under representative loading conditions. Considering that during skid-steering point turns the wheels on each side rotate in the same direction and at the same speed, the front and rear chassis are treated as a single rigid body. In this condition, the moment of inertia about the horizontal plane is relatively large. In comparison, the deformation effects caused by motor driving forces and tire-ground friction are negligible and therefore not considered. As shown in Figure 4-17, the FEA will primarily focus on the vertical loading due to gravity, applied along the weakest axis of the section's moment of inertia. A vertical load of 1500 N was applied to the chassis structure, corresponding to the estimated maximum operational load with a safety factor of 1.5 applied to account for dynamic loading and uneven terrain conditions. The simulation results indicate that the maximum von Mises stress within the structure is 37.02 MPa, which is significantly lower than the yield strength of 304 stainless steel (206.8 MPa).

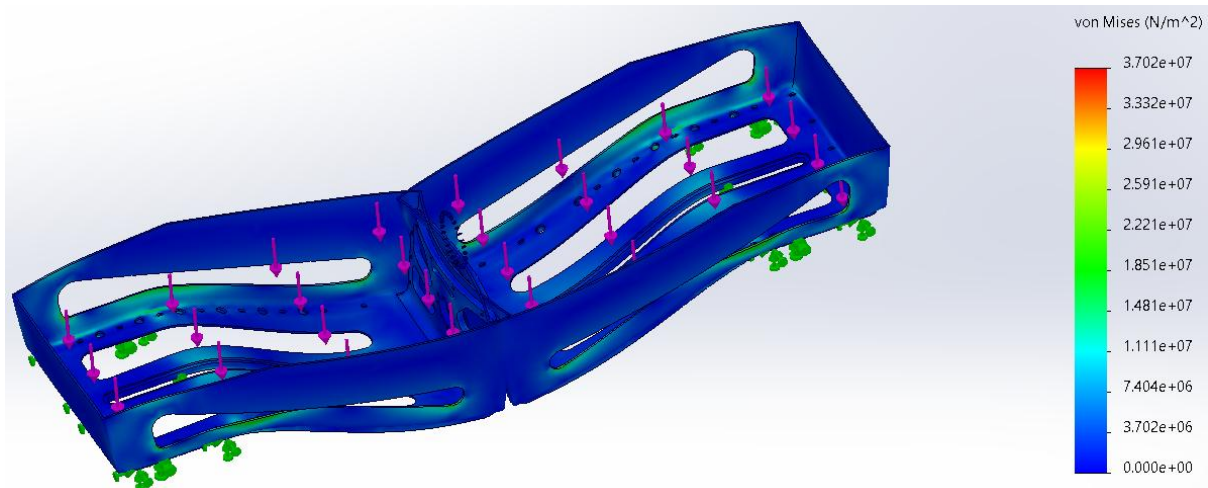


Figure 4-17 FEA for the semi-monocoque chassis assembly

Overall, the integrated layout of the body shell and reinforcement rail achieves a balance between structural strength, geometric adjustability, compact packaging, and manufacturability, making it suitable for use in the adjustable-geometry experimental vehicle platform.

4.2.2.4 Electronics Mounting Plate Design

To address the challenges of mounting diverse electronics - battery pack, motor controllers, communication modules, power management units, and sensors - with varying interfaces and evolving layouts, a modular universal mounting plate was developed, as shown in Figure 4-18. Fabricated from 3 mm thick aluminium alloy for stiffness at low mass, it features 50 rivet nuts arranged in a 40 mm $\times$ 40 mm grid, forming a dense array of threaded mounting interfaces that accommodates flexible and reconfigurable component placement. The plate is secured to the chassis using bolts and nylon washers, which provide electrical insulation, reduce vibration transmission, and prevent metal-to-metal surface damage.

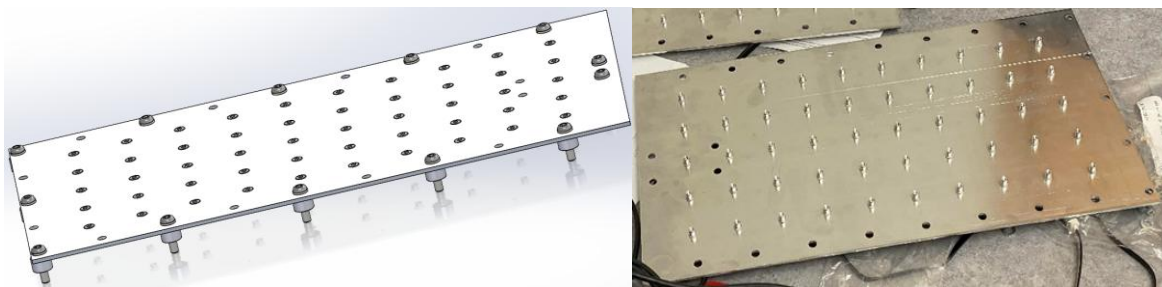


Figure 4-18 Electronics mounting plate with rivet nuts

Since the vehicle consists of two semi-monocoque chassis sections, a total of 100 threaded mounting points is available across the entire chassis. These threaded interfaces allow electronic components to be installed at flexible locations according to system requirements. In particular, custom-designed or 3D-printed electronic enclosures can be easily mounted and repositioned, enabling rapid installation, replacement, and adjustment of electronic modules, such as batteries shown in Figure 4-19. This significantly simplifies system integration and reduces the time required for experimental configuration changes.



Figure 4-19 DC battery mounted on the electronics mounting plate

In addition to supporting electronic components, the high-strength threaded mounting structure is capable of supporting small mechanical subsystems, such as auxiliary sensors, actuator modules, or lightweight robotic mechanisms.

4.2.3 Suspension Design

Conventional independent suspension systems offer excellent terrain adaptability but introduce excessive complexity and mass for a lightweight experimental platform, so a simpler articulated configuration was selected instead. To ensure symmetric load distribution and minimise experimental variability in motor energy consumption, the articulation axis was located at the vehicle's geometric centre rather than using a conventional offset pivot.

This required a rotational joint capable of accommodating electrical wiring between chassis sections while withstanding the bending moments from vehicle weight. An initial hollow shaft concept proved overly complex to manufacture. The final solution adopted a slewing bearing at the vehicle centre, which provides the required rotational freedom between front and rear chassis halves, offers sufficient structural stiffness to resist bending loads, and incorporates a hollow centre through which electrical cables are routed between chassis sections.

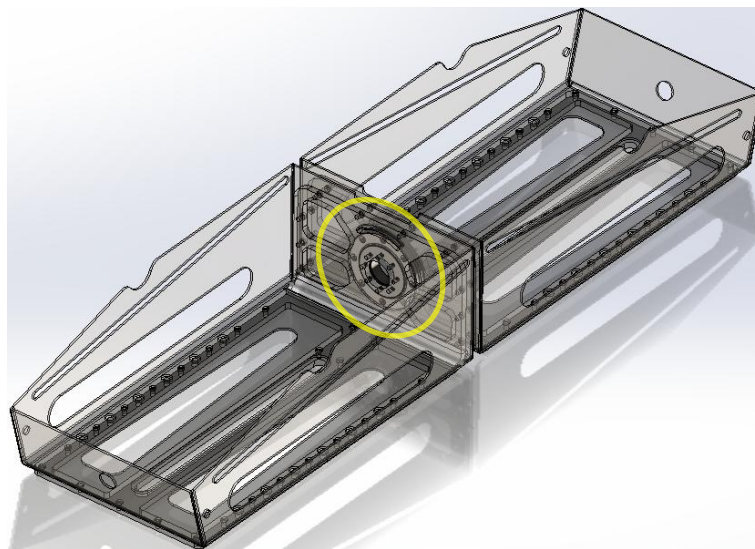


Figure 4-20 Slewing bearing within the semi-monocoque chassis

As highlighted in Figure 4-20, a slewing bearing was selected as the articulation interface between the front and rear chassis sections. Slewing bearings simultaneously support axial loads, radial loads, and overturning moments within a compact envelope, making them well-suited for centralised articulation joints (Aguirrebeitia et al., 2012). Mounting holes on both inner and outer rings allow direct bolted integration with structural components, eliminating

the need for a separate shaft and bearing housing assembly. Combined with the reinforced SMC structure, this provides a compact, rigid, and mechanically efficient connection between the two chassis halves. The inner structure is shown in Figure 4-21.

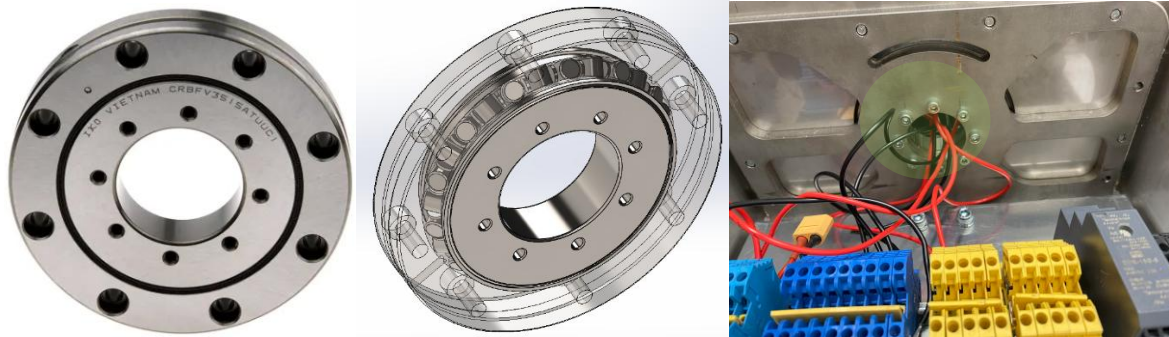


Figure 4-21 Slewing bearing: external appearance, roller structure, and application

4.2.4 Protection Cover Design

The protection cover shields internal electronics - battery, controllers, and communication modules - from mechanical impact, dust, and environmental exposure during outdoor operation.

Each of the two identical cover assemblies consists of a front and rear panel laser-cut and bent from 1 mm thick 304 stainless steel, giving four panels total. Rather than a conventional hinged or single-piece sliding cover, a segmented rotational-sliding mechanism is employed: the outer front panel rotates about a fixed pivot at the chassis edge, while the inner rear panel is pivotally connected to the front panel and simultaneously guided along slots in the body shell. This coupled motion allows smooth opening and closing without excessive clearance space.

As presented in Figure 4-22, the design provides three stable operating positions: fully closed, partially open, and fully open.



Figure 4-22 a. fully closed cover b. partially opened cover c. fully opened cover

When fully closed, a locating pin passively engages a semi-circular slot under gravity, ensuring panel alignment and preventing vibration-induced opening. When partially open (approximately 90° of outer panel rotation), the inner panel engages the curved slot geometry,

creating a second gravity-retained stable position. When fully open, both panels reach their maximum configuration, providing complete interior access.

This three-position design offers three key advantages. First, the articulated rotational-sliding mechanism significantly reduces operating torque compared to a conventional hinged or sliding cover of equivalent size (~ 0.5 m per assembly), improving ergonomic usability during frequent experimental access. Second, even fully open, the required clearance increases by only approximately 50% of vehicle length, enabling maintenance in confined laboratory environments. Third, the intermediate position improves operational safety during live testing: internal components remain visible for debugging and adjustment while the E-STOP remains accessible for rapid shutdown.

4.3 Summary of Mechanical Design

The mechanical design establishes an integrated structural framework balancing structural strength, functional flexibility, and experimental applicability.

At the subsystem level, the drive-steering system integrates propulsion and steering with quantitatively validated motor torque and power margins. Reinforced adapters, bearing-supported interfaces, and load-path optimisation mitigate combined torsional and bending stresses. The L-arm and adapter assemblies achieve sufficient strength under combined loading through material selection, geometric optimisation, and FEA validation, while decoupled load paths protect sensitive motor interfaces.

At the system level, the WSB, SMC, and reinforcement rails provide a mechanically efficient platform supporting both load transmission and dual-axis geometric adjustability. FEA confirms all primary structural components operate within acceptable stress limits under worst-case loading. The centrally articulated slewing bearing suspension maintains symmetric load distribution, reducing experimental uncertainty in energy consumption analysis. Modular features including the electronics mounting plate and protective cover further improve maintainability and operational usability.

The design provides a reliable physical foundation for subsequent system integration, control implementation, and experimental validation of vehicle geometry and steering strategy effects on energy consumption in off-road environments.

5 Electrical and Software Implementation

This chapter presents the electrical and software design of the vehicle, including the electrical system architecture, control system, and software implementation. Section 5.1 provides an overview of the electrical system design and the selection of the core electrical hardware. Section 5.2 describes the software implementation of the ESP32, which serves as the main vehicle controller. Section 5.3 presents the kinematic equations implemented within the control system to determine the individual wheel velocities of the drive motors and the steering positions of the steering motors. Finally, Section 5.4 provides a summary of the chapter.

5.1 Electrical System Design

An overview of the vehicle electrical system is shown in Figure 5-1. Three voltage levels are used within the system: a nominal 48 V supply for the drive motors, stepped down to 24 V for the steering motors and 5 V for the control system. A lithium-ion battery was selected as the power source due to its high energy density and established safety in mobile platform applications (Lipu et al., 2018).

Based on the maximum expected current draw, the battery positive terminal was protected by a 50 A fuse. Two emergency stop (E-stop) switches were installed at the front and rear of the vehicle and connected in series to control the main contactor. Activation of either switch de-energises the contactor and shuts down the entire system.

The use of modular converters and terminal blocks enables additional components or subsystems to be integrated with minimal redesign. This modularity is particularly advantageous for experimental and research platforms, where system configurations may evolve throughout the development process.

The low-voltage system is divided into two subsystems:

1. Vehicle control - processing the PS4 controller inputs, calculating the vehicle kinematics, and communicating with the motor controllers.
2. Data logging - periodically recording the currents of each of the eight motors in the vehicle.

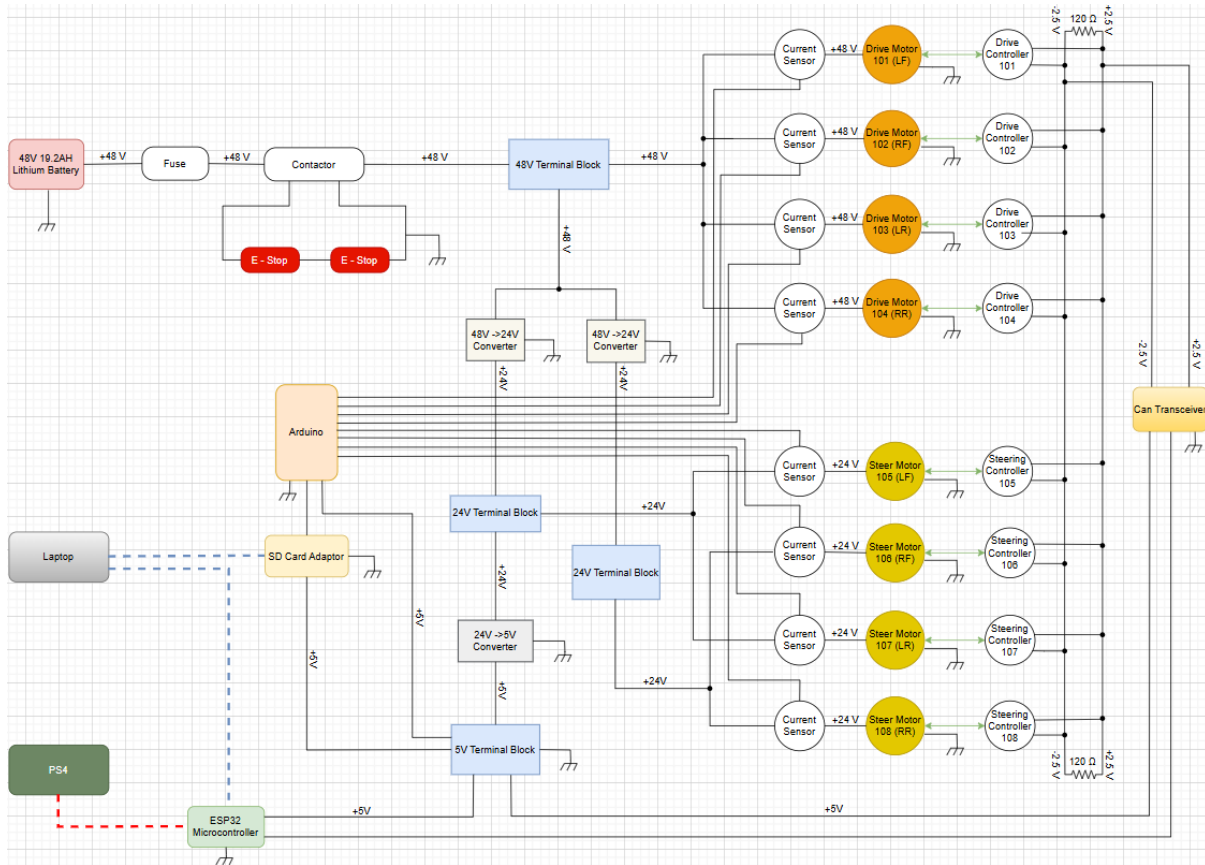


Figure 5-1 Electrical system schematic

The control system is implemented using an ESP32 microcontroller, which incorporates an internal CAN controller. An external CAN transceiver converts logic-level signals to the differential signalling required for CAN bus communication. Differential signalling over CAN High and CAN Low improves noise immunity and reduces susceptibility to electromagnetic interference, which is particularly important in high-current environments such as multi-motor electric vehicles (Husain, 2021).

Eight motors are connected to a single CAN bus in a linear topology, with all nodes connected in parallel to shared communication lines. This enables broadcast communication between devices. To ensure signal integrity, 120 Ω termination resistors are installed at both physical ends of the bus in accordance with the ISO 11898 CAN specification (Husain, 2021), providing impedance matching and reducing signal reflections.

The data acquisition (DAQ) system consists of a microcontroller, 20 A current sensors, and a storage module. The control and data logging functions are intentionally decoupled to ensure that the DAQ subsystem does not interfere with the core vehicle control architecture. An Arduino Mega is used for data processing and interfaces with multiple current sensors via its

analogue-to-digital converter (ADC), enabling simultaneous measurement of motor currents. Data is logged to an SD card using the SPI (Serial Peripheral Interface) protocol for subsequent analysis.

5.1.1 Core Electrical Component Selection

This section provides an explanation of the selection process for the key components used in the system, including the battery, remote controller, microcontroller, and data acquisition unit. The rationale behind each selection is discussed with respect to the functional requirements and overall system design.

5.1.1.1 Battery

Battery selection is primarily based on the required system voltage and energy capacity. The voltage must satisfy the highest requirement in the system, which is 48 V for the drive motors. To estimate the required capacity, the average mechanical power demand was first approximated using rolling resistance:

$$P = C_{rr} mgv$$

where C_{rr} is the coefficient of rolling resistance (0.1), m is the mass of the vehicle (100 kg) and v is the vehicle velocity. For an estimated continuous operating duration of 4 hours, the required energy is given by:

$$E = Pt$$

Applying a safety factor of 1.5 to account for uncertainties results in a design power of approximately 221 W, corresponding to an energy requirement of 884 Wh. This is equivalent to a minimum capacity of 18.4 Ah at 48 V. Based on this requirement, a 48 V, 19.2 Ah (922 Wh) lithium-ion battery pack was selected. Lithium-ion technology offers high energy density (approximately 150 - 200 Wh/kg), making it suitable for mobile robotic applications where weight and volume are critical constraints (Lipu et al., 2018). The selected pack uses cylindrical lithium-ion cells (21700 format) with a nominal voltage of 3.6 V and typical capacities of up to 5 Ah (Lipu et al., 2018). The battery is housed in an aluminium-reinforced enclosure with integrated mounting features to ensure secure installation within the vehicle chassis.

5.1.1.2 Microcontroller

The microcontroller functions as the central computing unit of the experimental vehicle. It receives linear velocity and steering commands from the PS4 controller and converts them into

individual wheel speed and position commands. These are transmitted over the CAN bus to the integrated motor controllers, which perform PID-based speed control (drive motors) or position control (steering motors).

The ESP32-WROOM-32D module was selected as the main controller due to its high computational capability, integrated peripherals, and wireless communication features (Espressif Systems, 2023). The ESP32 is based on a dual-core Tensilica LX6 processor operating at up to 240 MHz, providing significantly higher processing performance compared to common prototyping platforms such as Arduino. This enables real-time execution of control algorithms, sensor processing, and multi-tasking operations within the system.

Furthermore, the ESP32 supports FreeRTOS and multiple power management modes, allowing efficient task scheduling and energy optimisation (Espressif Systems, 2023). Combined with its relatively low cost and high level of integration, it provides a practical solution for embedded control applications.

Overall, the ESP32-WROOM-32D offers a suitable balance of performance, flexibility, and integration for implementing the control and communication requirements of the vehicle system.

5.1.1.3 Remote Controller

The remote controller provides the primary human - machine interface for commanding the vehicle, supplying inputs in the form of linear velocity and steering angle, as well as mode selection commands. A Sony DualShock 4 (PS4) controller was selected due to its suitability for teleoperated mobile robotics, where commercial gaming controllers are commonly used for intuitive and low-latency control (Naik & Srividya, 2021).

The controller provides both analogue and digital inputs, enabling direct mapping of user commands to vehicle motion. The analogue joystick is used for continuous control of linear velocity and steering angle, while digital inputs are assigned to mode switching and auxiliary functions.

Communication between the controller and the ESP32 is implemented via Bluetooth, eliminating the need for external interface hardware and reducing system complexity while maintaining adequate responsiveness for real-time teleoperation. Overall, the controller subsystem provides a simple and effective interface between the operator and the vehicle control architecture.

5.1.1.4 Data Acquisition Unit Selection

An Arduino Mega2560, in combination with a microSD card module and ACS712 20 A Hall-effect current sensors, was adopted as the data acquisition and logging unit. Arduino-based multi-channel data logging is a well-established approach for current and power measurement in experimental robotic platforms (Sabovic et al., 2020).

The system incorporates eight current sensors, each corresponding to an individual motor, requiring simultaneous multi-channel data acquisition. The Arduino Mega2560 provides 54 digital input/output pins and 16 analogue input channels, enabling direct interfacing with multiple sensors without the need for external multiplexing hardware.

This architecture provides a clear functional separation within the system, where the Arduino operates as a dedicated data acquisition node while the ESP32 handles real-time control tasks. This separation improves system modularity, reduces computational load on the main controller, and enhances overall maintainability.

5.2 Software Design

The software is implemented on an ESP32-based embedded control system responsible for vehicle motion control, actuator coordination, and system communication. An Arduino Mega operates in parallel as a dedicated data acquisition unit for multi-channel current measurement and data logging. The overall architecture separates real-time control tasks from experimental data recording to improve system determinism and modularity.

5.2.1 System Operation

The system operates in a continuous control loop following initialisation of communication interfaces, including Bluetooth connection to the PS4 controller, CAN bus communication, and SD card logging, using open-source libraries. User inputs from the PS4 controller are mapped to linear velocity and steering angle commands, as shown in Figure 5-2. A deadband filter is applied to remove noise and unintended small joystick inputs. These signals are then converted into vehicle motion commands.



Figure 5-2 Sony PS4 remote controller with its function

Vehicle motion is computed using a kinematic model based on classical vehicle dynamics (Gillespie, 1992; Wong, 2001). Key geometric parameters include wheelbase, track width, wheel diameter, and minimum turning radius. The turning radius is determined from the steering input, and angular velocity is calculated using:

$$\omega = \frac{v}{R}$$

Individual wheel velocities and steering angles are generated based on the selected drive mode, including skid steering, two-wheel steering, and four-wheel steering configurations. The corresponding kinematic relationships are presented in Section 5.2.2. In skid steering mode, differential wheel speeds are applied to each side of the vehicle, while steering-based modes employ Ackermann-type steering relationships. In four-wheel steering mode, rear wheel steering angles are applied in opposition to the front wheels to reduce the effective turning radius.

The drive motors operate in speed control mode, while the steering motors operate in position control mode using internal PID controllers. The vehicle supports two driving modes selectable via the controller: in-place rotation and radius-based turning.

In parallel, an Arduino Mega performs data acquisition by sampling motor current via ACS712 sensors and logging the data to an SD card for post-processing.

Following control computation, commands are converted into motor-specific units and transmitted via the CAN bus to eight distributed motor controllers, where local PID loops execute speed and position control.

5.2.2 Vehicle Kinematics

During vehicle operation, under different steering modes and turning radii, the rotational speeds and steering angles of each wheel vary accordingly. This implies that all four drive motors and four steering motors require individual control inputs. In this study, the angular velocity of the vehicle and the turning radius R are treated as key variables to establish the geometric and mathematical relationships governing the speeds of all eight motors. The mathematical models for all six steering scenarios are presented in the following sections.

5.2.2.1 Explicit Steering 2WS

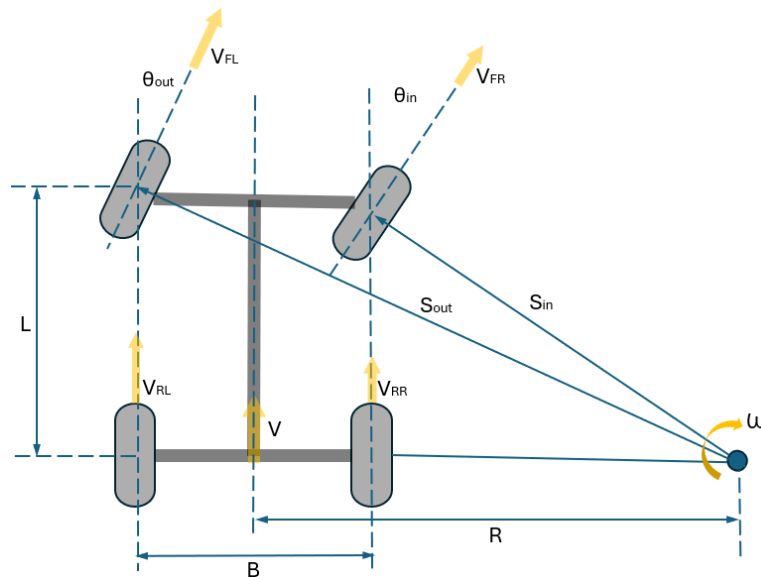


Figure 5-3 Two-wheel steering (2WS) configuration

For the two-wheel steering (2WS) configuration shown in Figure 5-3, only the front wheels are responsible for steering, while the rear wheels remain aligned with the vehicle body. Under a given angular velocity ω and turning radius R , the velocity and steering angle of each wheel can be derived based on the vehicle geometry. The velocities of the front wheels are:

$$v_{FL} = \omega S_{out} = \omega \sqrt{L^2 + \left(R + \frac{B}{2}\right)^2} \quad (5.1)$$

$$v_{FR} = \omega S_{in} = \omega \sqrt{L^2 + \left(R - \frac{B}{2}\right)^2} \quad (5.2)$$

The rear wheel velocities are:

$$v_{RL} = \omega \left(R + \frac{B}{2} \right), v_{RR} = \omega \left(R - \frac{B}{2} \right) \quad (5.3)$$

The steering angles of the front wheels are:

$$\theta_{FL} = \theta_{out} = \arctan \left(\frac{L}{R + \frac{B}{2}} \right) \quad (5.4)$$

$$\theta_{FR} = \theta_{in} = \arctan \left(\frac{L}{R - \frac{B}{2}} \right) \quad (5.5)$$

Since the rear wheels do not participate in steering:

$$\theta_{RL} = 0, \theta_{RR} = 0 \quad (5.6)$$

5.2.2.2 Explicit Steering 4WS

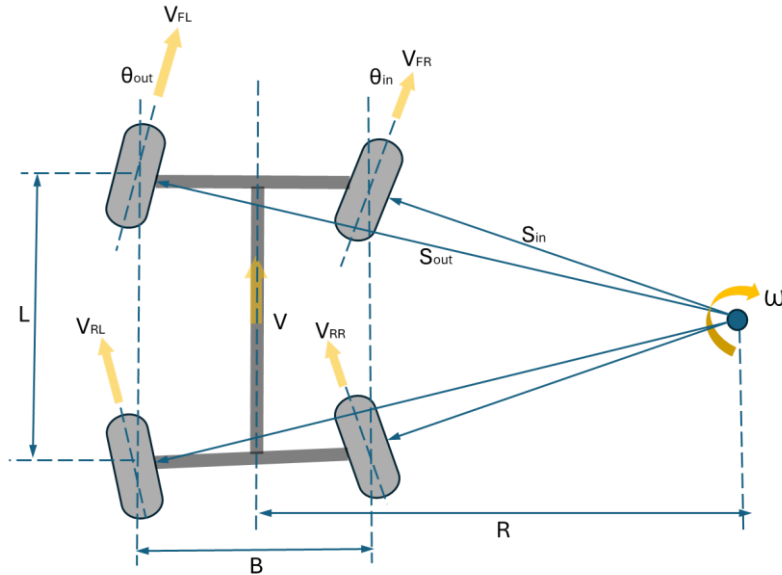


Figure 5-4 Four-wheel steering (4WS) configuration

For the four-wheel steering (4WS) configuration shown in Figure 5-4, both the front and rear wheels contribute to the steering motion. Under a given angular velocity ω and turning radius R , the velocity and steering angle of each wheel can be derived based on vehicle geometry. The velocities of the front wheels are expressed as:

$$v_{FL} = \omega S_{out} = \omega \sqrt{\left(\frac{L}{2} \right)^2 + \left(R + \frac{B}{2} \right)^2} \quad (5.7)$$

$$v_{FR} = \omega S_{in} = \omega \sqrt{\left(\frac{L}{2} \right)^2 + \left(R - \frac{B}{2} \right)^2} \quad (5.8)$$

For the rear wheels, due to symmetry:

$$v_{RL} = v_{FL}, v_{RR} = v_{FR} \quad (5.9)$$

The steering angles of the front wheels are:

$$\theta_{FL} = \theta_{out} = \arctan\left(\frac{L/2}{R + B/2}\right) \quad (5.10)$$

$$\theta_{FR} = \theta_{in} = \arctan\left(\frac{L/2}{R - B/2}\right) \quad (5.11)$$

The rear wheel steering angles are defined to be symmetric in the opposite direction:

5.2.2.3 Explicit Skid Steering

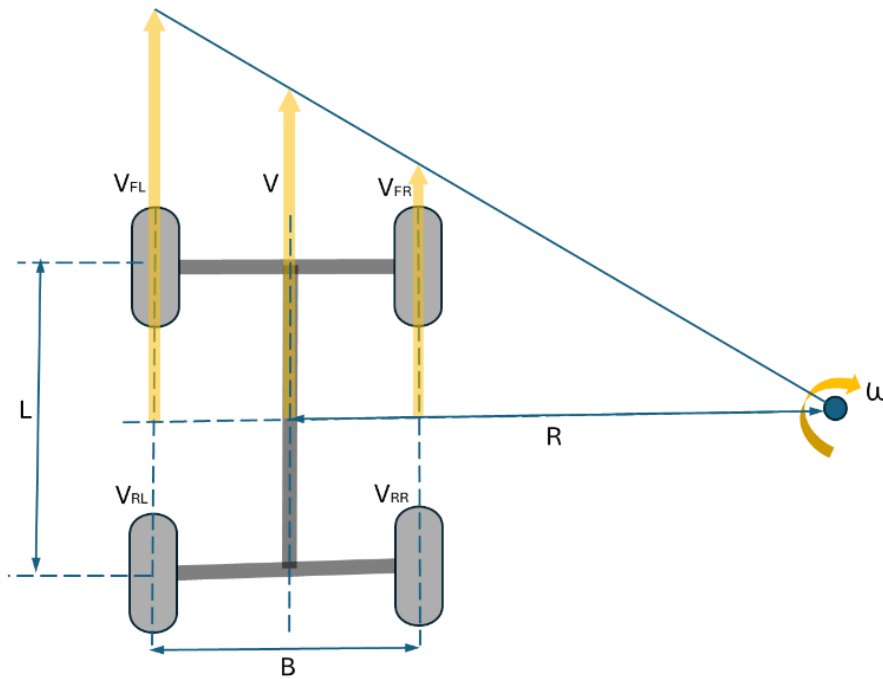


Figure 5-5 Skid steering (SS) configuration

For the skid steering (SS) configuration shown in Figure 5-5, the vehicle changes its direction by differential wheel speeds rather than steering angles. All wheels remain aligned with the vehicle body, and turning is achieved through speed differences between the left and right sides. The velocities of the wheels are given by:

$$v_{FL} = \omega \left(R + \frac{B}{2}\right), v_{FR} = \omega \left(R - \frac{B}{2}\right) \quad (5.12)$$

Due to symmetry between the front and rear wheels:

$$v_{RL} = v_{FL}, v_{RR} = v_{FR} \quad (5.13)$$

Since no steering mechanism is involved, all wheel steering angles are zero:

$$\theta_{FL} = 0, \theta_{FR} = 0, \theta_{RL} = 0, \theta_{RR} = 0 \quad (5.14)$$

5.2.2.4 Point Turn 2WS

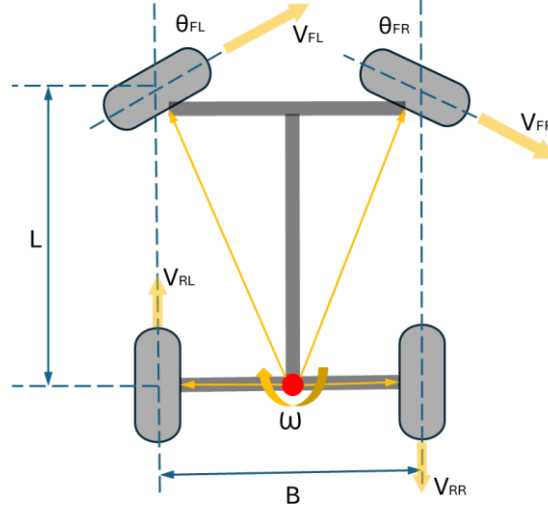


Figure 5-6 Two-wheel steering (2WS) configuration at zero turning radius ($R=0$)

For the two-wheel steering (2WS) configuration at zero turning radius ($R = 0$), which is shown in Figure 5-6, the vehicle performs a pivot-like motion about its the rear axle centre. In this case, the kinematic relationships simplify accordingly. The velocity of the front left wheel is:

$$v_{FL} = \omega \sqrt{L^2 + \left(\frac{B}{2}\right)^2} \quad (5.15)$$

Due to symmetry, the front right wheel rotates in the opposite direction:

$$v_{FR} = -v_{FL} \quad (5.16)$$

The rear wheel velocities are:

$$v_{RL} = \omega \frac{B}{2}, v_{RR} = -v_{RL} \quad (5.17)$$

The steering angle of the front left wheel is:

$$\theta_{FL} = \arctan\left(\frac{L}{B/2}\right) \quad (5.18)$$

The front right wheel follows a symmetric relationship:

$$\theta_{FR} = -\theta_{FL} \quad (5.19)$$

Since the rear wheels do not steer:

$$\theta_{RL} = 0, \theta_{RR} = 0 \quad (5.20)$$

5.2.2.5 Point Turn 4WS

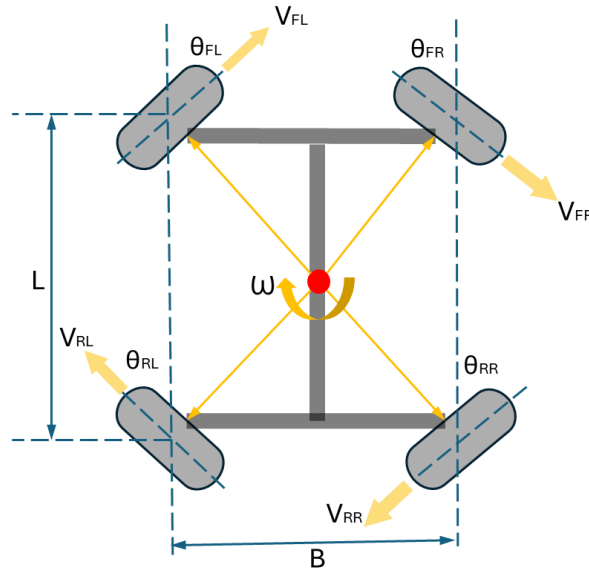


Figure 5-7 Four-wheel steering (4WS) configuration at zero turning radius ($R=0$)

For the four-wheel steering (4WS) configuration at zero turning radius ($R = 0$), which is shown in Figure 5-7, the vehicle performs a pivot-like motion about its geometric centre. In this condition, both the front and rear wheels actively contribute to the steering, resulting in a fully symmetric kinematic configuration. The velocity of the front left wheel is:

$$v_{FL} = \omega \sqrt{\left(\frac{L}{2}\right)^2 + \left(\frac{B}{2}\right)^2} \quad (5.21)$$

Due to symmetry, the front right wheel rotates in the opposite direction (5.16):

$$v_{FR} = -v_{FL}$$

The rear wheel velocities follow the same symmetry (5.13):

$$v_{RL} = v_{FL}, v_{RR} = v_{FR}$$

The steering angle of the front left wheel is:

$$\theta_{FL} = \arctan\left(\frac{L/2}{B/2}\right) \quad (5.22)$$

The front right wheel follows (5.19):

$$\theta_{FR} = -\theta_{FL}$$

For the rear wheels, the steering angles are coordinated with the front wheels:

$$\theta_{RL} = \theta_{FR}, \theta_{RR} = \theta_{FL} \quad (5.23)$$

5.2.2.6 Point Turn Skid Steering

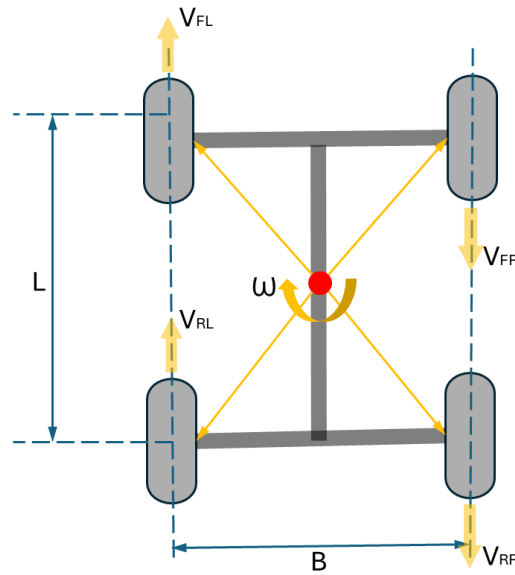


Figure 5-8 Skid steering (SS) configuration at zero turning radius ($R=0$)

For the skid steering (SS) configuration at zero turning radius ($R = 0$), which is shown in Figure 5-8, the vehicle performs a pure pivoting motion about its centre by means of differential wheel speeds. In this case, steering is achieved entirely through velocity differences between the left and right wheels, while all steering angles remain zero. The velocity of the front left wheel is:

$$v_{FL} = \omega \frac{B}{2} \quad (5.24)$$

The front right wheel rotates in the opposite direction (5.16):

$$v_{FR} = -v_{FL}$$

The rear wheel velocities follow the same symmetry (5.13):

$$v_{RL} = v_{FL}, v_{RR} = v_{FR}$$

Since no steering mechanism is involved, all wheel steering angles are zero.

5.3 Summary

This chapter presented the electrical and software design of the vehicle, including the electrical architecture, component selection, and implementation of the control and data acquisition systems. The software architecture and vehicle kinematic models used to determine wheel speeds and steering angles for the various drive configurations were also described. The resulting system provides the functional basis required for experimental evaluation under operating conditions. The following chapter presents the testing methodology and performance assessment of the completed vehicle system.

6 Experimental Analysis

6.1 Introduction

Unlike conventional passenger vehicles that predominantly operate on paved highways, agricultural machinery is required to navigate complex outdoor environments characterised by unpredictable terrain - including orchards, vineyards, and grass-covered farmland. These environments impose unique demands on vehicle design and energy management that differ fundamentally from those encountered in road transport. Orchards, in particular, present a structured but highly demanding operational context: vehicles must manoeuvre through narrow tree rows, execute frequent directional changes at row ends, negotiate uneven ground surfaces, and maintain precise path control while minimising soil compaction and crop disturbance.

The structure of orchard environments means that a significant proportion of total field operation time is spent on turning manoeuvres rather than straight-line travel. Studies of agricultural robot path planning have established that end-of-row headland turning can account for 20-40% of total field traversal time and energy, depending on field geometry, vehicle size, and turning strategy (Hameed et al., 2010; Oksanen & Visala, 2009; Bochtis & Vougioukas, 2008). As precision agricultural robots increasingly replace or supplement human-operated machinery, the energy efficiency of these turning manoeuvres takes on growing importance: battery-powered autonomous platforms must complete full operational cycles within the constraints of a single charge, making turning energy a critical design and operational parameter. A comprehensive review of agricultural robotics by Oliveira et al. (2021) confirms that energy management is among the most critical operational challenges for battery-powered field robots.

The energy consumed during a turning manoeuvre is governed by the interaction between steering strategy, vehicle geometry, wheel-ground interaction, and terrain inclination. Different steering mechanisms exhibit fundamentally different energy profiles: skid-steering systems (widely reviewed in Raji & Lal, 2023) - widely used in compact robotic platforms for their mechanical simplicity - generate propulsive force through differential wheel speed, producing lateral wheel slip and substantial frictional energy dissipation. In contrast, wheel-controlled steering strategies (two-wheel steering and four-wheel steering) steer through angular deflection of one or more wheel pairs, reducing lateral slip and the associated energy losses. The relative merits of these strategies, and their sensitivity to geometric parameters such as

wheelbase and track width, are not well established under realistic agricultural conditions, particularly on sloped terrain.

This chapter presents a systematic experimental investigation of the energy consumption of a four-wheel-independent-steering (4WIS) agricultural robot platform across a comprehensive matrix of geometric configurations and steering modes. Seven distinct motion modes are evaluated, representing the full range of steering strategies encountered in practical orchard and field operations.

The seven motion modes evaluated in this study are listed in Table 6.1.

Table 6.1 Summary of Motion Modes: Steering and Drive Actuation, and Energy Correction Requirement

Mode	Steering Actuation	Drive Motor Behaviour	Energy Correction
Straight-line driving	None (baseline)	All 4 wheels, same direction, same speed	Reference value
2WS arc ($R = 1.5$ m)	Front 2 wheels only, equal angles	Rear 2 wheels, opposing directions, speed matched to turning rate	Not required
4WS arc ($R = 1.5$ m)	All 4 wheels; same-side front/rear counter-rotate	All 4 wheels same direction; outer > inner speed (Ackermann)	Not required
Skid-steer arc ($R = 1.5$ m)	None	Differential side speeds (same side equal, sides differ)	Not required
2WS point turn ($R = 0$)	Front 2 wheels, equal & opposite angles	Rear 2 wheels, opposite directions, matched speed; pivots about rear-axle centre	Required
4WS point turn ($R = 0$)	All 4 wheels; same-side pairs equal speed, opposite angular direction	All 4 wheels same direction; outer pair > inner pair	Required
Skid-steer point turn ($R = 0$)	None	Two sides equal speed, opposite directions; pivots about geometric centre	Required

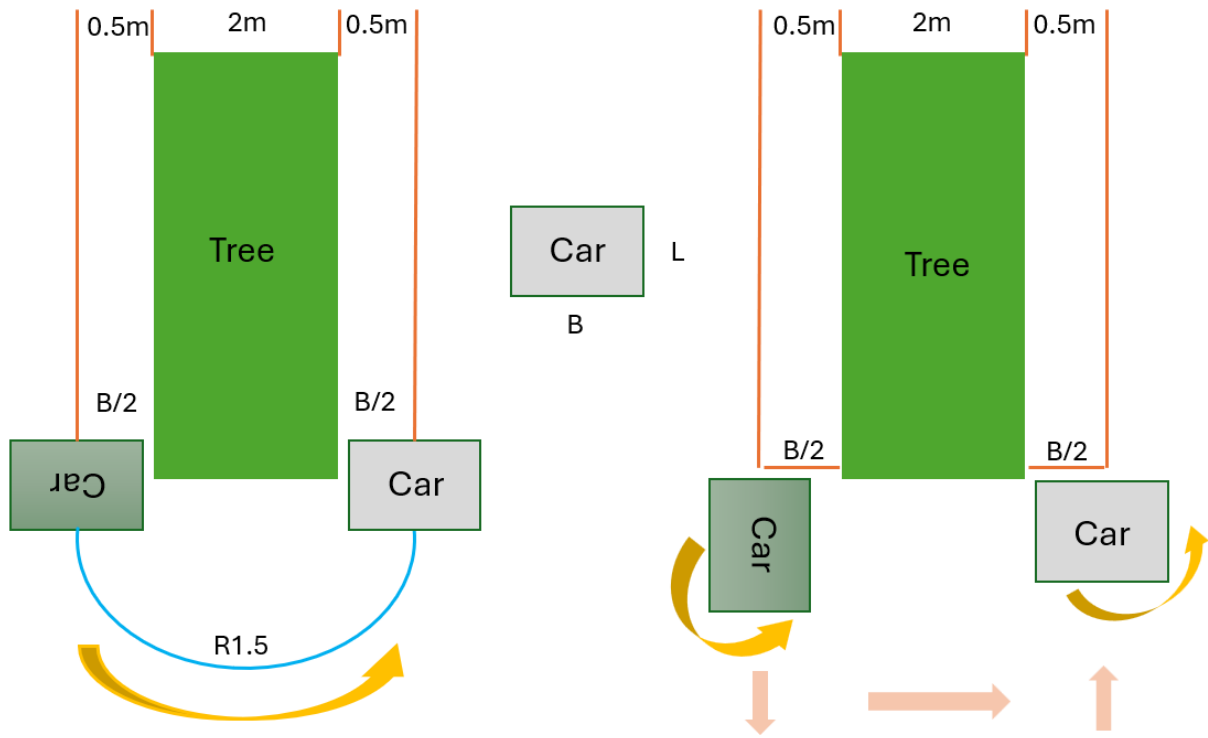


Figure 6-1 Steering R1.5 vs R0

In Figure 6-1, the relationship between vehicle geometry and turning energy is another key focus of this study. Vehicle mass is held constant throughout all experiments, making the geometric configuration - specifically, the combination of track width B and wheelbase L - the primary independent variable governing manoeuvre dynamics. Three values of each parameter are tested: $B \in \{400, 600, 800\}$ mm and $L \in \{400, 600, 800\}$ mm, yielding a full 3×3 factorial matrix of nine geometric configurations for flat-terrain tests, and three $B = 800$ mm configurations for slope tests.

The nine geometric configurations tested on flat terrain are: $B800 \times L800$, $B600 \times L800$, $B400 \times L800$, $B800 \times L600$, $B600 \times L600$, $B400 \times L600$, $B800 \times L400$, $B600 \times L400$, and $B400 \times L400$.

All experiments are conducted on grass terrain at two inclination levels: near-horizontal (mean inclination 0.216° , hereafter referred to as flat terrain, $D = 0^\circ$) and a steep grass slope (mean inclination 17.618° , referred to as the 18° slope, $D = 18^\circ$). Together, these two conditions encompass the range of surface gradients encountered in representative orchard and pasture environments. The combination of seven motion modes, nine geometric configurations, and two terrain inclinations produces a comprehensive experimental dataset from which energy efficiency rankings, geometric sensitivity analyses, and predictive regression models are derived.

6.2 Flat Ground Measurement Results and Data Analysis

Before energy measurements were conducted, ground inclination at both test sites was characterised using a digital inclinometer. One hundred uniformly distributed measurement points were sampled at a 30m diameter flat terrain site, yielding a mean inclination of 0.216° . Terrain characterisation using a digital inclinometer prior to energy testing is consistent with best practice in outdoor agricultural robot experiments, as terrain slope directly influences gravitational energy components (Taghavifar & Mardani, 2017). (standard deviation $< 1.7^\circ$). This value is sufficiently close to horizontal that gravitational energy contributions to wheel-ground interaction are negligible, and the site is classified as flat terrain ($D = 0^\circ$) for all subsequent analyses. One hundred points were similarly sampled at the slope test site, returning a mean inclination of 17.62° (standard deviation $< 2^\circ$), confirming a consistent and extreme grass slope. These two precisely characterised terrain conditions provide a well-controlled experimental basis for comparing energy consumption under fundamentally different gravitational loading regimes.

6.2.1 Individual Analysis by Configuration - Flat Ground

Based on the ratio of track width to wheelbase, B400×L600 and B600×L800 can be classified within the scope of B400×L800, and are therefore represented by the B400×L800 analysis, which exhibits a more pronounced dimensional ratio; these configurations are not described in detail in this section. Similarly, B600×L400 and B800×L600 fall within the scope of B800×L400 and are represented by that analysis, and are likewise not described in detail. For square configurations, the largest B800×L800 are retained for full analysis, while B600×L600 and B400×L400 are not described in detail. Detailed data and analysis for all configurations not covered in this section are provided in the Appendix.

6.2.1.1 B800×L800 Robot Platform Data Analysis

This section presents a comprehensive analysis of the energy consumption characteristics of the B800×L800 robot platform (wheelbase 800 mm, track width 800 mm) operating on flat terrain ($D = 0^\circ$). Drive motors are powered at 48 V and steer motors at 24 V; energy is computed as $E = \sum |I| \cdot V \cdot \Delta t$. Six motion modes are evaluated: R=0 two-wheel steering (2WS), R=0 four-wheel steering (4WS), R=0 Skid Steer, and the corresponding R=1.5 m arc-path variants. The straight-drive correction totals 359.23 J, apportioned according to the measured straight-line power ratio into 343.45 J for drive motors (95.6%) and 15.78 J for steer motors (4.4%), applied to R=0 modes only; the R=1.5 m arc paths already contain the required displacement. For the

R1.5 skid-steering test, the vehicle did not achieve the intended 1.5 m turning radius. Instead, an actual turning radius of approximately 3.75 m was observed during the manoeuvre.

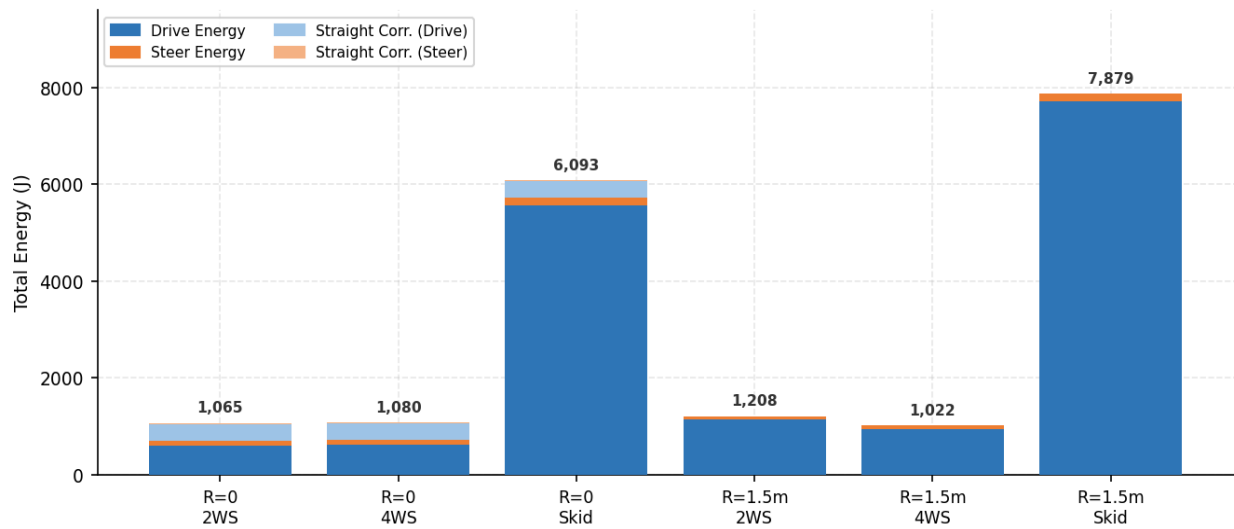


Figure 6-2 Total energy by steering mode - B800×L800 platform (blue=drive, orange=steer, light=straight correction; grey hatching=failed/invalid)

Figure 6-2 reveal the energy efficiency hierarchy across all valid steering modes. Ranked from lowest to highest total energy: 1st: R=1.5m 4WS (1,021.62 J); 2st: R=0 2WS (1,064.63 J); 3st: R=0 4WS (1,079.98 J); 4st: R=1.5m 2WS (1,207.85 J); 5st: R=0 Skid Steer (6,092.74 J); 6st: R=1.5m Skid (7,879.10 J). Skid Steer modes consume substantially more energy than wheel-controlled steering due to sustained wheel-slip converting drive energy into frictional heat. Among valid modes, R=1.5m 4WS at 1,021.62 J is the most efficient; R=1.5m Skid at 7,879.10 J is the least efficient. Steer motor energy constitutes only 2.1-9.8% of the total in all modes, confirming that steering actuation is not a primary energy driver.

Figures 6-3-6-4 present per-motor energy data for all six modes, providing a quantitative basis for drive-force distribution analysis.

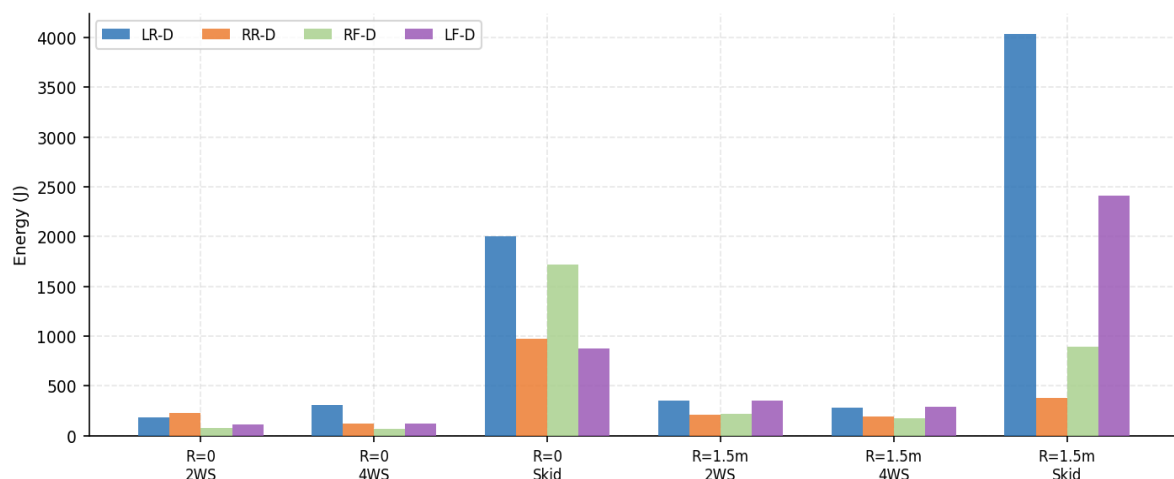


Figure 6-3 Drive motor energy by mode - B800×L800 platform

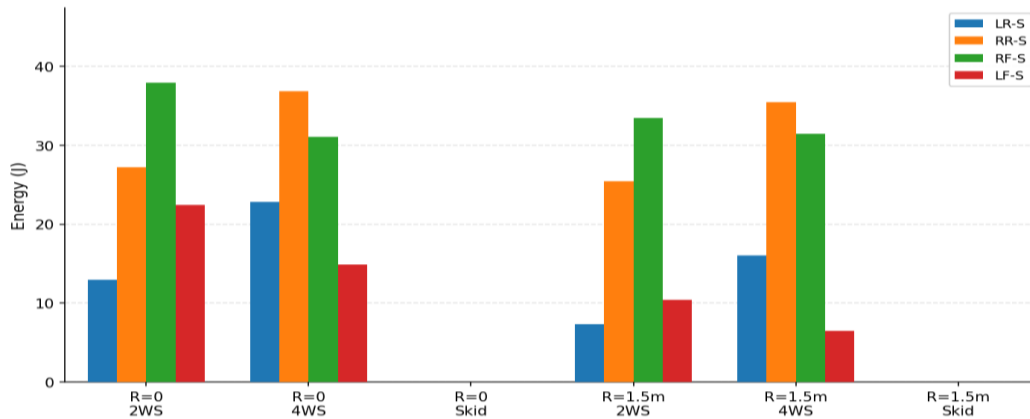


Figure 6-4 Steer motor energy by mode - B800×L800 platform

The drive motor load distribution is analysed mode by mode below. For R=0 Skid Steer, the LR-D + RF-D diagonal pair carries 66.9% of drive energy, compared with 33.1% for the opposing pair. This diagonal dominance is characteristic of skid-steer kinematics, where counter-rotating wheel pairs generate the turning moment. For R=1.5m 2WS, the left-side pair (LR-D + LF-D) absorbs 62.2% of drive energy versus 37.8% for the opposite side. For R=1.5m 4WS, the left-side pair (LR-D + LF-D) absorbs 60.3% of drive energy versus 39.7% for the opposite side. The higher energy consumption of the left-side drive motors reflects the characteristics of the clockwise (right-turning) manoeuvre. For R=1.5m Skid, the LR-D + RF-D diagonal pair carries 63.9% of drive energy, compared with 36.1% for the opposing pair.

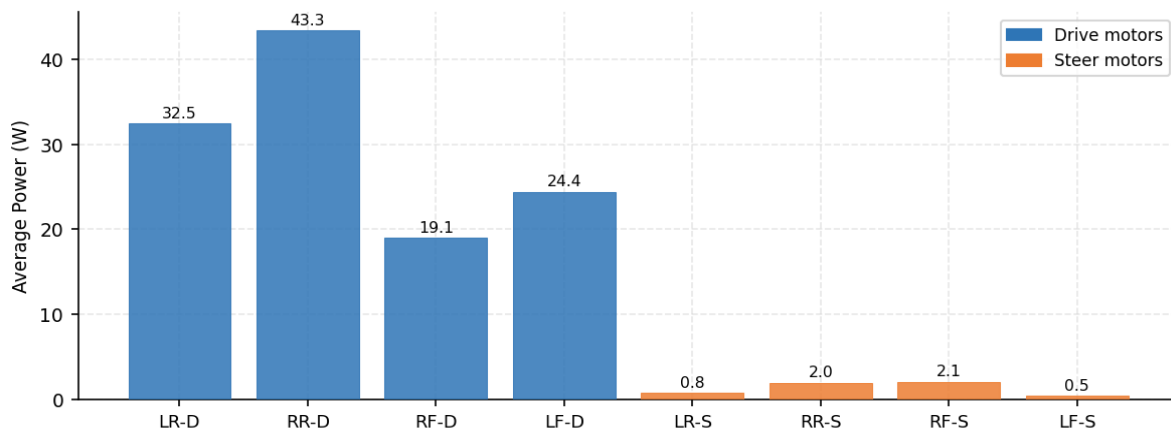


Figure 6-5 Straight-drive average power per motor - B800×L800 platform

During straight-line travel shown in Figure 6-5, the overall average power is 124.73 W, of which drive motors consume 119.25 W (95.6%) and steer motors 5.48 W (4.4%). The rear-axle pair consumes 63.5% of total drive power, indicating a rearward longitudinal weight bias under

steady-state rolling conditions. Lateral load distribution is well balanced (left: 47.7%, right: 52.3%).

In summary, the experimental data demonstrate that R=1.5m 4WS (1,021.62 J) is the most energy-efficient valid steering strategy for the B800×L800 platform. Skid Steer modes impose a substantial energy penalty and skid-steer operation should be avoided or restricted in energy-sensitive applications.

6.2.1.2 B600×L600 Robot Platform Data Analysis

This section presents the energy consumption data for the B600×L600 platform (wheelbase 600 mm, track width 600 mm, B/L = 1.00) on flat terrain. Together with B800×L800 (Section 6.2.1.1) and B400×L400 (Section 6.2.1.3), this forms the B = L group. The analysis below focuses on key similarities and differences relative to the B800×L800 reference (Section 6.2.1.1).

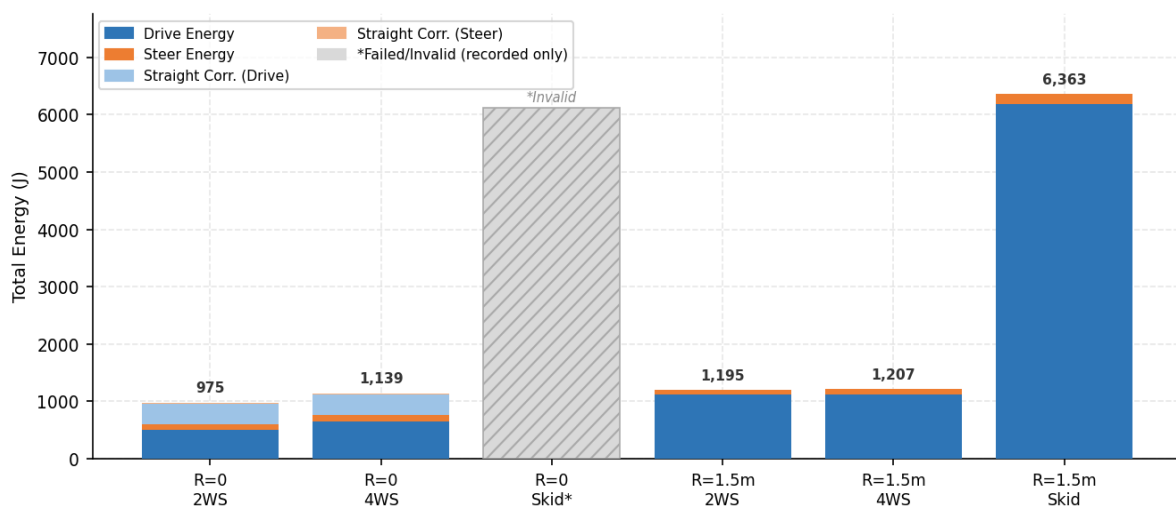


Figure 6-6 Total energy by steering mode - B600 x L600 platform (blue=drive, orange=steer, light=straight correction, grey hatching=failed/invalid)

Figure 6-6 shows the total energy by steering mode for B600×L600. The energy hierarchy is consistent with B800×L800 (Section 6.2.1.1): wheel-steered modes are substantially more efficient than Skid Steer, with R=0 2WS (974.82 J) the most efficient valid mode - 4.6% lower

than B800×L800's optimum of 1,021.62 J, reflecting the reduced 600 mm track width. R=1.5m Skid at 6,194.26 J is the most energy-intensive mode.

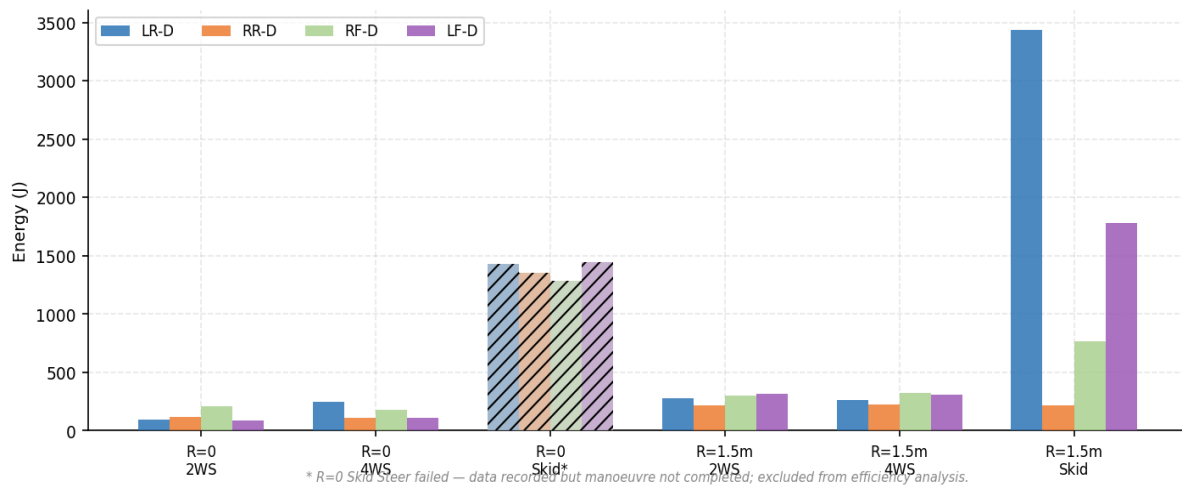


Figure 6-7 Drive motor energy by mode - B600 x L600 platform

Figure 6-7 shows per-drive-motor energy for B600×L600. Load distribution qualitatively mirrors B800×L800: the left-side pair (LR-D + LF-D) absorbs a larger share in R=1.5m arc modes due to the clockwise turning geometry, and the LR-D + RF-D diagonal dominates in R=1.5m Skid. Absolute per-motor values are proportionally lower than B800×L800.

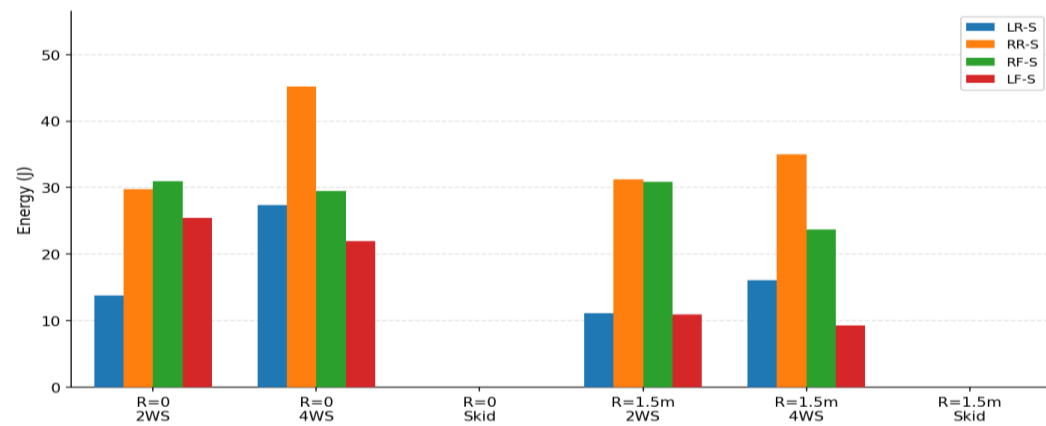


Figure 6-8 Steer motor energy by mode - B600 x L600 platform

Figure 6-8 shows steer motor energy, which is zero for all Skid Steer modes (mechanically locked) and ranges from 7.0% to 10.9% of total energy in wheel-steered modes - comparable to B800×L800 (6.3 - 9.8%). Steer actuation remains a secondary energy contributor across all modes.

In summary, the experimental data demonstrate that R=0 2WS (974.82 J) is the most energy-efficient valid steering strategy for the B600×L600 platform. The R=0 Skid Steer mode cannot

be executed on this platform; skid-steer operation should be avoided or restricted in energy-sensitive applications.

6.2.1.3 B400×L400 Robot Platform Data Analysis

This section presents the energy consumption data for the B400×L400 platform (wheelbase 400 mm, track width 400 mm, B/L = 1.00) on flat terrain. This is the smallest configuration in the B = L group and the smallest platform tested overall. The analysis focuses on similarities and differences relative to the B800×L800 reference (Section 6.2.1.1).

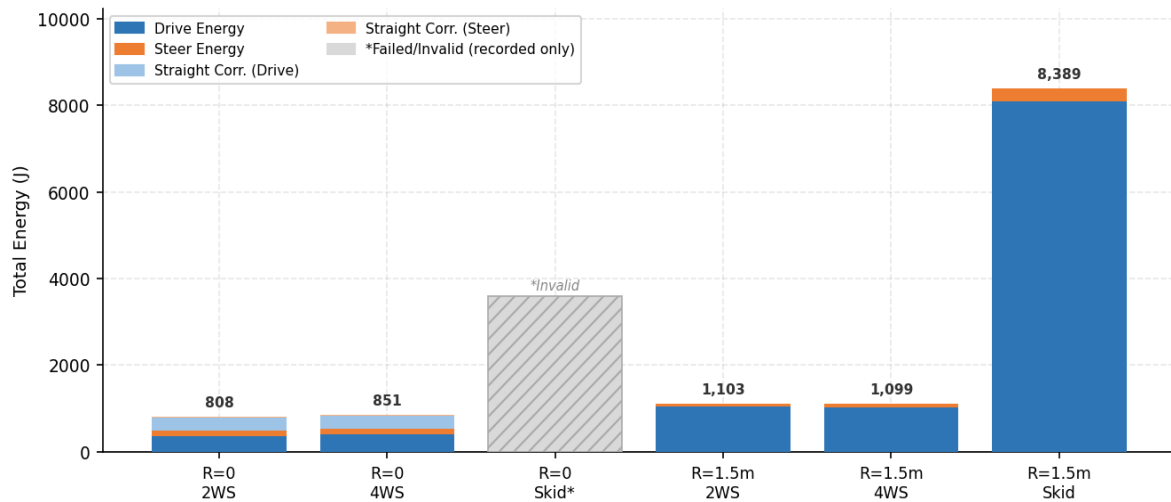


Figure 6-9 Total energy by steering mode -- B400xL400 platform (blue=drive, orange=steer, light=straight correction; grey hatching=failed/invalid)

Figure 6-9 shows the total energy for B400×L400. The energy hierarchy follows the same pattern as B800×L800 (Section 6.2.1.1), but absolute values are notably lower: R=0 2WS at 807.88 J is the most efficient mode, 21% below B800×L800's optimum (1,021.62 J), attributable to the smaller 400 mm track width substantially reducing pivot-turn resistance. R=0 Skid Steer failed on this platform; R=1.5m Skid at 8,093.28 J is the costliest valid mode.

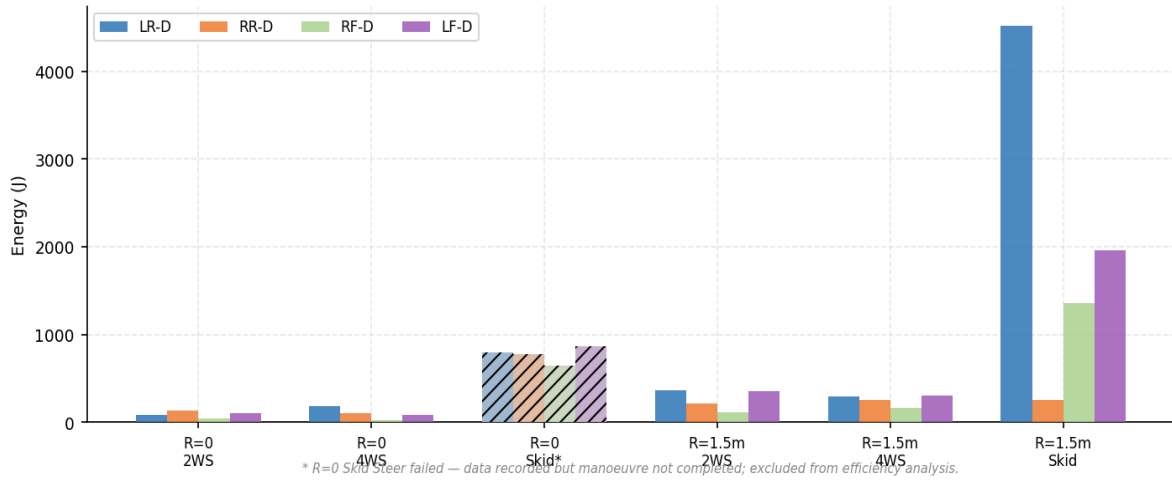


Figure 6-10 Drive motor energy by mode - B400 x L400 platform

Figure 6-10 shows per-drive-motor energy for B400×L400. The distribution pattern is qualitatively consistent with B800×L800: left-side dominance in R=1.5m arc modes and LR-D + RF-D diagonal dominance in R=1.5m Skid (LR-D carries 55.8% of total drive energy). Absolute values are lower than B800×L800 due to the reduced platform size.

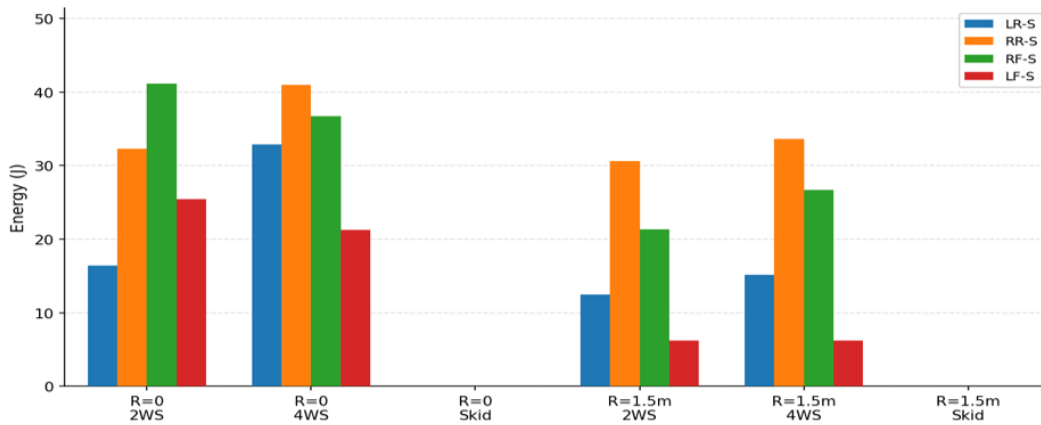


Figure 6-11 Steer motor energy by mode - B400 x L400 platform

Figure 6-11 shows steer motor energy, zero for Skid Steer modes and ranging from 6.4% to 15.5% for wheel-steered modes - broader than B800×L800 (6.3 - 9.8%). The higher steer fraction in R=0 pivot modes reflect the proportionally greater steer effort relative to the lower total drive energy of the compact B400×L400 platform.

In summary, the experimental data demonstrate that R=0 2WS (807.88 J) is the most energy-efficient valid steering strategy for the B400×L400 platform. The R=0 Skid Steer mode cannot be executed on this platform; skid-steer operation should be avoided or restricted in energy-sensitive applications.

6.2.1.4 B400×L800 Robot Platform Data Analysis

This section presents a comprehensive analysis of the energy consumption characteristics of the B400×L800 robot platform (wheelbase 800 mm, track width 400 mm) operating on flat terrain ($D = 0^\circ$). The straight-drive correction totals 385.96 J, apportioned according to the measured straight-line power ratio into 364.48 J for drive motors (94.4%) and 21.48 J for steer motors (5.6%), applied to R=0 modes only; the R=1.5 m arc paths already contain the required displacement. The R=0 Skid Steer mode (marked *) could not be completed on this platform - the robot ran and recorded energy data for the listed duration, but the manoeuvre was not executed; those values are shown for reference only and excluded from all efficiency comparisons. For the R1.5 skid-steering test, the vehicle did not achieve the intended 1.5 m turning radius. Instead, an actual turning radius of approximately 15 m was observed during the manoeuvre.

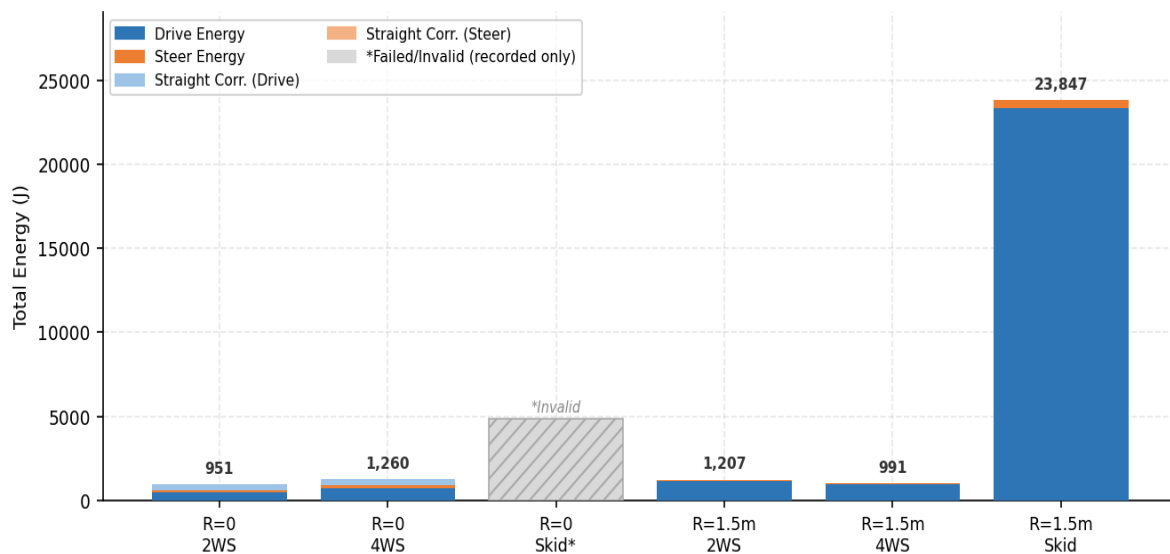


Figure 6-12 Total energy by steering mode - B400 x L800 platform (blue=drive, orange=steer, light=straight correction, grey hatching=failed/invalid)

Figure 6-12 reveal the energy efficiency hierarchy across all valid steering modes. Ranked from lowest to highest total energy: 1st: R=0 2WS (950.76 J); 2st: R=1.5m 4WS (990.58 J); 3st: R=1.5m 2WS (1,207.13 J); 4st: R=0 4WS (1,260.49 J); 5st: R=1.5m Skid (23,847.19 J). The R=0 Skid Steer mode is excluded from this comparison as the manoeuvre was not successfully executed. Skid Steer modes consume substantially more energy than wheel-controlled steering due to sustained wheel-slip converting drive energy into frictional heat. Among valid modes, R=0 2WS at 950.76 J is the most efficient; R=1.5m Skid at 23,847.19 J is the least efficient.

Steer motor energy constitutes only 2.1-11.9% of the total in all modes, confirming that steering actuation is not a primary energy driver.

Figures 6-13-6-14 present per-motor energy data for all six modes, providing a quantitative basis for drive-force distribution analysis.

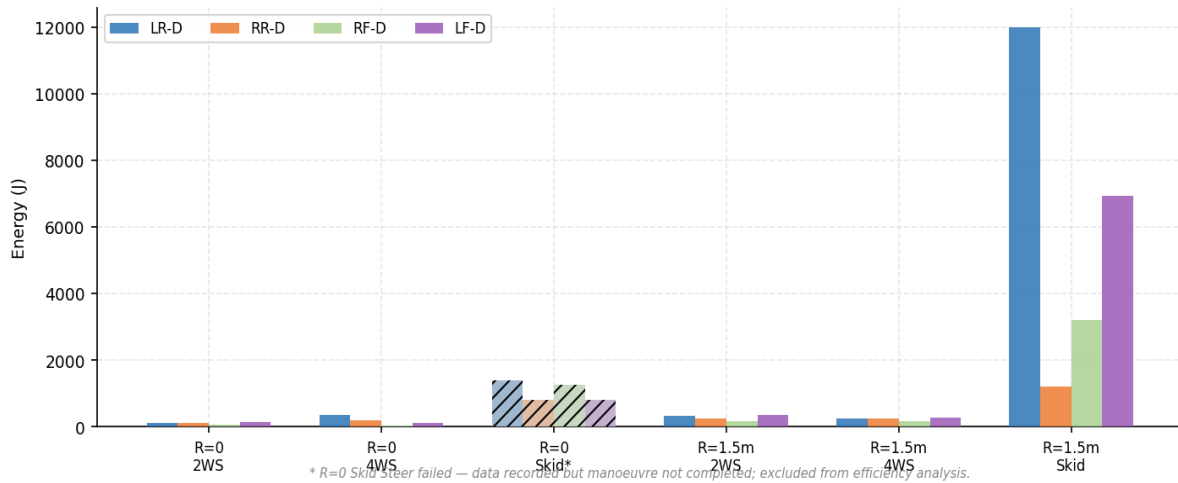


Figure 6-13 Drive motor energy by mode - B400 x L800 platform

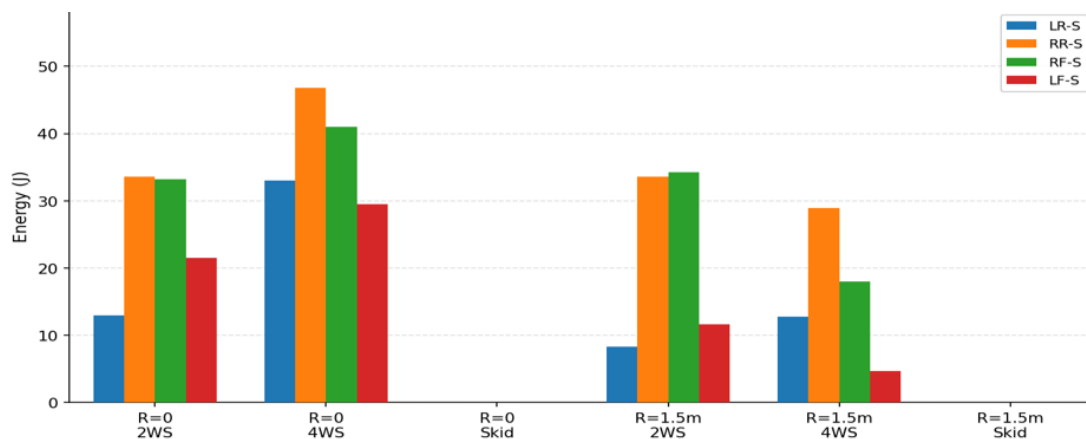


Figure 6-14 Steer motor energy by mode - B400 x L800 platform

R=0 Skid* (failed mode): motor data was recorded but the manoeuvre was not completed; the observed load distribution does not represent a valid steering state and is shown for reference only. For R=1.5m 2WS, the left-side pair (LR-D + LF-D) absorbs 62.0% of drive energy versus 38.0% for the opposite side. For R=1.5m 4WS, drive load is well distributed laterally (left: 55.7%, right: 44.3%). The higher energy consumption of the left-side drive motors reflects the characteristics of the clockwise (right-turning) manoeuvre. For R=1.5m Skid, the LR-D + RF-D diagonal pair carries 65.1% of drive energy, compared with 34.9% for the opposing pair. This diagonal dominance seems a characteristic of skid-steer kinematics, where counter-rotating wheel pairs generate the turning moment.

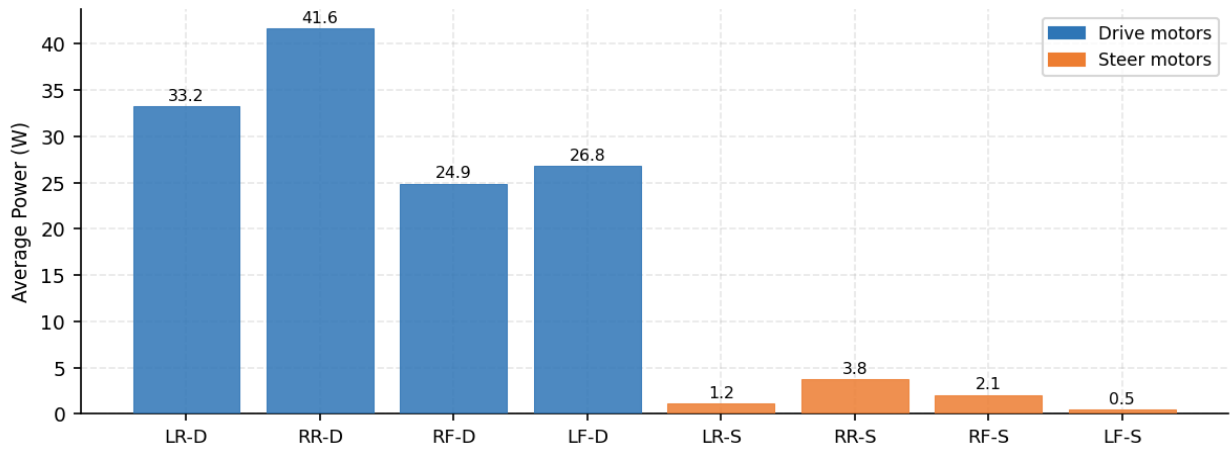


Figure 6-15 Straight-drive average power per motor - B400 x L800 platform

During straight-line travel shown in Figure 6-15, the overall average power is 134.02 W, of which drive motors consume 126.56 W (94.4%) and steer motors 7.46 W (5.6%). The rear-axle pair consumes 59.1% of total drive power, indicating a rearward longitudinal weight bias under steady-state rolling conditions.

In summary, the experimental data demonstrate that R=0 2WS (950.76 J) is the most energy-efficient valid steering strategy for the B400×L800 platform. The R=0 Skid Steer mode cannot be executed on this platform; skid-steer operation should be avoided or restricted in energy-sensitive applications.

6.2.1.5 B400×L600 Robot Platform Data Analysis

This section presents the energy consumption data for the B400×L600 platform (wheelbase 600 mm, track width 400 mm, B/L = 0.67) on flat terrain. Together with B400×L800 (B/L = 0.50, Section 6.2.1.4) and B600×L800 (B/L = 0.75, Section 6.2.1.6), this forms the B < L group, ordered by increasing B/L ratio. The analysis compares with the B400×L800 reference, examining the effect of reducing wheelbase from 800 mm to 600 mm at the same track width.

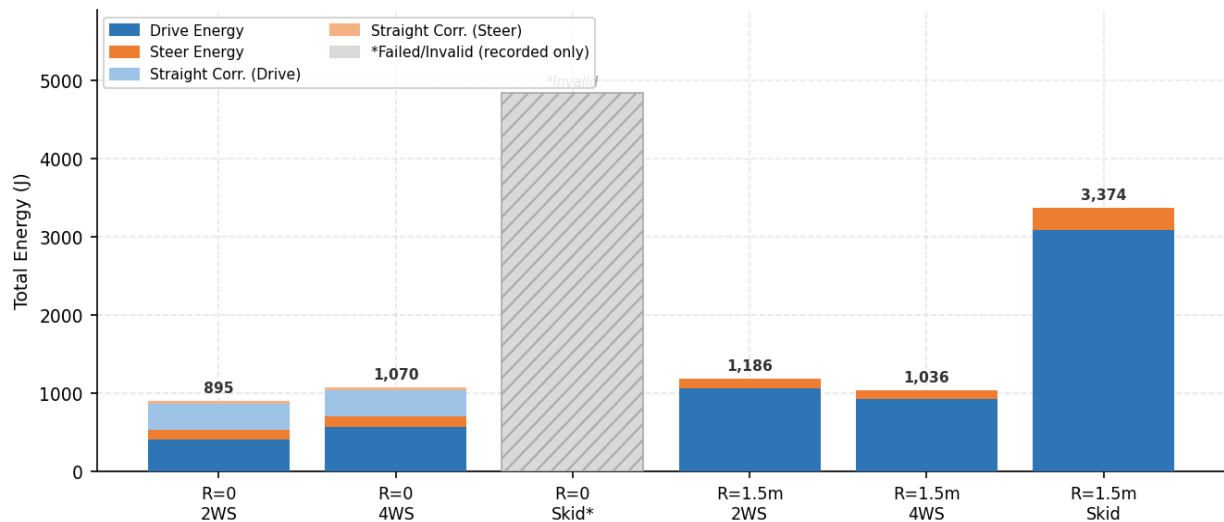


Figure 6-16 Total energy by steering mode - B400 x L600 platform (blue=drive, orange=steer, light=straight correction, grey hatching=failed/invalid)

Figure 6-16 shows the total energy for B400×L600. The energy hierarchy is consistent with B400×L800 (Section 6.2.1.4): R=1.5m 4WS is the most efficient mode at 1,035.71 J - a modest 4.5% increase over B400×L800 (990.58 J) - attributable to the shorter wheelbase ($L = 600$ mm vs 800 mm). R=0 Skid Steer also failed on this platform. Table 6.4 and Figures 6.7-6.8 present per-motor energy data for all six modes, providing a quantitative basis for drive-force distribution analysis.

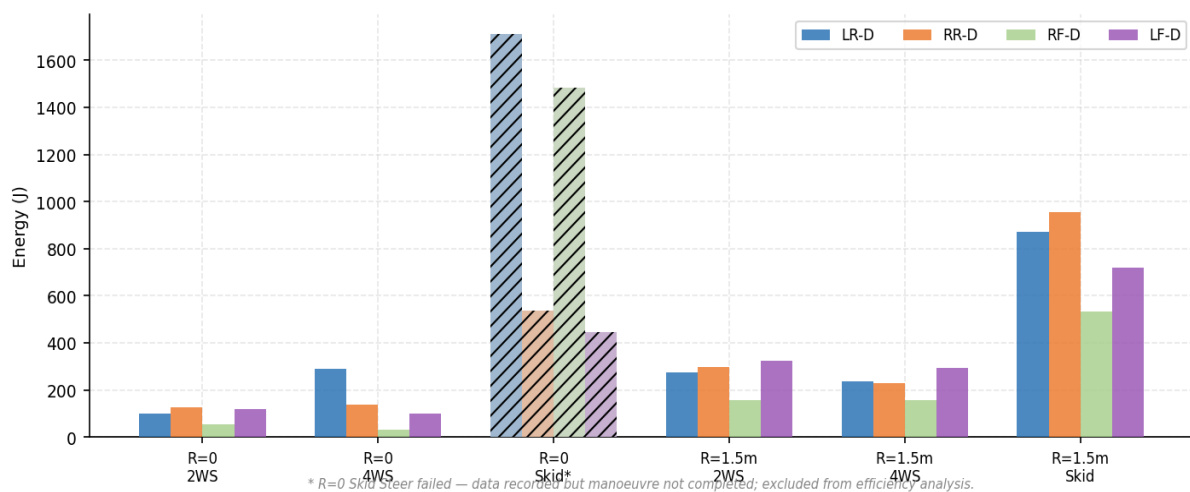


Figure 6-17 Drive motor energy by mode - B400 x L600 platform

Figure 6-17 shows per-drive-motor energy for B400×L600. Distribution patterns are qualitatively consistent with B400×L800, with left-side dominance in R=1.5m 2WS and 4WS arc modes.

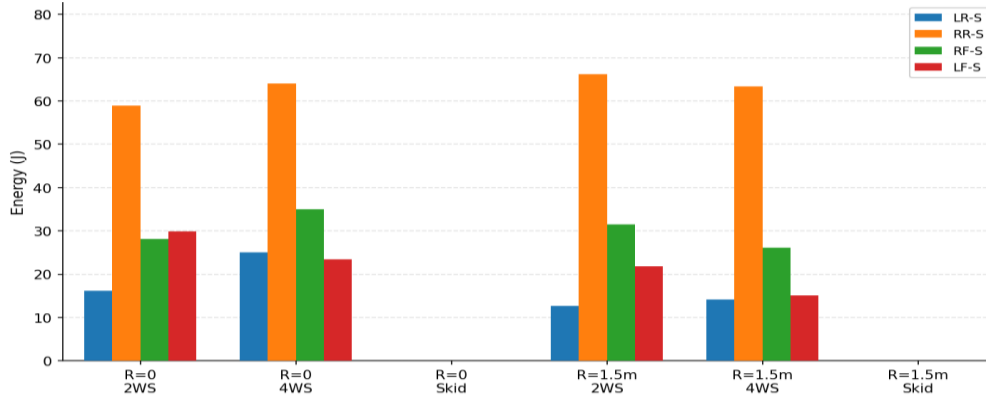


Figure 6-18 Steer motor energy by mode - B400 x L600 platform

Figure 6-18 shows steer motor energy, zero for Skid Steer modes and ranging from 11.1% to 14.9% for wheel-steered modes - higher than B400×L800 (6.5 - 11.9%). The elevated steer fraction reflects the increased steering effort required on the shorter-wheelbase B400×L600 configuration.

In summary, the experimental data demonstrate that R=0 2WS (895.07 J) is the most energy-efficient valid steering strategy for the B400×L600 platform. The R=0 Skid Steer mode cannot be executed on this platform; skid-steer operation should be avoided or restricted in energy-sensitive applications.

6.2.1.6 B600×L800 Robot Platform Data Analysis

This section presents the energy consumption data for the B600×L800 platform (wheelbase 800 mm, track width 600 mm, B/L = 0.75) on flat terrain. This configuration has the same wheelbase (L = 800 mm) as the reference B400×L800 (Section 6.2.1.4) but a wider track (B = 600 mm vs 400 mm). The analysis examines the effect of widening the track at a constant, long wheelbase within the $B < L$ group.

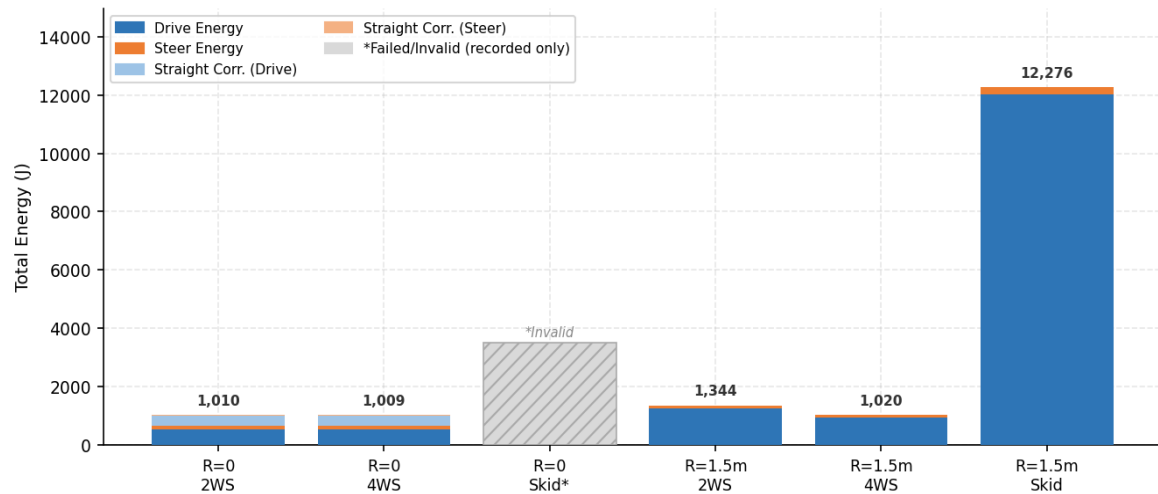


Figure 6-19 Total energy by steering mode - B600 x L800 platform (blue=drive, orange=steer, light=straight correction, grey hatching=failed/invalid)

Figure 6-19 shows total energy for B600×L800. Compared to B400×L800 (Section 6.2.1.4), energy profiles are broadly similar: R=1.5m 4WS remains optimal at 1,020.11 J (vs 990.58 J for B400×L800, a 3% difference), confirming that the shared long wheelbase ($L = 800$ mm) governs turning energy more than track width. R=0 Skid Steer also failed on B600×L800.

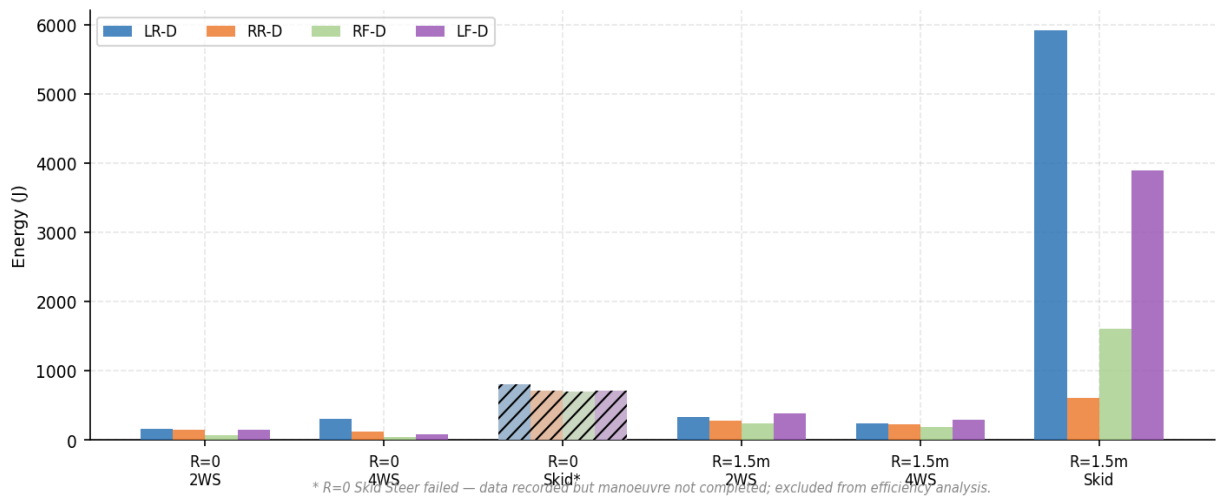


Figure 6-20 Drive motor energy by mode - B600 x L800 platform

Figure 6-20 shows per-drive-motor energy for B600×L800. Load distribution is qualitatively consistent with B400×L800, with left-side dominance in R=1.5m arc modes. The wider track width ($B = 600$ mm vs 400 mm) results in moderately higher individual motor energy in R=0 pivot modes due to increased lateral spacing.

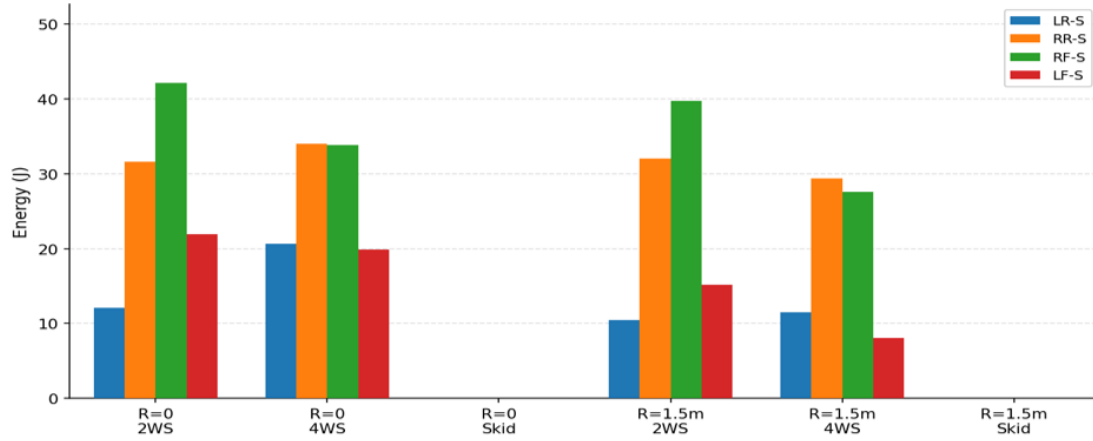


Figure 6-21 Steer motor energy by mode - B600 x L800 platform

Figure 6-21 shows steer motor energy, zero for Skid Steer modes and ranging from 7.2% to 10.7% for wheel-steered modes - comparable to B400×L800 (6.5 - 11.9%). The similarity confirms that varying track width B has minimal effect on steer motor contribution when wheelbase L is held constant.

In summary, the experimental data demonstrate that R=0 4WS (1,008.90 J) is the most energy-efficient valid steering strategy for the B600×L800 platform. The R=0 Skid Steer mode cannot be executed on this platform; skid-steer operation should be avoided or restricted in energy-sensitive applications.

6.2.1.7 B800×L400 Robot Platform Data Analysis

This section presents a comprehensive analysis of the energy consumption characteristics of the B800×L400 robot platform (wheelbase 400 mm, track width 800 mm) operating on flat terrain ($D = 0^\circ$). The straight-drive correction totals 364.46 J, apportioned according to the measured straight-line power ratio into 343.21 J for drive motors (94.2%) and 21.25 J for steer motors (5.8%), applied to R=0 modes only; the R=1.5 m arc paths already contain the required displacement. For the R1.5 skid-steering test, the vehicle achieved the intended turning performance, with a measured turning radius of approximately 1.5 m.

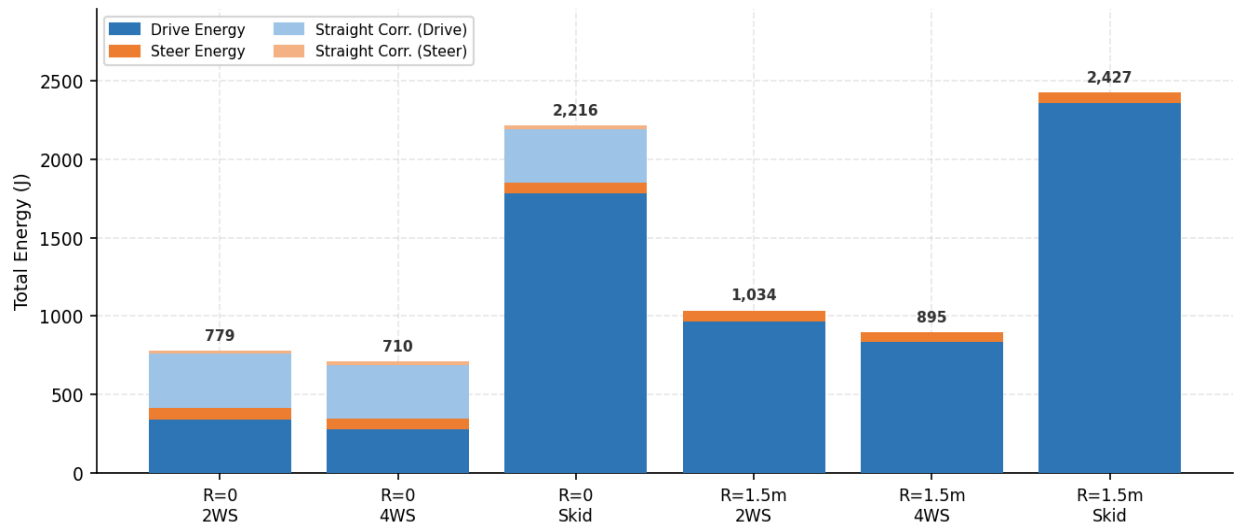


Figure 6-22 Total Energy by steering mode - B800 x L400 platform (blue=drive, orange=steer, light=straight correction, grey hatching=failed/invalid)

Figure 6-22 reveal the energy efficiency hierarchy across all valid steering modes. Ranked from lowest to highest total energy: 1st: R=0 4WS (709.57 J); 2st: R=0 2WS (778.74 J); 3st: R=1.5m 4WS (894.87 J); 4st: R=1.5m 2WS (1,034.45 J); 5st: R=0 Skid (2,215.98 J); 6st: R=1.5m Skid (2,426.99 J). Skid Steer modes consume substantially more energy than wheel-controlled steering due to sustained wheel-slip converting drive energy into frictional heat. Among valid modes, R=0 4WS at 709.57 J is the most efficient; R=1.5m Skid at 2,426.99 J is the least efficient. Steer motor energy constitutes only 2.8-9.7% of the total in all modes, confirming that steering actuation is not a primary energy driver.

Figures 6-23-6-24 present per-motor energy data for all six modes, providing a quantitative basis for drive-force distribution analysis.

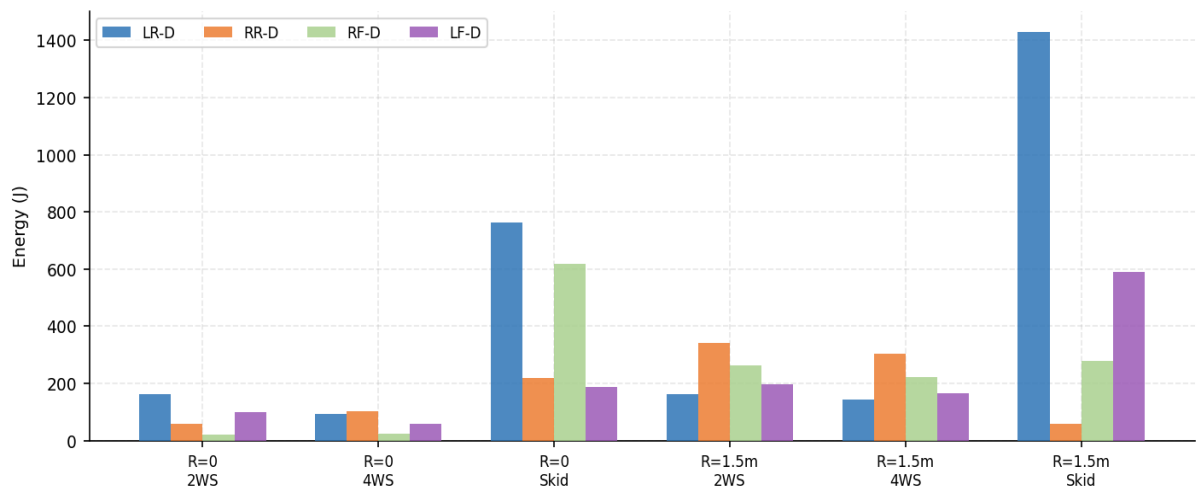


Figure 6-23 Drive motor energy by mode - B800 x L400 platform

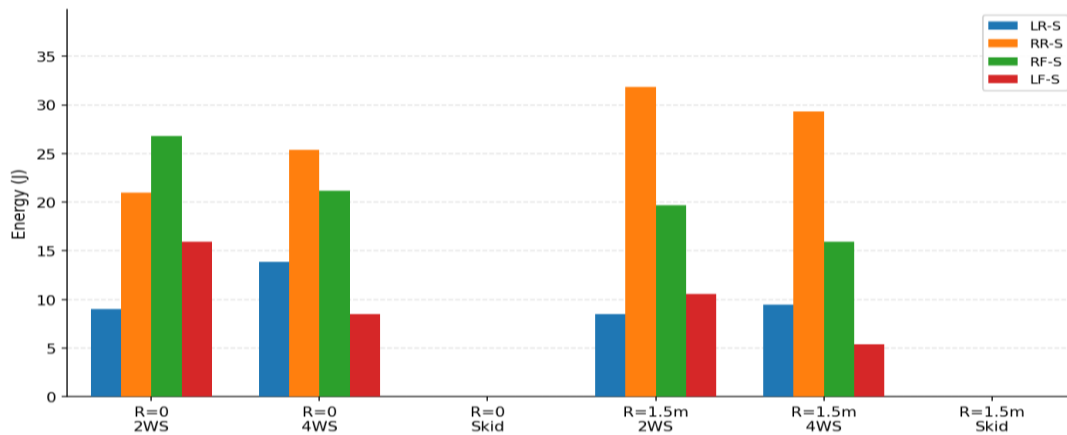


Figure 6-24 Steer motor energy by mode - B800xL400 platform

For R=0 Skid, the LR-D + RF-D diagonal pair carries 77.3% of drive energy, compared with 22.7% for the opposing pair. For R=1.5m 2WS, the right-side pair (RR-D + RF-D) absorbs 62.7% of drive energy versus 37.3% for the opposite side. For R=1.5m 4WS, the right-side pair (RR-D + RF-D) absorbs 63.2% of drive energy versus 36.8% for the opposite side. A notable difference in the R1.5 turning test was that the right-side motors exhibited higher energy consumption than the left-side motors. This test was conducted in a counterclockwise direction (continuous left turn), unlike the previous manoeuvres. The observed energy distribution is consistent with the greater energy demand experienced by the motors located on the outside of the turn. For R=1.5m Skid, the LR-D + RF-D diagonal pair carries 72.4% of drive energy, compared with 27.6% for the opposing pair. This diagonal dominance is characteristic of skid-steer kinematics, where counter-rotating wheel pairs generate the turning moment.

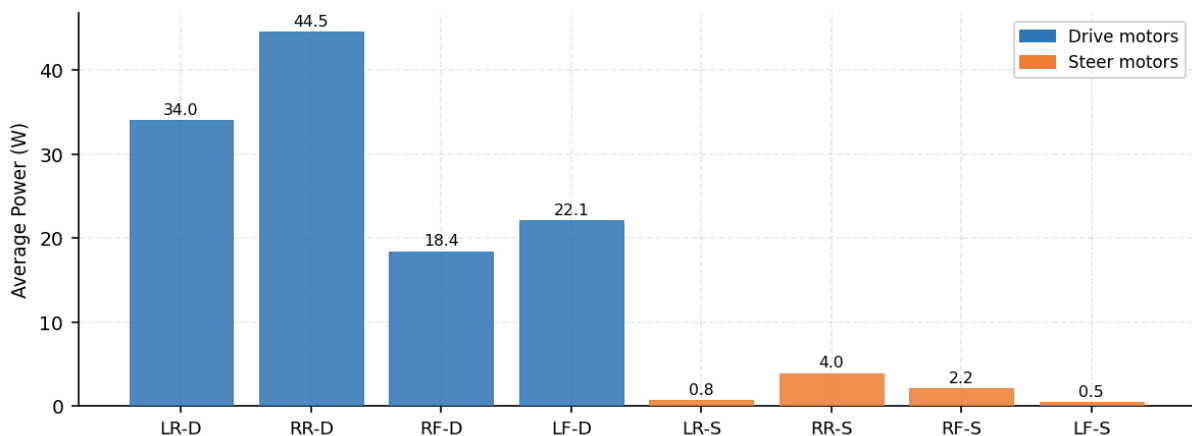


Figure 6-25 Straight-drive average power per motor - B800xL400 platform

During straight-line travel shown in Figure 6-25, the overall average power is 126.57 W, of which drive motors consume 119.19 W (94.2%) and steer motors 7.38 W (5.8%). The rear-axle pair consumes 65.9% of total drive power, indicating a rearward longitudinal weight bias under

steady-state rolling conditions. Lateral load distribution is well balanced (left: 47.1%, right: 52.9%).

In summary, the experimental data demonstrate that R=0 4WS (709.57 J) is the most energy-efficient valid steering strategy for the B800×L400 platform. Skid Steer modes impose a substantial energy penalty.

6.2.1.8 B600×L400 Robot Platform Data Analysis

This section presents the energy consumption data for the B600×L400 platform (wheelbase 400 mm, track width 600 mm, B/L = 1.50) on flat terrain. Together with B800×L400 (B/L = 2.00, Section 6.2.1.7) and B800×L600 (B/L = 1.33, Section 6.2.1.9), this forms the B > L group, ordered by decreasing B/L ratio. The analysis compares with the B800×L400 reference, examining the effect of narrowing the track from 800 mm to 600 mm at the same short wheelbase.

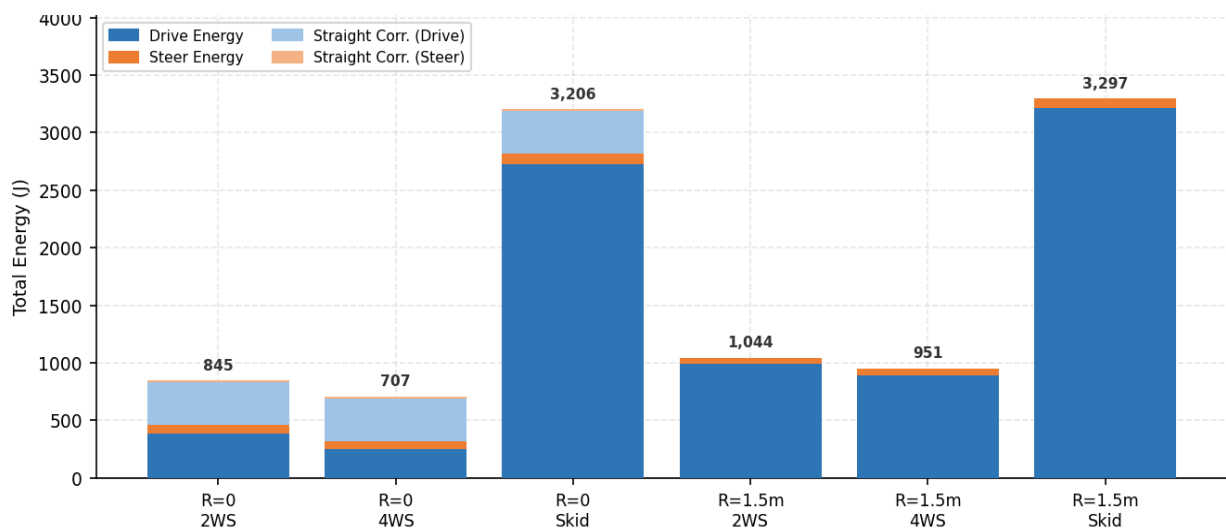


Figure 6-26 Total energy by steering mode - B600 x L400 platform (blue=drive, orange=steer, light=straight correction, grey hatching=failed/invalid)

Figure 6-26 shows total energy for B600×L400. The energy hierarchy closely mirrors B800×L400 (Section 6.2.1.7): R=0 4WS is optimal at 706.95 J - virtually equal to B800×L400's 709.57 J - confirming that the short wheelbase (L = 400 mm) dominates pivot-mode energy in the B > L group regardless of track width.

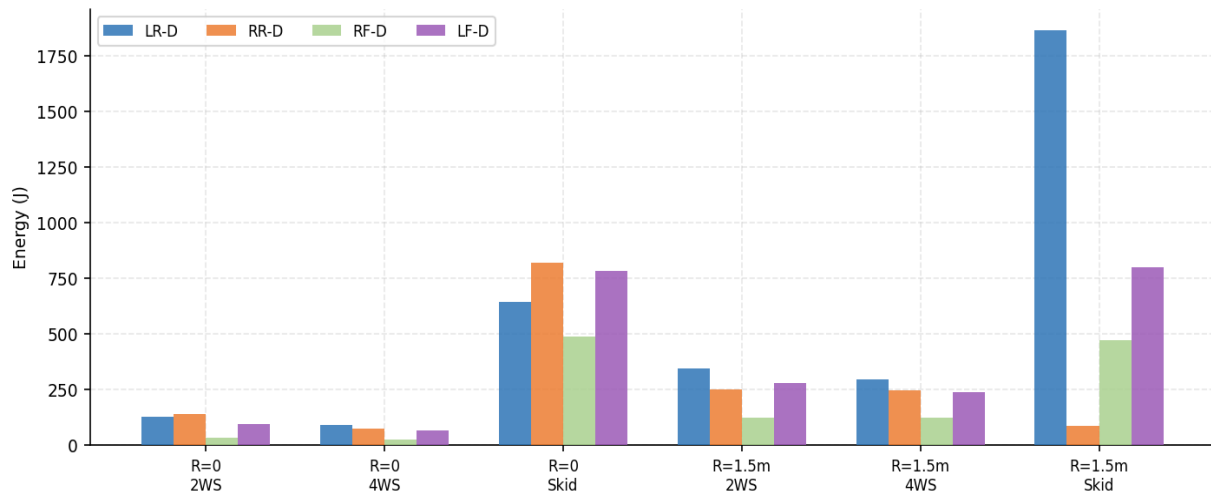


Figure 6-27 Drive motor energy by mode - B600 x L400 platform

Figure 6-27 shows per-drive-motor energy for B600×L400. Distribution is qualitatively consistent with B800×L400: right-side dominance (RR-D + RF-D) in R=1.5m arc modes reflects the clockwise turning geometry, and the LR-D + RF-D diagonal pair dominates in R=1.5m Skid. Per-motor absolute values are close to B800×L400.

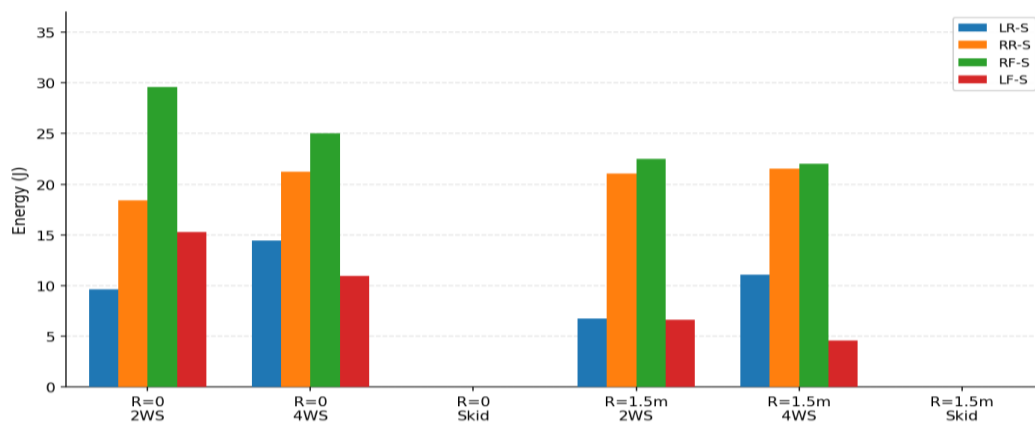


Figure 6-28 Steer motor energy by mode - B600 x L400 platform

Figure 6-28 shows steer motor energy, zero for Skid Steer and ranging from 5.4% to 10.1% for wheel-steered modes - closely matching B800×L400 (6.7 - 9.7%). The near-identical steer fraction confirms that track width B has negligible influence on steer motor contribution.

In summary, the experimental data demonstrate that R=0 4WS (706.95 J) is the most energy-efficient valid steering strategy for the B600×L400 platform. Skid Steer modes impose a substantial energy penalty and skid-steer operation should be avoided or restricted in energy-sensitive applications.

6.2.1.9 B800×L600 Robot Platform Data Analysis

This section presents the energy consumption data for the B800×L600 platform (wheelbase 600 mm, track width 800 mm, $B/L = 1.33$) on flat terrain. This configuration shares the same track width ($B = 800$ mm) as the reference B800×L400 (Section 6.2.1.7) but has a longer wheelbase ($L = 600$ mm vs 400 mm). The analysis examines how increasing wheelbase at constant track width affects energy consumption within the $B > L$ group.

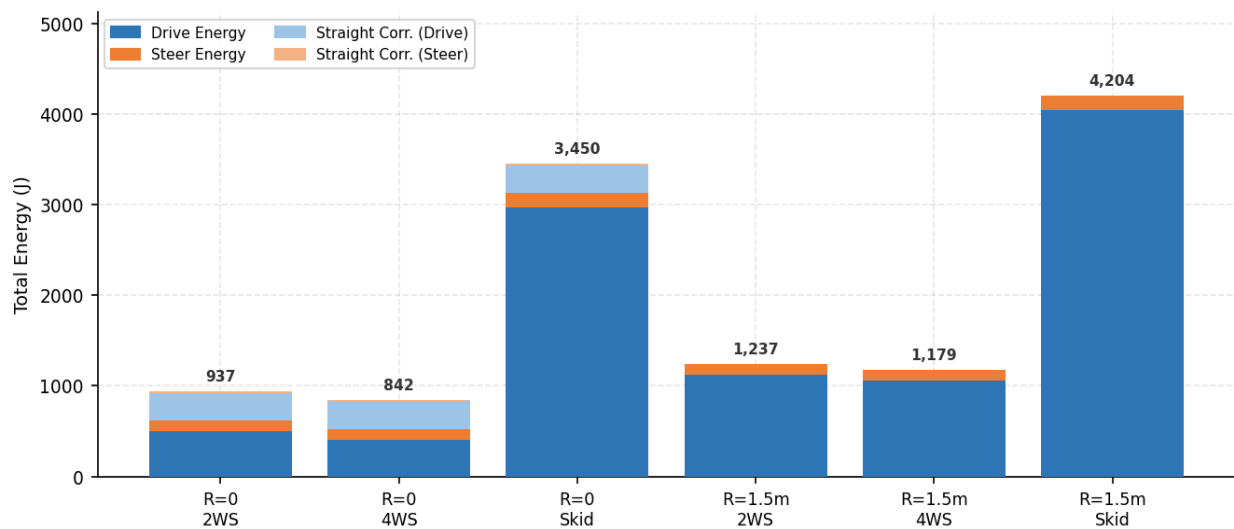


Figure 6-29 Total energy by steering mode - B800×L600 platform (blue=drive, orange=steer, light=straight correction; grey hatching=failed/invalid)

Figure 6-29 shows total energy for B800×L600. Compared to B800×L400 (Section 6.2.1.7), all modes show substantially higher energy due to the longer wheelbase ($L = 600$ mm vs 400 mm): R=0 4WS increases from 709.57 J to 842.39 J (+18.7%), and R=1.5m Skid from 2,360.24 J to 4,044.96 J (+71.4%). This directly demonstrates the dominant influence of L on turning energy across all steering modes.

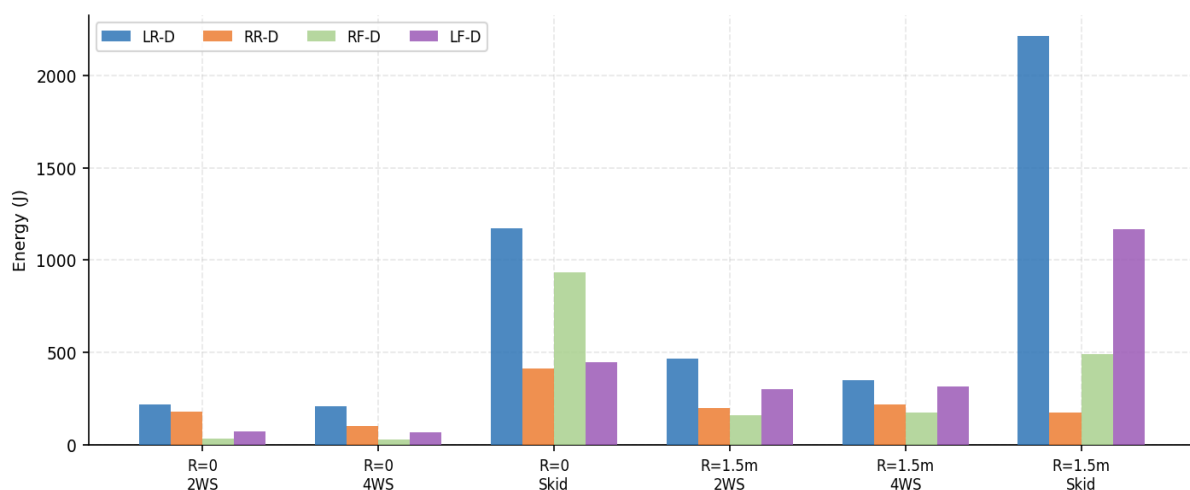


Figure 6-30 Drive motor energy by mode - B800×L600 platform

Figure 6-30 shows per-drive-motor energy for B800×L600. As with B800×L400, the left-side pair dominates in R=1.5m arc modes and the LR-D + RF-D diagonal pair dominates in R=1.5m Skid. Per-motor absolute values are proportionally higher than B800×L400 due to the greater energy demand of the longer wheelbase.

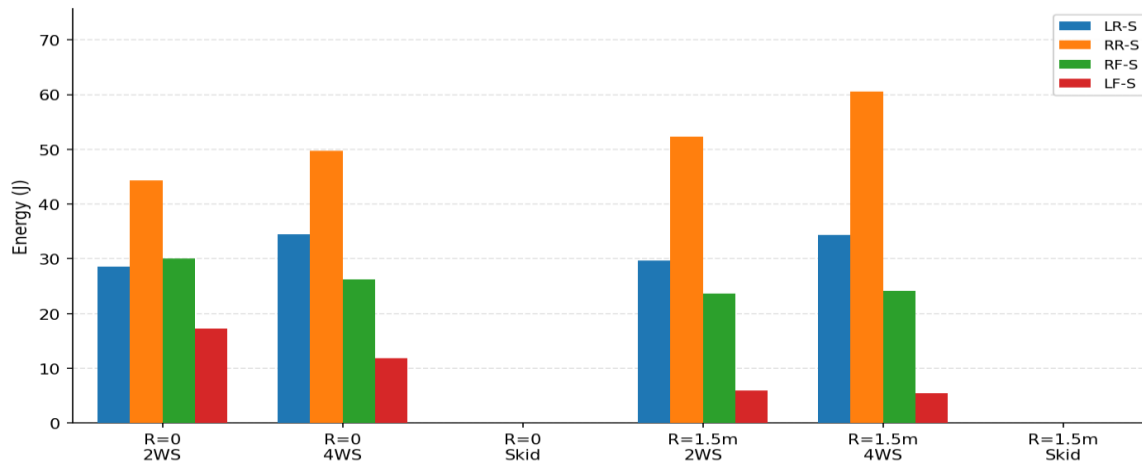


Figure 6-31 Steer motor energy by mode - B800×L600 platform

Figure 6-31 shows steer motor energy, zero for Skid Steer modes and ranging from 9.0% to 14.5% for wheel-steered modes - higher than B800×L400 (6.7 - 9.7%). The elevated steer fraction for L = 600 mm reflects the greater steering actuation effort required for pivot and arc manoeuvres on the longer-wheelbase configuration.

In summary, the experimental data demonstrate that R=0 4WS (842.39 J) is the most energy-efficient valid steering strategy for the B800×L600 platform. Skid Steer modes impose a substantial energy penalty and skid-steer operation should be avoided or restricted in energy-sensitive applications.

6.2.2 Cross-Configuration Energy Consumption Analysis for Flat Ground

This section presents a systematic, cross-configuration energy consumption analysis of a four-wheel-independent-steering agricultural robot platform operating on flat terrain (inclination $D = 0^\circ$).

Among the 54 valid workloads examined, the B600×L400 and B800×L400 configurations in R=0 4WS mode achieved the global minimum energy of approximately 707-710 J, consistently demonstrating that short wheelbase ($L = 400$ mm) is the dominant factor in minimising energy consumption across all steering modes. Skid Steer modes impose energy penalties of 3-25× relative to the optimal equivalent mode, with R=1.5 m Skid Steer being the least efficient workload across all configurations. Five R=0 Skid Steer tests (B600×L600, B400×L600,

B400×L400, B600×L800, B400×L800) failed to complete the manoeuvre and their energy data are recorded for reference only, excluded from comparative analysis.

Correlation analysis reveals that wheelbase L exhibits the strongest positive correlation with energy consumption across most steering modes (Pearson $r = 0.80-0.90$). The B/L ratio shows strong negative correlation for R=0 4WS ($r = -0.85$) and R=1.5 m Skid Steer ($r = -0.67$). Track width B alone has negligible independent correlation. A quadratic bivariate regression model $E = f(B, L)$ achieves $R^2 = 0.70-0.98$ across modes, with the highest fidelity for R=0 2WS ($R^2 = 0.982$). Simplified engineering formulae are derived for each steering mode, enabling energy estimation from geometric parameters during the design stage.

6.2.2.1 Platform Configurations and Steering Modes

Nine platform configurations were constructed by combining three values of track width $B \in \{400, 600, 800 \text{ mm}\}$ with three values of wheelbase $L \in \{400, 600, 800 \text{ mm}\}$, yielding a full 3×3 factorial design. Each configuration is denoted B{B}×L{L}; for example, B600×L400 designates B = 600 mm, L = 400 mm. All tests were conducted on flat terrain (inclination angle $D = 0^\circ$) to eliminate slope as a confounding variable.

Table 6.2 Nine Platform Configurations in the 3×3 Factorial Design

Config	B (mm)	L (mm)	B/L Ratio	L ² /B (mm)	Straight Power (W)
B800×L800	800	800	1.00	800	124.73
B600×L800	600	800	0.75	1067	127.14
B400×L800	400	800	0.50	1600	134.02
B800×L600	800	600	1.33	450	110.58
B600×L600	600	600	1.00	600	129.58
B400×L600	400	600	0.67	900	126.53
B800×L400	800	400	2.00	200	126.57
B600×L400	600	400	1.50	267	134.09
B400×L400	400	400	1.00	400	113.21

Six distinct steering modes are evaluated. R=0 modes involve pivot manoeuvres at zero turning radius (in-place rotation), while R=1.5 m modes involve arc-path manoeuvres along a 1.5 m radius curve. Within each radius group, three steering strategies are tested: Two-Wheel Steering (2WS, front axle only), Four-Wheel Steering (4WS, all four wheels coordinated), and Skid Steer (differential wheel speed, no dedicated steer actuation). Energy is computed as $E = \sum |I| \cdot V \cdot \Delta t$ over the complete recorded manoeuvre.

For R=0 pivot modes, a straight-drive correction, which is shown in Table 6.2, is applied to account for the net travel distance that would be required in a real navigation scenario. The correction is computed as $C = P_s \times t_{\text{cross}}$, where P_s is the measured straight-line average total power and $t_{\text{cross}} = 2.8798$ s is the time to traverse one body width plus the full tree-wall width at fastest travel speed. The correction is allocated to drive and steer sub-components proportionally to the straight-drive power split. R=1.5 m arc modes inherently incorporate forward displacement and require no correction.

6.2.2.2 Overall Energy Ranking - Platform Configuration Comparison

For each platform configuration, the minimum total energy across all valid steering modes is identified as the representative "best achievable" energy. This reflects the configuration's optimal performance, agnostic of which specific manoeuvre achieves it.

Table 6.3 Overall Energy Ranking - Best Valid Mode per Configuration

Rank	Configuration	B (mm)	L (mm)	B/L	Best Energy (J)	Achieved by
1	B600×L400	600	400	1.50	706.95	R=0 4WS
2	B800×L400	800	400	2.00	709.57	R=0 4WS
3	B400×L400	400	400	1.00	807.88	R=0 2WS
4	B800×L600	800	600	1.33	842.39	R=0 4WS
5	B400×L600	400	600	0.67	895.07	R=0 2WS
6	B400×L800	400	800	0.50	950.76	R=0 2WS
7	B600×L600	600	600	1.00	974.82	R=0 2WS
8	B600×L800	600	800	0.75	1,008.90	R=0 4WS
9	B800×L800	800	800	1.00	1,021.62	R=1.5m 4WS

The most striking finding from Table 6.3 is that the top three configurations all share $L = 400$ mm. The two globally optimal configurations ($B600 \times L400$ and $B800 \times L400$) achieve nearly identical best energies of ~ 707 - 710 J via $R=0$ 4WS, representing a 31% reduction compared to the worst configuration ($B800 \times L800$, 1,022 J). This establishes short wheelbase as the primary geometric driver of energy efficiency.

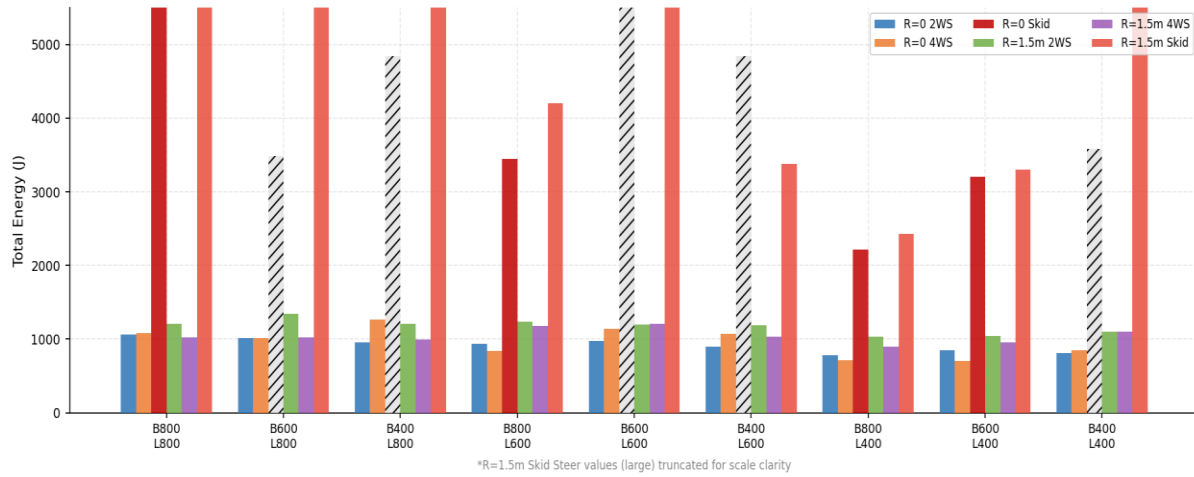


Figure 6-32 Figure Energy consumption across all configurations and steering modes. Grey hatching denotes failed $R=0$ Skid Steer tests (excluded from analysis). Skid Steer bars for $R=1.5$ m are partially truncated; actual values are listed in the text and tables.

Figure 6-32 presents the complete picture of energy consumption across all nine configurations and all six steering modes. Several important observations emerge from this aggregated view.

First, the $R=0$ Skid Steer and $R=1.5$ m Skid Steer modes are dramatically more energy-intensive than the wheel-controlled steering modes (2WS and 4WS). This is a universal finding across all configurations, attributable to the sustained wheel-ground slip that converts drive energy into frictional heat rather than useful rotational motion. Second, within the wheel-controlled modes, the ordering of configurations is remarkably consistent: $L=400$ mm configurations consistently occupy the low-energy region, regardless of track width B . Third, $R=1.5$ m Skid Steer for $B400 \times L800$ (23,847 J) is by far the worst single workload in the dataset, representing a $34\times$ energy penalty relative to the global optimum.

6.2.2.3 Optimal Configuration by Steering Mode

Different steering modes impose different load patterns on the vehicle, corresponding to distinct vehicle design philosophies. Two-wheel steering favours narrow turning circles achieved by front-wheel pivoting, while four-wheel steering achieves tighter in-place pivots through coordinated four-wheel geometry. Skid steer eliminates dedicated steering actuation at

the cost of high wheel slip. A designer choosing a steering mode first should then select the B×L configuration that minimises energy within that mode.



Figure 6-33 Energy ranking by steering mode - configurations sorted from most to least efficient. Asterisked grey bars represent failed/invalid R=0 Skid Steer tests.

For $r=0$ Skid Steer, only four configurations with valid data are included (B800×L800, B800×L600, B800×L400, B600×L400); five configurations failed to complete the manoeuvre. As shown in Figure 6-33, the data reveal a clear pattern: B800×L400 and B600×L400 rank first or second in every single steering mode - both for pivot and arc-path manoeuvres. This remarkable cross-mode consistency confirms that the L=400 mm short-wheelbase geometry is universally advantageous. Within the L=400 mm group, B=600 or B=800 mm is preferred over B=400 mm for 4WS modes, while B=800 mm leads for 2WS modes, reflecting how track width influences load distribution during coordinated steering.

6.2.2.4 Energy Trends - Influence of B, L and B/L

Wheelbase L is the single geometric parameter most strongly correlated with energy consumption across all tested modes. Table 6.4 and Figure 6-34 summarise the Pearson correlation coefficients.

Table 6.4 Pearson Correlation: Energy vs Geometric Variables (all valid data)

Steering Mode	r vs B	r vs L	r vs B/L	r vs L ² /B	Dominant Variable
R=0 2WS	+0.193	+0.903	-0.572	+0.582	L (strong positive)
R=0 4WS	-0.407	+0.802	-0.854	+0.865	B/L & L ² /B (strong)
R=0 Skid*	+0.216	+0.928	-0.913	+0.975	L ² /B (very strong)
R=1.5m 2WS	-0.025	+0.849	-0.675	+0.649	L (strong)
R=1.5m 4WS	-0.043	+0.123	-0.306	-0.035	Nondominant (R ² <0.1)
R=1.5m Skid	-0.453	+0.642	-0.667	+0.863	L ² /B (strong)

For R=0 2WS and R=1.5m 2WS, L has a strong linear positive relationship with energy ($r = 0.90$ and 0.85 , respectively). As wheelbase increases from 400 mm to 800 mm, the average energy in these modes rises by approximately 35-40%.

For R=0 4WS, the compound variable L²/B emerges as a better predictor ($r = +0.865$) than L alone ($r = +0.802$), indicating that wider track width B partially compensates for the penalty of longer wheelbase. This is consistent with four-wheel-coordinated pivoting where the B/L ratio governs the geometric efficiency of the manoeuvre - a wider, shorter platform pivots more cleanly.

R=1.5m 4WS is the only mode where no single geometric variable dominates (maximum $|r| = 0.31$). One hypothesis is: arc-path four-wheel steering involves a complex interplay of inner-wheel constraint, outer-wheel trajectory, and four-wheel coordination geometry that does not reduce to a simple function of B or L alone. The scatter in energy values across configurations is lower for this mode ($CV \approx 14\%$), suggesting that four-wheel arc steering is relatively configuration insensitive.

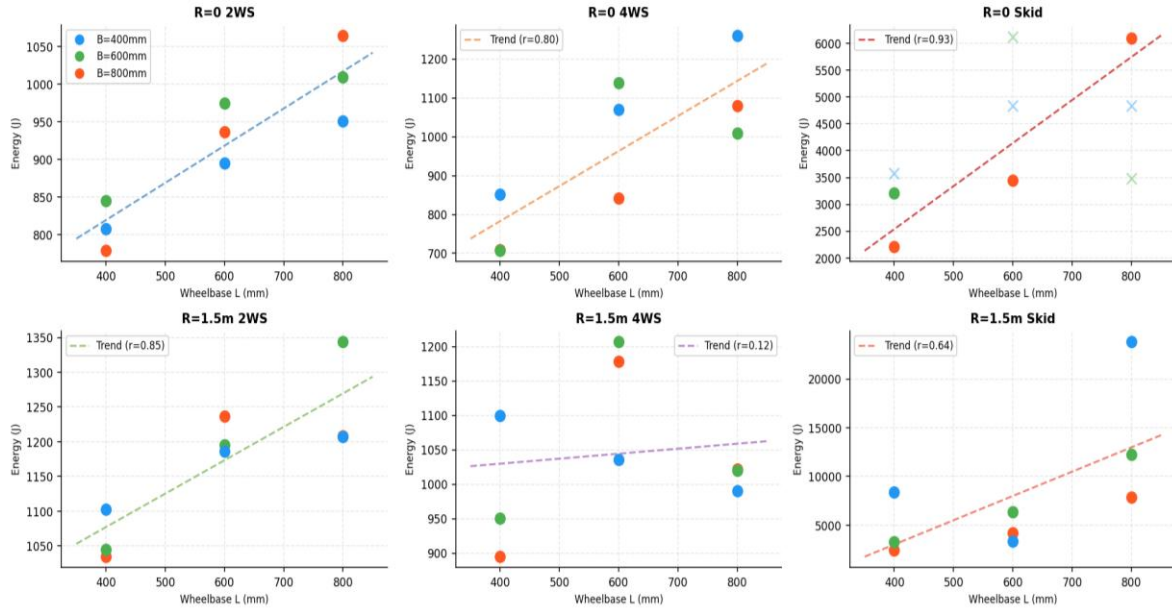


Figure 6-34 Energy consumption versus wheelbase L for all six steering modes. Marker colour indicates track width B . Dashed lines are linear trend fits (r values shown in legend). Failed $R=0$ Skid Steer tests are shown as crosses ($\times$) and excluded from fits.

Track width B has weak independent correlation with energy when considered alone ($|r| \leq 0.45$ across all modes). This does not imply that B is irrelevant, but rather that its effect is mediated through the B/L ratio. Within any fixed value of L , however, a clear pattern emerges wider B tends to reduce energy in 4WS modes but may slightly increase or have negligible effect in 2WS modes. For $L=400$ mm specifically, the progression $B=600 \rightarrow B=800$ produces a negligible change in $R=0$ 4WS energy (706.95 J vs 709.57 J, a difference of less than 0.4%), while $B=400$ is substantially worse at 851.30 J - suggesting a non-linear response with a minimum near $B=600-800$ mm for short-wheelbase 4WS.

The B/L ratio is a dimensionless geometric index that captures the "aspect ratio" of the platform footprint. For $R=0$ 4WS, B/L has a strong negative correlation with energy ($r = -0.854$): more "square" or "wide-and-short" platforms are intrinsically more energy-efficient for this manoeuvre. This finding has direct implications for vehicle design: a robot intended for frequent in-place manoeuvring in constrained environments should prioritise a high B/L ratio (i.e., wider track and shorter wheelbase).

The variable L^2/B is physically interpretable as the moment arm of the pivot constraint scaled by wheelbase squared, and represents the dominant predictor for skid steer modes ($r = +0.863-0.975$). The strong dependence on L^2 explains the catastrophic energy penalty for $B400 \times L800$ in $R=1.5m$ Skid Steer (23,847 J): doubling L quadruples this predictor variable, driving energy consumption to extreme levels.

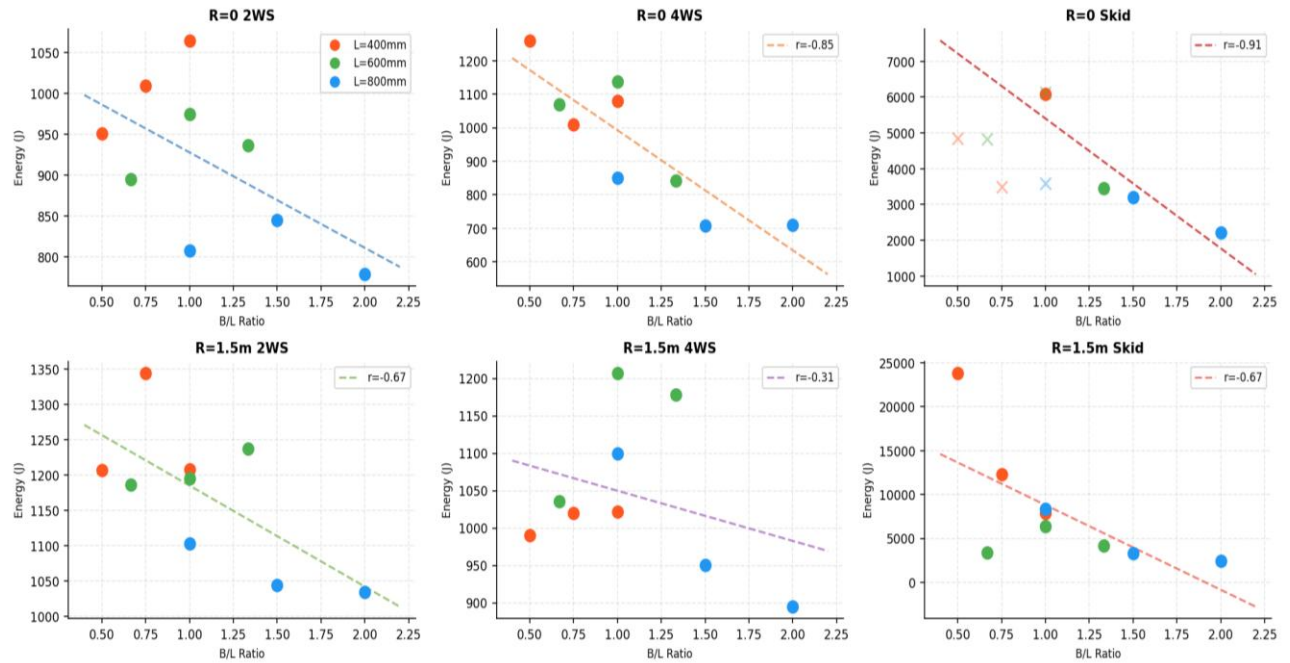


Figure 6-35 Energy consumption versus B/L ratio for all six steering modes. Marker colour indicates wheelbase L

in Figure 6-35, a negative slope energy decreasing with B/L is most pronounced for Skid Steer modes.

The actual turning diameter produced by $R = 1.5$ m skid-steer arc manoeuvres was measured in the field for all nine configurations and converted to radius (radius = diameter $\div$ 2), as summarised in Table 6.5. Only the B800 $\times$ L400 configuration achieved the theoretical target ($R = 1.50$ m). All other configurations deviated substantially, with actual radii ranging from 2.00 m (B800 $\times$ L600, +33%) to 15.00 m (B400 $\times$ L800, +900%).

Table 6-A Actual $R = 1.5$ m skid-steer turning radius vs theoretical target.

Configuration	B/L	L/B (mm)	Measured Diam. (m)	Actual Radius (m)	Deviation from 1.5 m	R=0 Skid Steer
B800$\times$L400	2.00	200	3.0	1.50	0% (target met)	Valid
B800 $\times$ L600	1.33	450	4.0	2.00	+33%	Valid
B600 $\times$ L400	1.50	267	5.5	2.75	+83%	Valid
B800 $\times$ L800	1.00	800	7.5	3.75	+150%	Valid
B600 $\times$ L600	1.00	600	8.0	4.00	+167%	Failed
B400 $\times$ L400	1.00	400	12.0	6.00	+300%	Failed
B600 $\times$ L800	0.75	1067	15.0	7.50	+400%	Failed
B400 $\times$ L600	0.67	900	18.0	9.00	+500%	Failed
B400 $\times$ L800	0.50	1600	30.0	15.00	+900%	Failed

As shown in Figure 6-36, the actual turning radius is well described by the L/B ratio across all nine configurations (Pearson $r = +0.975$, $R^2 = 0.951$), with the linear relationship:

$$R_{\text{actual}} \approx 9.02 \cdot (L/B) - 4.05 \quad (R^2 = 0.95, R \text{ in m})$$

This relationship is physically intuitive: L/B governs the balance between the resisting ground-friction moment (proportional to L) and the available differential driving moment (proportional to B). As L/B increases - the platform becomes longer and narrower - the resistance moment increasingly dominates, forcing a much larger actual arc than commanded. B800×L400 has the lowest L/B in the test matrix (0.50), which is why it is the only configuration to satisfy the kinematic target. These findings are consistent with the energy results: configurations with high L/B deviate most from the target path and simultaneously incur the highest skid-steer energy penalties.

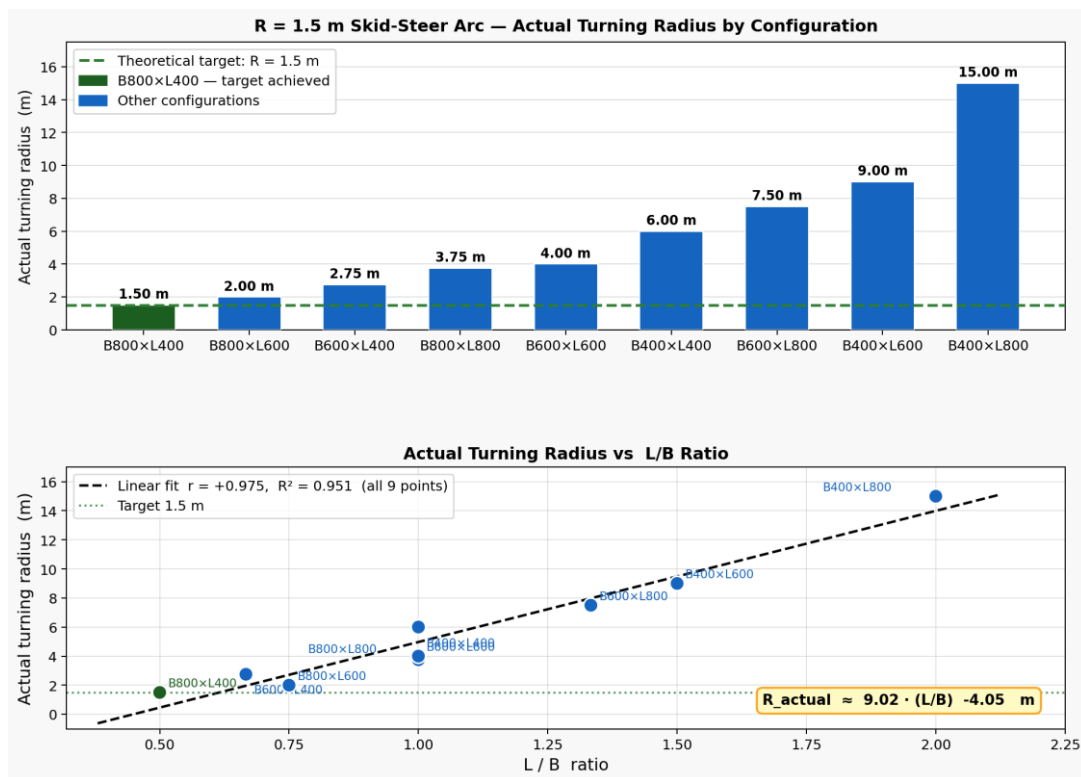


Figure 6-36 Actual $R = 1.5 \text{ m}$ skid-steer turning radius by configuration (top) and linear regression against L/B ratio (bottom).

Straight-line average power, shown in Figure 6-37, ranges from 110.6 W (B800×L600) to 134.1 W (B600×L400), a spread of approximately 21%. No statistically significant correlation exists between straight-line power and B , L , or their ratios (maximum $|r| = 0.21$, $p > 0.58$). This suggests that rolling resistance in straight-line travel is determined primarily by surface-specific conditions, tyre properties, and mass distribution rather than platform geometry per se. The highest power observed for B600×L400 (134.1 W) may partly reflect locally elevated ground resistance during that test run.

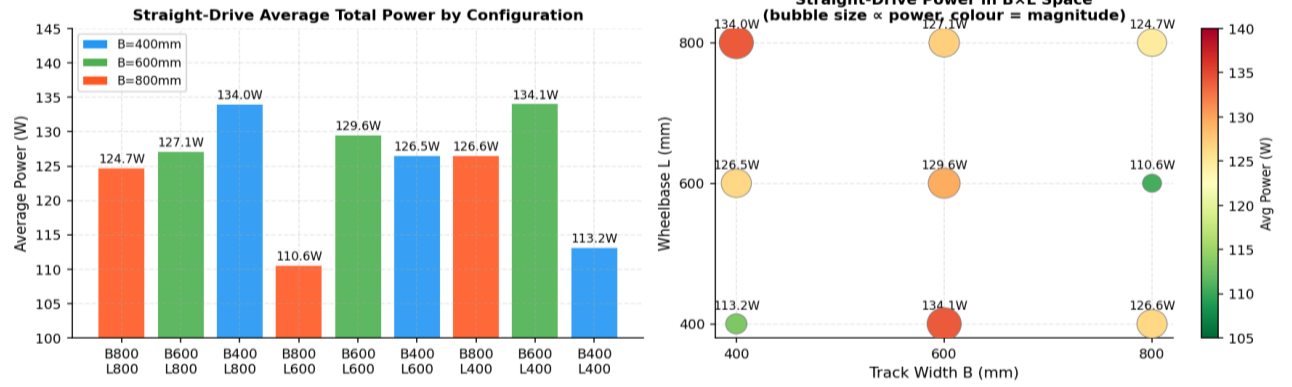


Figure 6-37 Straight-line drive average power by configuration. Left: bar chart coloured by track width B. Right: bubble plot in B×L space (size and colour proportional to power). No significant geometric trend is observed.

6.2.2.5 Regression Modelling - Energy as a Function of B and L

This chapter derives empirical formulae expressing total manoeuvre energy E (J) as a function of track width B (mm) and wheelbase L (mm). With nine data points per mode (four for R=0 Skid Steer after excluding failed tests), a parsimony-versus-fit trade-off must be balanced. Three model families are evaluated:

Model 1 (Bilinear): $E = \alpha + \beta_1 \cdot B + \beta_2 \cdot L$

Model 2 ($L + B/L$): $E = \alpha + \beta_1 \cdot L + \beta_2 \cdot (B/L)$

Model 3 (L^2/B): $E = \alpha + \beta \cdot (L^2/B)$

Model 4 (Quadratic): $E = \alpha + \beta_1 B + \beta_2 L + \beta_3 B^2 + \beta_4 L^2 + \beta_5 \cdot (B \cdot L)$

Model 4 saturates the degrees of freedom (6 parameters, 9 or 4 data points) and is provided as a goodness-of-fit upper bound. Models 1-3 are more interpretable for engineering use. The R^2 values for all models are tabulated below, and the quadratic model is depicted in Figure 6.46 (energy vs geometric).

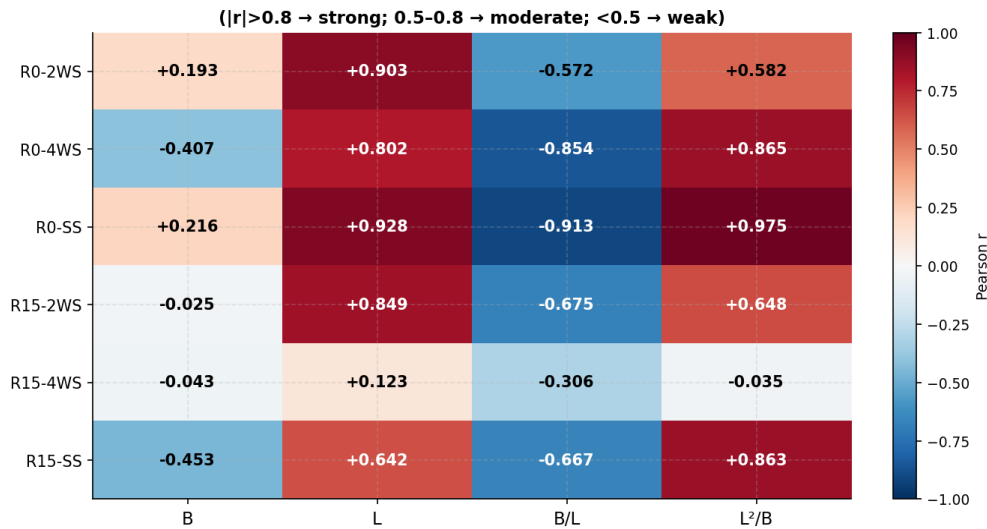


Figure 6-38 Pearson correlation matrix: energy vs geometric variables across all steering modes. Colour indicates direction and magnitude (blue = negative correlation, red = positive). Bold values indicate $|r| > 0.6$.

In Figure 6-38, the bilinear model $E = \alpha + \beta_1 \cdot B + \beta_2 \cdot L$ provides an interpretable first approximation. It achieves adequate R^2 values for R=0 2WS (0.853) and R=1.5m 2WS (0.721), but is less effective for modes where the B/L interaction dominates. The positive coefficient for L ($\beta_2 > 0$) is consistent across all modes, confirming that longer wheelbase universally increases energy consumption. The sign of β_1 (coefficient for B) varies by mode: it is positive for 2WS modes (wider B slightly increases energy) and negative for 4WS modes (wider B reduces energy, reflecting improved steering geometry). This sign reversal motivates the inclusion of interaction or ratio terms.

The full quadratic model provides the best achievable fit within the polynomial family. Figure 6-39 shows the actual versus predicted energy for all modes using the quadratic model; most points lie close to the 1:1 ideal line, confirming reasonable predictive power. The most notable exception is R=1.5m 4WS ($R^2 = 0.70$), where the scatter of actual values does not follow a clear geometric trend - consistent with the weak correlation found for this mode in Section 5.

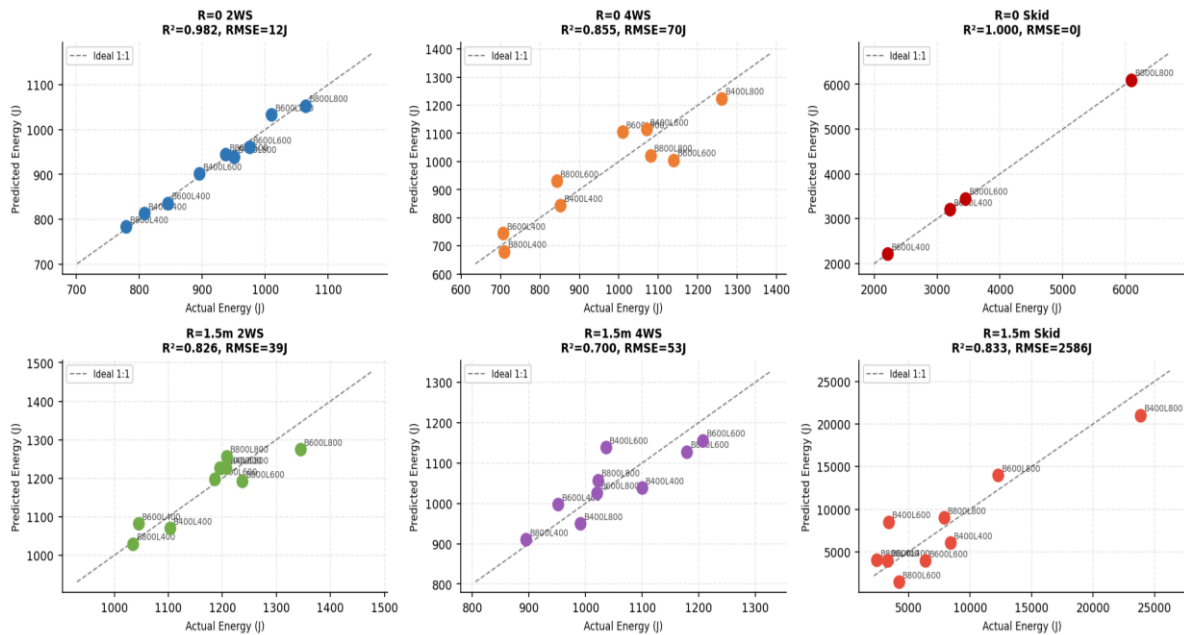


Figure 6-39 Actual vs predicted energy using quadratic regression $E = f(B, L)$ for each steering mode. The dashed 1:1 line represents perfect prediction. Configuration labels are annotated for reference.

Table 6.5 presents the recommended engineering formulae for each steering mode, selected to balance predictive accuracy and interpretability. The quadratic formula is recommended when $R^2 \geq 0.83$; simplified forms (bilinear or L^2/B) are recommended when the quadratic overfits.

Table 6.5 Summary of Regression Formulae for All Six Steering Modes (B in mm, L in mm, E in J)

Mode	Best-Fit Formula (Quadratic)	R^2 (Quad)	RMSE (J)	Simple Formula	R^2 (Simple)
R=0 2WS	$E = 349.5 + 0.6963B + 0.7413L - 0.000939B^2 - 0.000653L^2 + 0.000894 \cdot BL$	0.982	12	$E = 558 + 0.105B + 0.494L$	0.853
R=0 4WS	$E = 82.1 - 0.8353B + 3.4695L + 0.000436B^2 - 0.002019L^2 - 0.000242 \cdot BL$	0.855	70	$E = 697 - 0.458B + 0.901L$	0.809
R=0 Skid*	$E = 7.313B + 4.875L - 0.002933B^2 + 0.017605L^2 - 0.020386 \cdot BL$	1.000†	0	$E = 3002 - 6.122B + 9.692L$	0.960
R=1.5m 2WS	$E = 371 + 0.6891B + 1.6994L - 0.000803B^2 - 0.001232L^2 + 0.000433 \cdot BL$	0.826	39	$E = 893 - 0.014B + 0.481L$	0.721

Mode	Best-Fit Formula (Quadratic)	R ² (Quad)	RMSE (J)	Simple Formula	R ² (Simple)
R=1.5m 4WS	$E = 154 - 0.2287B + 3.5196L - 0.000567B^2 - 0.003609L^2 + 0.001472 \cdot BL$	0.700	53	$E \approx 1,016$ (near-constant)	0.017
R=1.5m Skid	$E = 31766 - 11.30B - 88.74L + 0.02603B^2 + 0.12597L^2 - 0.06254 \cdot BL$	0.833	2,586	$E = -1062 + 12.99 \cdot (L^2/B)$	0.745

† R=0 Skid Steer quadratic R²=1.000 results from fitting 6 parameters to only 4 valid data points (overfitting). The bilinear model (R²=0.960) is the recommended formula for this mode.

Key interpretations from Table 6.6:

- (1) R=0 2WS achieves the highest regression quality (R²=0.982 quadratic, 0.853 bilinear), confirming that two-wheel pivot energy has a clean geometric dependence on B and L. Both B and L contribute positively in the bilinear model, with L dominating (coefficient 0.494 vs 0.105).
- (2) R=0 4WS shows a negative coefficient for B in the bilinear model ($\beta_1 = -0.458$), reflecting the physical advantage of wider track for four-wheel coordinated pivoting - wider platforms generate better force balance between inner and outer wheels.
- (3) R=1.5m 4WS is the most geometrically robust mode: energy is nearly configuration-independent (R²=0.70 quadratic, 0.017 bilinear), with a near-constant mean of approximately 1,016 J. This mode is recommended when energy consistency across varying platform sizes is a design priority.
- (4) R=1.5m Skid Steer has the highest energy sensitivity to geometry, with the L²/B formula capturing the dominant trend (R²=0.745). This mode should be avoided in energy-critical applications, particularly for large-L, small-B configurations.

6.2.2.6 Discussion

The most robust finding across this study is the dominant influence of wheelbase L on energy consumption. This is consistent across five of six steering modes (the exception being R=1.5m 4WS). The quadratic dependence of Skid Steer energy on L - manifested through the L²/B predictor - is physically motivated by the moment arm of the friction forces generated during differential slip. For pivot-type manoeuvres, longer wheelbase forces the front-axle wheels to

travel longer arcs per revolution of the vehicle body, creating proportionally higher resistive moments.

From a practical design standpoint, selecting $L = 400$ mm offers a consistent 25-35% reduction in turning energy compared to $L = 800$ mm across all controlled-steering modes. This energy saving is achieved without sacrifice of vehicle functionality or payload capacity, making short wheelbase the highest-priority geometric recommendation arising from this study.

While B has weak independent influence, the B/L ratio serves as an important compound variable for 4WS-type modes. A wide, short platform (high B/L) achieves better four-wheel coordinated pivot efficiency. This suggests that for a given L , a larger B is beneficial for 4WS operation - a recommendation that must be balanced against structural, stability, and field-row-width constraints of the specific agricultural application.

The near-constant energy profile of $R=1.5$ m 4WS (mean 1,016 J, CV=14%) is a practically important finding for robot design. This mode effectively decouples turning energy from geometric configuration, allowing a designer to prioritise other parameters (stability, ground clearance, agricultural row spacing) without significantly compromising arc-path turning energy efficiency. The $R=1.5$ m 4WS energy is also competitive with the best $R=0$ modes (within 20-40% of the global minimum), confirming it as a generally efficient choice when pure in-place pivoting is not required.

Several limitations affect the generalisability of this study. First, soil conditions varied slightly across test runs, which may account for outlier points noted in individual configuration analyses (particularly $B600 \times L400$ straight-line power). Second, the failure of $R=0$ Skid Steer for five configurations reduces the dataset for that mode to four points, limiting the statistical confidence of its regression model. Third, the quadratic model satisfies the data tightly by construction but cannot extrapolate reliably beyond the tested parameter range.

6.2.2.7 Conclusions and Design Recommendations

This thesis has conducted a comprehensive cross-configuration energy analysis of a 4WIS agricultural robot across nine $B \times L$ geometric configurations and six steering modes on flat terrain. The principal conclusions are as follows:

1. Wheelbase L is the dominant geometric parameter influencing turning energy. $L = 400$ mm configurations achieve 25-35% lower energy than $L = 800$ mm equivalents across controlled-steering modes. This finding is statistically robust ($r = 0.80$ - 0.90 for most modes) and consistent across all six steering strategies.

2. The globally optimal configuration is B600×L400 or B800×L400 in R=0 4WS mode, achieving approximately 707-710 J - the minimum energy across all 54 valid workloads. Both configurations perform nearly identically, providing design flexibility in track width selection.
3. Track width B has weak independent influence but governs energy through the B/L ratio. Higher B/L (wider, shorter platform) reduces energy for 4WS modes. The bilinear coefficient for B is negative for 4WS (−0.458) and positive for 2WS (+0.105), reflecting the mode-specific role of wheel coordination geometry.
4. R=1.5m 4WS is the most geometrically robust mode, with near-constant energy (~1,016 J, CV=14%) across all configurations. It is recommended when design flexibility is required across a range of B×L values.
5. Skid Steer modes impose 3-25× energy penalties relative to the optimal equivalent mode and should be avoided in energy-critical applications. The L^2/B predictor ($r = +0.86-0.98$) captures the dominant trend for Skid Steer energy, with large-L, small-B configurations incurring catastrophic energy costs.
6. Quadratic bivariate regression models $E = f(B, L)$ achieve $R^2 = 0.70-0.98$ across the six steering modes. The highest fidelity is achieved for R=0 2WS ($R^2=0.982$) and R=0 Skid Steer ($R^2=0.960$ bilinear). R=1.5m 4WS is the only mode where geometric regression fails to explain variation meaningfully ($R^2=0.017$ bilinear).

Based on the above conclusions, the following design guidelines in Table 6.6 are proposed for future agricultural robot platform development:

Table 6.6 Design Recommendations by Operational Priority

Priority	Recommendation	Supporting Evidence
Minimise turning energy (any mode)	Select L = 400 mm (short wheelbase) as primary geometric decision	Consistent top-4 ranking for all L=400 configurations; L dominant in correlations
Optimise for in-place pivoting (R=0)	Prefer R=0 4WS over 2WS; use B = 600-800 mm with L = 400 mm	B600×L400 R=0 4WS achieves global minimum 706.95 J; negative B coefficient for 4WS
Maximise geometric robustness	Choose R=1.5m 4WS steering mode; B×L configuration less critical	CV=14% across all 9 configs; configuration-insensitive energy profile

Priority	Recommendation	Supporting Evidence
Avoid high energy penalties	Never use Skid Steer for $L > 600$ mm; strictly avoid B400×L800 with Skid Steer	B400×L800 R=1.5m Skid: 23,847 J - 34× global minimum; L^2 dependence confirmed
Estimate design-stage energy	Use bilinear formula as first approximation; refine with quadratic if B/L varies widely	Bilinear $R^2=0.853$ (R=0 2WS), 0.809 (R=0 4WS); quadratic adds $\leq 13\%$ R^2 improvement

6.3 Slope Ground Measurement Results and Data Analysis

On the 18° slope test site, safety constraints required that only full-width-track configurations ($B = 800$ mm) be tested to ensure adequate lateral stability during manoeuvres on the steep incline. Three configurations were evaluated: B800×L400 ($B/L = 2.00$), B800×L600 ($B/L = 1.33$), and B800×L800 ($B/L = 1.00$). For R=0 pivot modes on slope, the straight-drive correction uses the flat-ground average power of the corresponding configuration, since bidirectional slope travel at the turning point cancels gravitational potential energy, making the net energy equivalent to flat straight-line travel.

6.3.1 Individual Analysis by Configuration - 18° Slope

6.3.1.1 B800×L400 Robot Platform Data Analysis

This section presents a comprehensive analysis of the energy consumption characteristics of the B800×L400 robot platform (wheelbase 400 mm, track width 800 mm, $B/L = 2.00$) operating on an 18° grass slope (measured mean inclination 17.618°). Drive motors are powered at 48 V and steer motors at 24 V; energy is computed as $E = \sum |I| \cdot V \cdot \Delta t$. Six steering modes identical to the flat-terrain tests are evaluated. For R=0 pivot modes, a straight-drive correction of +364.50 J is applied, derived from the flat-ground average power (126.57 W) multiplied by the body-width crossing time of 2.8798 s. The physical rationale is that bidirectional slope travel at the turning point cancels gravitational potential energy, making the net energy equivalent to flat straight-line travel. No correction is applied to R=1.5 m arc modes, which inherently incorporate the required displacement. Table 6.26 summarises the total energy consumption for all six modes.

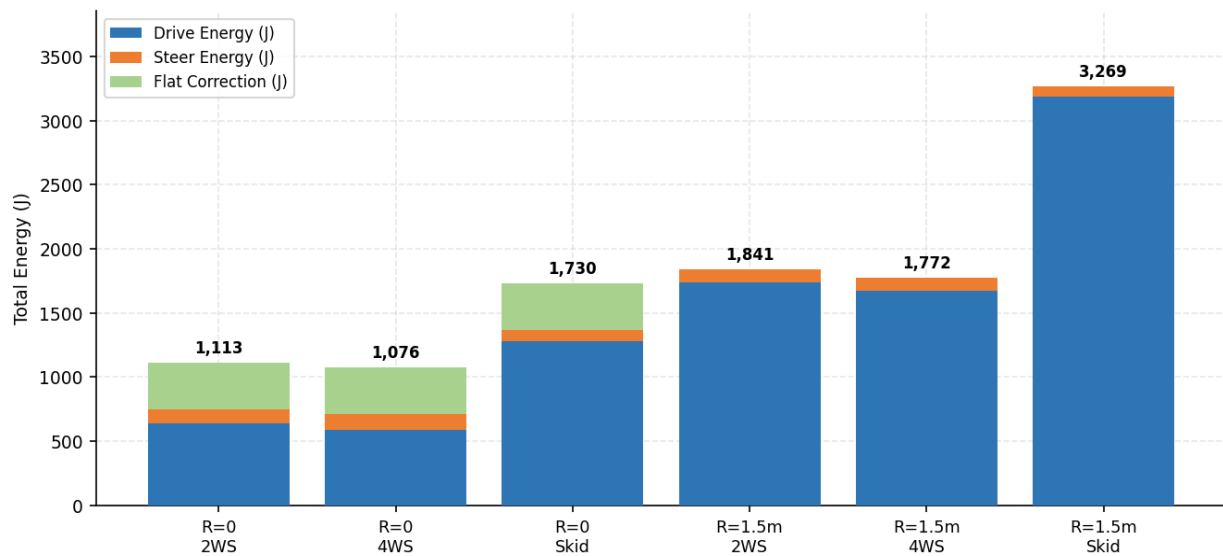


Figure 6-40 Total energy by steering mode - B800xL400 platform, 18° slope (blue=drive, orange=steer, green=flat correction)

Figure 6-40 reveals the energy efficiency hierarchy across the six modes on the 18° slope. Ranked from lowest to highest total energy: 1st: R=0 4WS (1,076.21 J); 2nd: R=0 2WS (1,113.20 J); 3rd: R=0 Skid Steer (1,729.70 J); 4th: R=1.5m 4WS (1,771.96 J); 5th: R=1.5m 2WS (1,840.90 J); 6th: R=1.5m Skid (3,268.62 J). The most efficient mode is R=0 4WS at 1,076.21 J and the least efficient is R=1.5m Skid at 3,268.62 J - a 204% spread. Skid Steer modes consume substantially more energy than wheel-controlled modes due to sustained wheel-ground slip. Notably, the reduced wheel-ground normal force on the slope ($N = mg \cdot \cos\theta < mg$) slightly decreases lateral friction torque, which is why some Skid Steer values on slope are lower than or comparable to their flat-ground equivalents. Steer motor energy constitutes only 2.4-11.7% of the total across all modes.

Figures 6-41 - 6-42 present per-motor energy data for all six modes, providing a quantitative basis for drive-force distribution analysis.

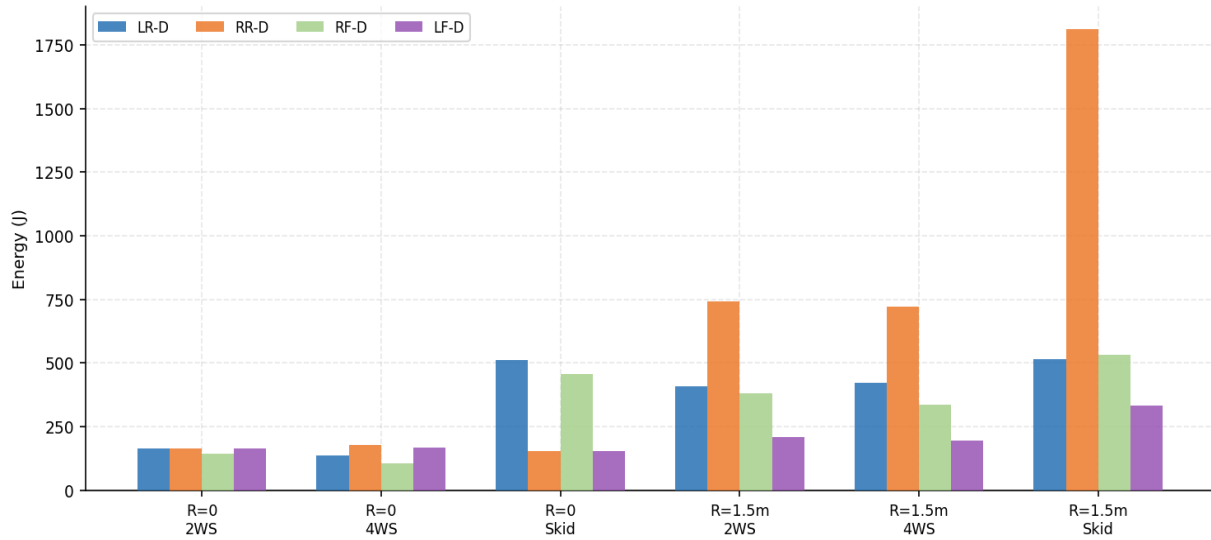


Figure 6-41 Drive motor energy by mode - B800xL400 platform, 18° slope

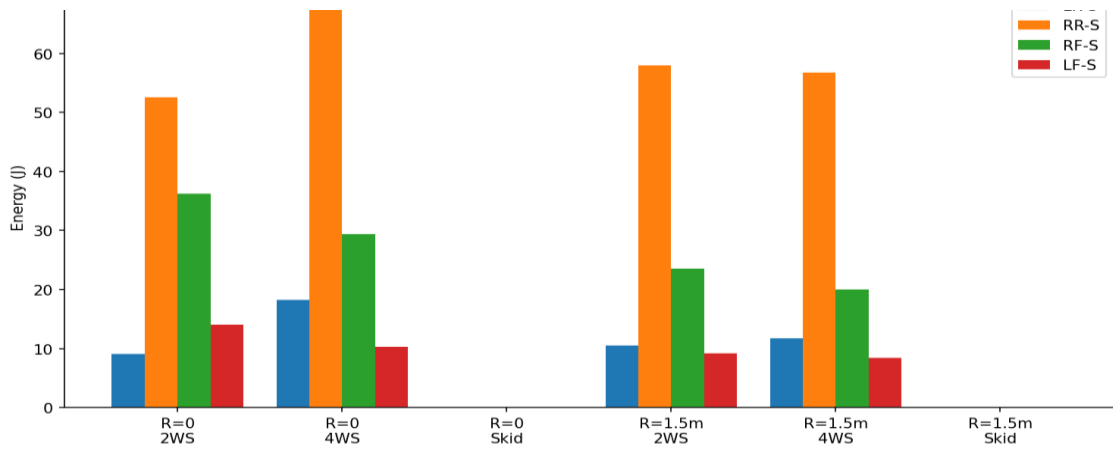


Figure 6-42 Steer motor energy by mode - B800xL400 platform, 18° slope

The drive motor load distribution is analysed mode by mode in Figure 6-43. For R=0 2WS, drive motor loads are broadly balanced with no dominant pair concentration. For R=0 4WS, drive motor loads are broadly balanced with no dominant pair concentration. For R=0 Skid Steer, the diagonal A (LR-D + RF-D) (967 J combined) carries $3.1\times$ the energy of the diagonal B (RR-D + LF-D) (310 J), indicating a clear load concentration pattern characteristic of this manoeuvre on the slope. For R=1.5m 2WS, the rear axle pair (LR-D + RR-D) (1148 J combined) carries $1.9\times$ the energy of the front axle pair (591 J), indicating a clear load concentration pattern characteristic of this manoeuvre on the slope. For R=1.5m 4WS, the rear axle pair (LR-D + RR-D) (1144 J combined) carries $2.2\times$ the energy of the front axle pair (531 J), indicating a clear load concentration pattern characteristic of this manoeuvre on the slope. For R=1.5m Skid, the right-side pair (RR-D + RF-D) (2343 J combined) carries $2.8\times$ the energy of the left-

side pair (848 J), indicating that the vehicle was performing a left-turning manoeuvre.. Across the three R=1.5 m arc modes combined, the rear axle pair (LR-D + RR-D) carries on average 2.3× the energy of the front axle pair (RF-D + LF-D). This systematic rear-axle concentration is a defining characteristic of slope operation: the vehicle centre of gravity shifts rearward and downhill on the incline, placing a greater gravitational load on the rear wheels and requiring proportionally higher drive energy to sustain the turning manoeuvre. This pattern differs markedly from flat-ground results and represents the most important slope-specific finding for mechanical design.

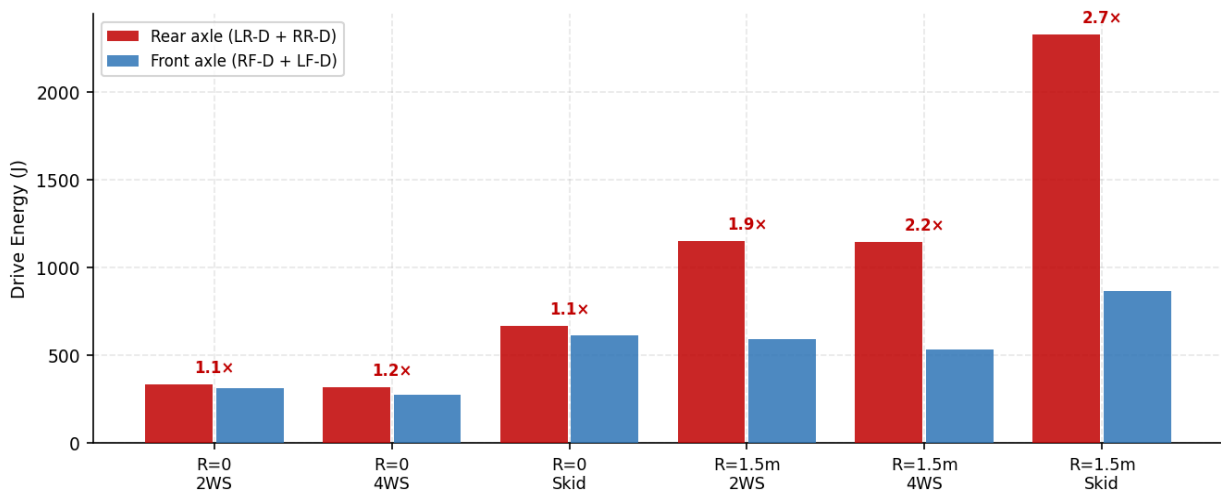


Figure 6-43 Rear vs front axle drive energy by mode - B800×L400 platform, 18° slope (number labels = rear/front ratio)

During straight-line travel on flat ground, the overall average power is 126.57 W, of which drive motors consume 119.19 W (94.2%) and steer motors 7.38 W (5.8%). This flat-ground power value is used as the correction baseline for all R=0 slope modes, as described above.

In summary, the experimental data demonstrate that R=0 4WS (1,076.21 J) is the most energy-efficient steering strategy for the B800×L400 platform on the 18° slope. Skid Steer modes should be avoided in energy-sensitive slope applications. The most significant slope-specific finding is the systematic rear-axle drive motor concentration during arc manoeuvres, which directly informs rear-wheel motor sizing and the mechanical design of slope-capable agricultural robot platforms.

6.3.1.2 B800×L600 Robot Platform Data Analysis

This section presents a comprehensive analysis of the energy consumption characteristics of the B800×L600 robot platform (wheelbase 600 mm, track width 800 mm, B/L = 1.33) operating on an 18° grass slope (measured mean inclination 17.618°).

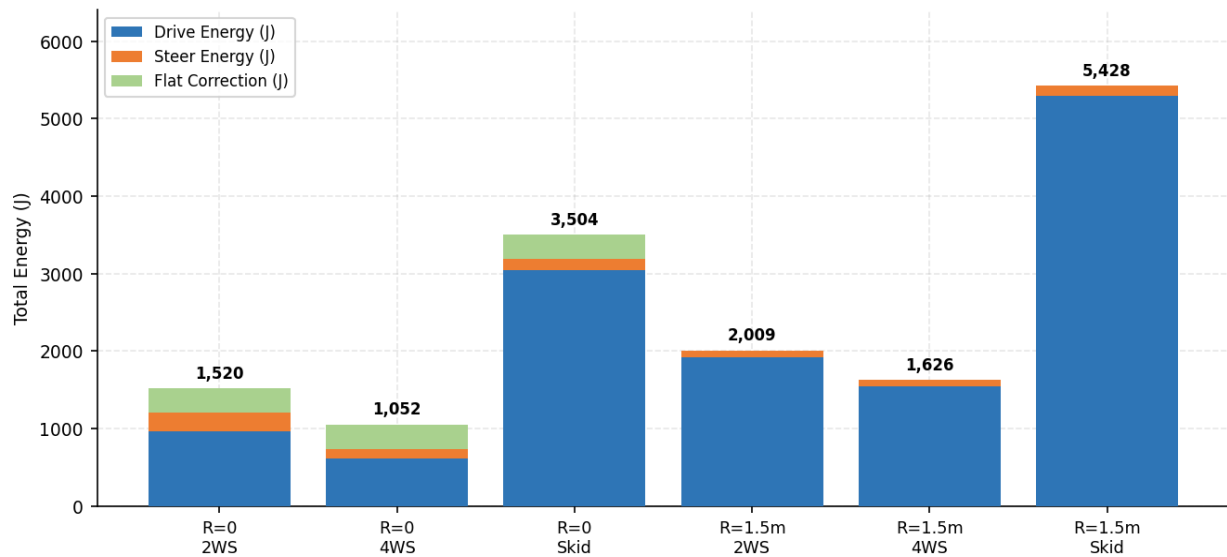


Figure 6-44 Total energy by steering mode - B800×L600 platform, 18° slope (blue=drive, orange=steer, green=flat correction)

Figure 6-44 reveal the energy efficiency hierarchy across the six modes on the 18° slope. Ranked from lowest to highest total energy: 1st: R=0 4WS (1,051.68 J); 2nd: R=0 2WS (1,520.30 J); 3rd: R=1.5m 4WS (1,625.94 J); 4th: R=1.5m 2WS (2,008.77 J); 5th: R=0 Skid Steer (3,504.40 J); 6th: R=1.5m Skid (5,427.66 J). The most efficient mode is R=0 4WS at 1,051.68 J and the least efficient is R=1.5m Skid at 5,427.66 J - a 416% spread. Skid Steer modes consume substantially more energy than wheel-controlled modes due to sustained wheel-ground slip. Notably, the reduced wheel-ground normal force on the slope ($N = mg \cdot \cos\theta < mg$) slightly decreases lateral friction torque, which is why some Skid Steer values on slope are lower than or comparable to their flat-ground equivalents. Steer motor energy constitutes only 2.4-15.6% of the total across all modes.

Figures 6-45 – 6-46 present per-motor energy data for all six modes, providing a quantitative basis for drive-force distribution analysis.

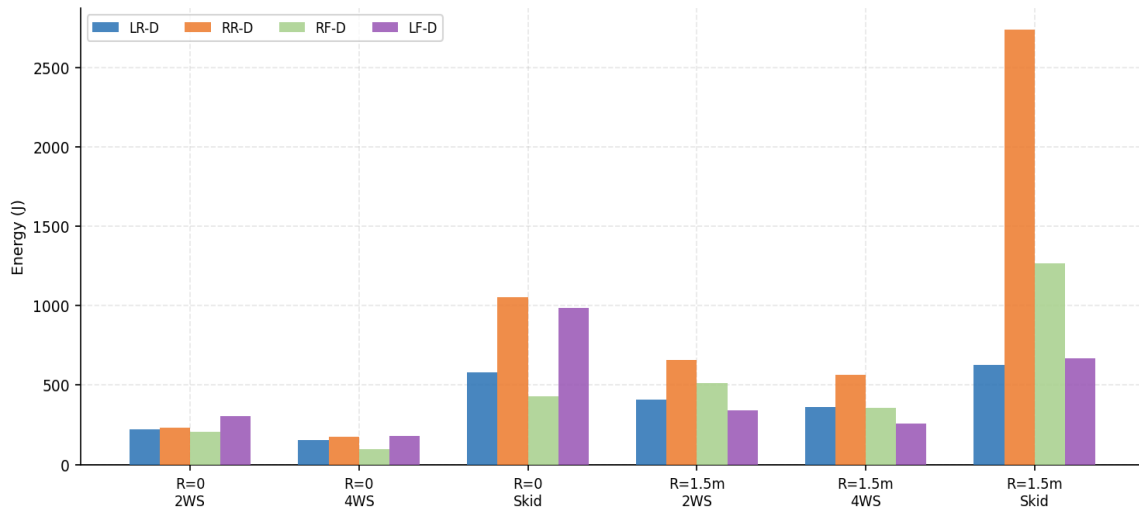


Figure 6-45 Drive motor energy by mode - B800xL600 platform, 18° slope

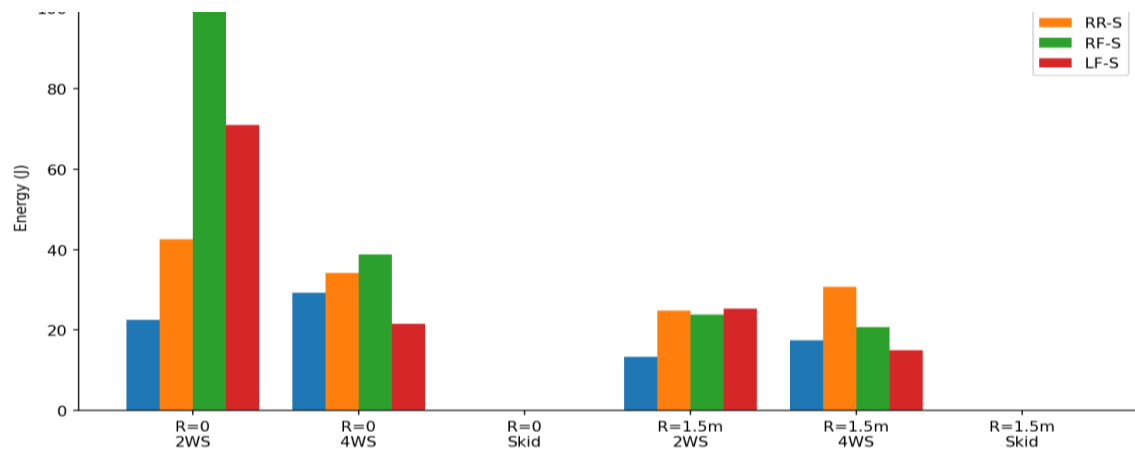


Figure 6-46 Steer motor energy by mode - B800xL600 platform, 18° slope

The drive motor load distribution is analysed mode by mode below shown in Figure 6-47. For R=0 2WS, drive motor loads are broadly balanced with no dominant pair concentration. For R=0 4WS, drive motor loads are broadly balanced with no dominant pair concentration. For R=0 Skid Steer, the diagonal B (RR-D + LF-D) (2039 J combined) carries 2.0× the energy of the diagonal A (LR-D + RF-D) (1005 J), indicating a clear load concentration pattern characteristic of this manoeuvre on the slope. For R=1.5m 2WS, the right-side pair (RR-D + RF-D) (1174 J combined) carries 1.6× the energy of the left-side pair (748 J), indicating a clear load concentration pattern characteristic of this manoeuvre on the slope. For R=1.5m 4WS, the rear axle pair (LR-D + RR-D) (928 J combined) carries 1.5× the energy of the front axle pair (614 J), indicating a clear load concentration pattern characteristic of this manoeuvre on the slope. For R=1.5m Skid, the right-side pair (RR-D + RF-D) (4002 J combined) carries 3.1× the energy of the left-side pair (1296 J), indicating that the vehicle was performing a left-turning

manoeuvre. Across the three $R=1.5$ m arc modes combined, the rear axle pair (LR-D + RR-D) carries on average $1.5\times$ the energy of the front axle pair (RF-D + LF-D). This systematic rear-axle concentration is a defining characteristic of slope operation: the vehicle centre of gravity shifts rearward and downhill on the incline, placing a greater gravitational load on the rear wheels and requiring proportionally higher drive energy to sustain the turning manoeuvre. This pattern differs markedly from flat-ground results and represents the most important slope-specific finding for mechanical design.

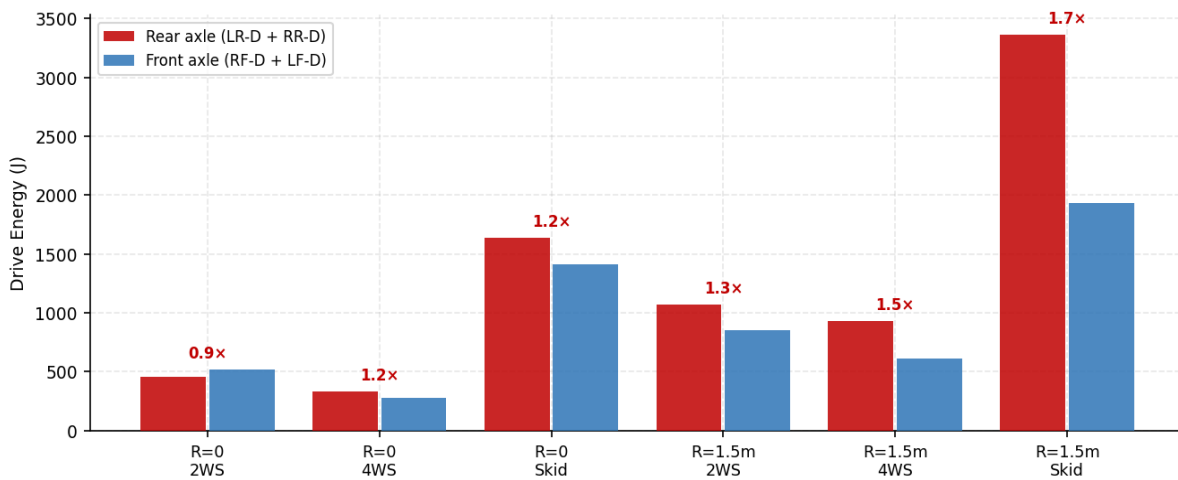


Figure 6-47 Rear vs front axle drive energy by mode - B800×L600 platform, 18° slope (number labels = rear/front ratio)

During straight-line travel on flat ground, the overall average power is 110.04 W, of which drive motors consume 105.44 W (95.8%) and steer motors 4.60 W (4.2%). This flat-ground power value is used as the correction baseline for all $R=0$ slope modes, as described above.

In summary, the experimental data demonstrate that $R=0$ 4WS (1,051.68 J) is the most energy-efficient steering strategy for the B800×L600 platform on the 18° slope. Skid Steer modes should be avoided in energy-sensitive slope applications.

6.3.1.3 B800×L800 Robot Platform Data Analysis

This section presents a comprehensive analysis of the energy consumption characteristics of the B800×L800 robot platform (wheelbase 800 mm, track width 800 mm, $B/L = 1.00$) operating on the 18° grass slope.

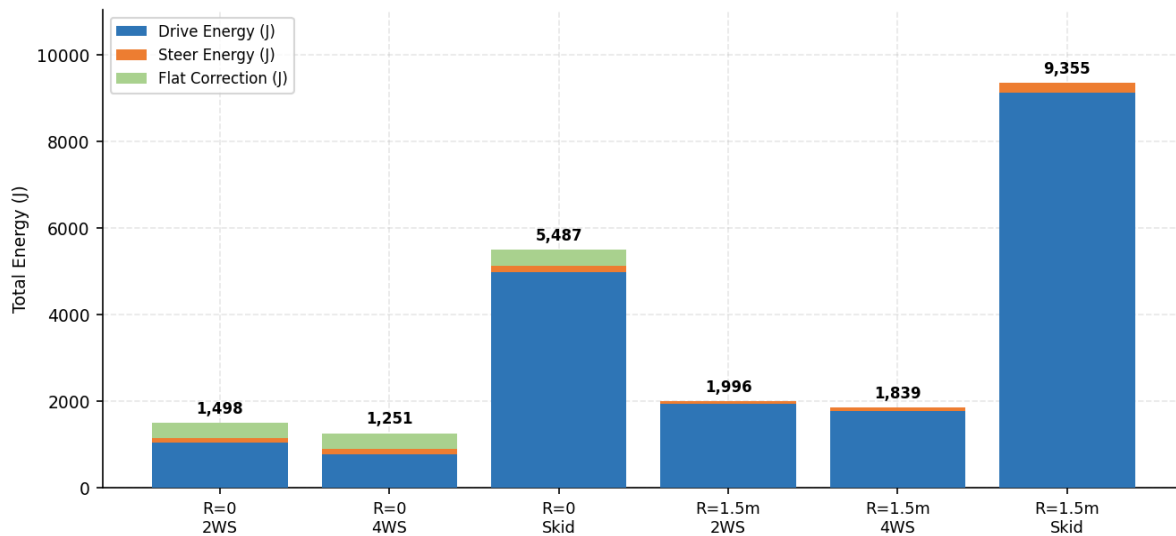


Figure 6-48 Total energy by steering mode - B800×L800 platform, 18° slope (blue=drive, orange=steer, green=flat correction)

Figure 6-48 reveal the energy efficiency hierarchy across the six modes on the 18° slope. Ranked from lowest to highest total energy: 1st: R=0 4WS (1,251.16 J); 2nd: R=0 2WS (1,498.31 J); 3rd: R=1.5m 4WS (1,838.74 J); 4th: R=1.5m 2WS (1,995.59 J); 5th: R=0 Skid Steer (5,486.59 J); 6th: R=1.5m Skid (9,354.61 J). The most efficient mode is R=0 4WS at 1,251.16 J and the least efficient is R=1.5m Skid at 9,354.61 J - a 648% spread. Skid Steer modes consume substantially more energy than wheel-controlled modes due to sustained wheel-ground slip. Notably, the reduced wheel-ground normal force on the slope ($N = mg \cdot \cos\theta < mg$) slightly decreases lateral friction torque, which is why some Skid Steer values on slope are lower than or comparable to their flat-ground equivalents. Steer motor energy constitutes only 2.5-9.7% of the total across all modes.

Figures 6-49 – 6-50 present per-motor energy data for all six modes, providing a quantitative basis for drive-force distribution analysis.

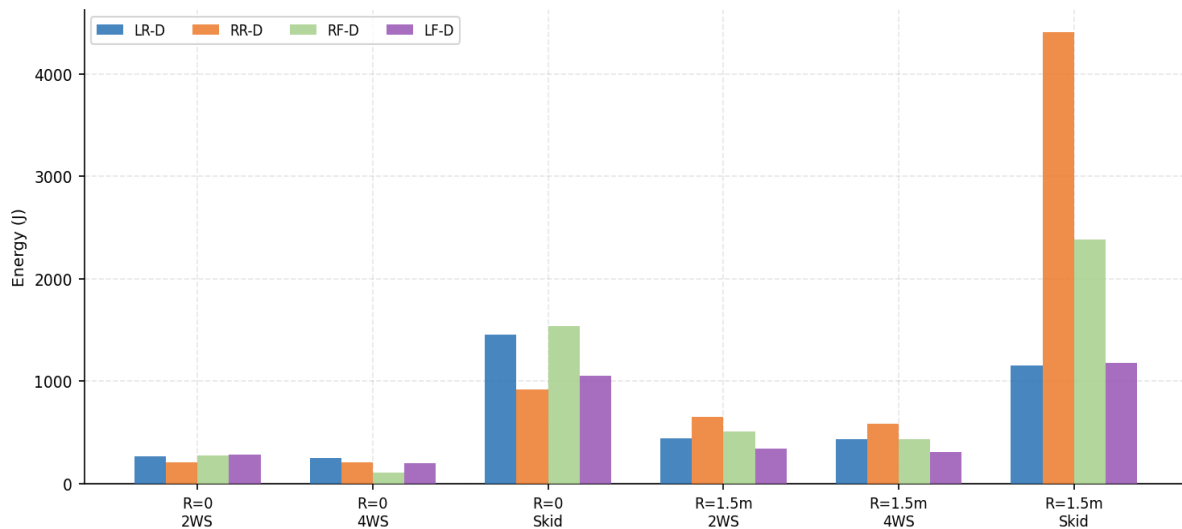


Figure 6-49 Drive motor energy by mode - B800xL800 platform, 18° slope

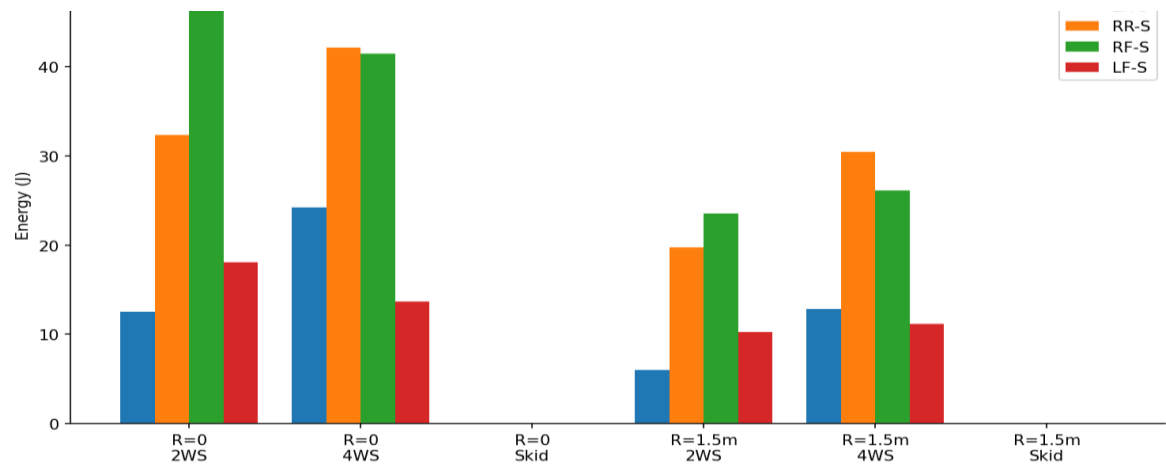


Figure 6-50 Steer motor energy by mode - B800xL800 platform, 18° slope

The drive motor load distribution is analysed mode by mode below in Figure 6-51. For R=0 2WS, drive motor loads are broadly balanced with no dominant pair concentration. For R=0 4WS, drive motor loads are broadly balanced with no dominant pair concentration. For R=0 Skid Steer, the diagonal A (LR-D + RF-D) (2990 J combined) carries 1.5× the energy of the diagonal B (RR-D + LF-D) (1974 J), indicating a clear load concentration pattern characteristic of this manoeuvre on the slope. For R=1.5m 2WS, drive motor loads are broadly balanced with no dominant pair concentration. For R=1.5m 4WS, drive motor loads are broadly balanced with no dominant pair concentration. For R=1.5m Skid, the right-side pair (RR-D + RF-D) (6794 J combined) carries 2.9× the energy of the left-side pair (2325 J), indicating that the vehicle was performing a left-turning manoeuvre. Across the three R=1.5 m arc modes combined, the rear axle pair (LR-D + RR-D) carries on average 1.4× the energy of the front

axle pair (RF-D + LF-D). This systematic rear-axle concentration is a defining characteristic of slope operation: the vehicle centre of gravity shifts rearward and downhill on the incline, placing a greater gravitational load on the rear wheels and requiring proportionally higher drive energy to sustain the turning manoeuvre. This pattern differs markedly from flat-ground results and represents the most important slope-specific finding for mechanical design.

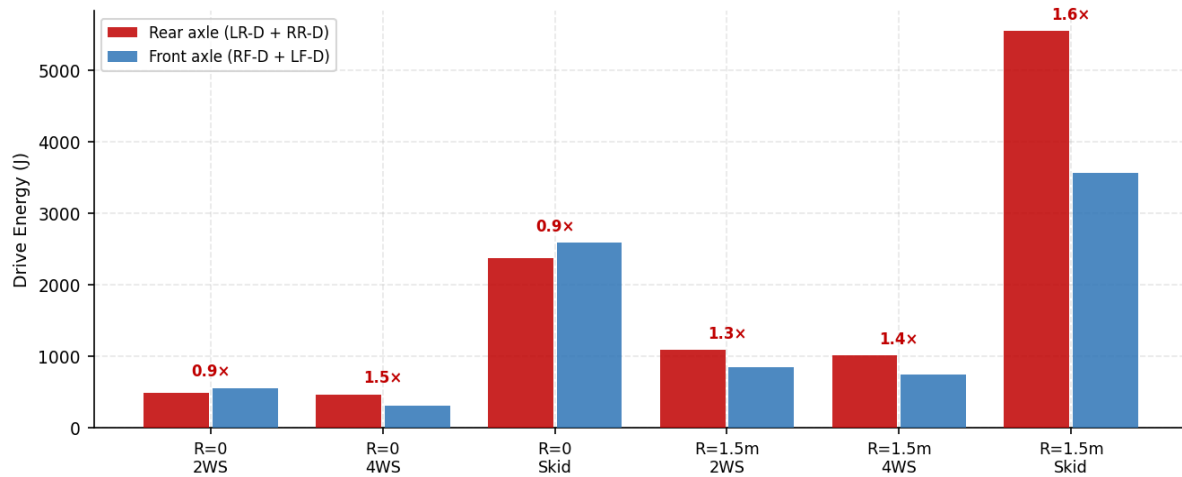


Figure 6-51 Rear vs front axle drive energy by mode - B800×L800 platform, 18° slope (number labels = rear/front ratio)

During straight-line travel on flat ground, the overall average power is 124.73 W, of which drive motors consume 119.25 W (95.6%) and steer motors 5.48 W (4.4%). This flat-ground power value is used as the correction baseline for all R=0 slope modes, as described above.

In summary, the experimental data demonstrate that R=0 4WS (1,251.16 J) is the most energy-efficient steering strategy for the B800×L800 platform on the 18° slope. Skid Steer modes should be avoided in energy-sensitive slope applications. The most significant slope-specific finding is the systematic rear-axle drive motor concentration during arc manoeuvres, which directly informs rear-wheel motor sizing and the mechanical design of slope-capable agricultural robot platforms.

6.3.2 Cross-Configuration Energy Consumption Analysis for 18° Slope

This section presents a systematic cross-configuration energy analysis of the B=800 mm (full-width track) robot platform at three wheelbases (L=800/600/400 mm) on an 18° extreme grass slope (measured 17.618°). Safety constraints required full-width-track configurations. Analysis covers: overall energy ranking, optimal configuration per steering mode, energy trends with L and B/L, and regression formulae (with R² confidence) for all six modes.

6.3.2.1 Best Achievable Energy per Configuration

Figure 6-52 ranks the three configurations by minimum energy across all modes. B800×L600 ranks first, achieving 1,051.68 J in R=0 4WS; B800×L400 follows at 1,076.21 J - a gap of only 24.5 J (2.3%). The minor slope advantage of B800×L600 over B800×L400 differs from the flat-terrain pattern where B800×L400 leads; the slightly longer wheelbase (600 mm) may provide marginally better geometric stability on the incline for coordinated four-wheel steering. Given the 2.3% difference, both configurations are practically co-optimal for slope operation. B800×L800 ranks third at 1,251.16 J, approximately 16-19% higher.

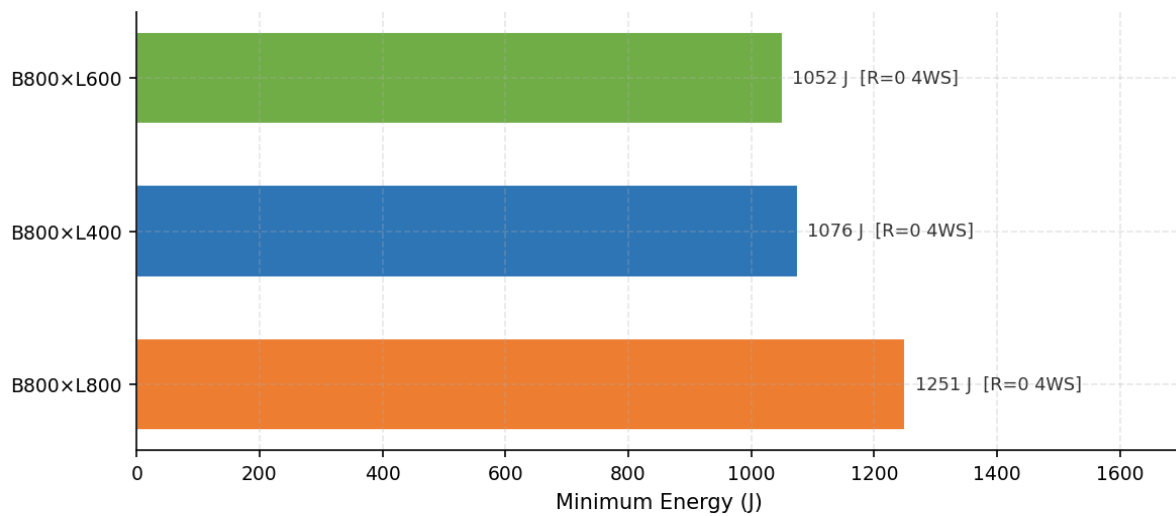


Figure 6-52 Overall slope energy ranking - minimum energy per configuration across all valid modes.

6.3.2.2 Best Configuration per Steering Mode

Figure 6-53 shows the top three configurations per mode. B800×L400 ranks first in 4 modes (R=0 2WS, R=0 Skid, R=1.5m 2WS, R=1.5m Skid); B800×L600 leads in 2 modes (R=0 4WS, R=1.5m 4WS). B800×L800 does not rank first in any mode.

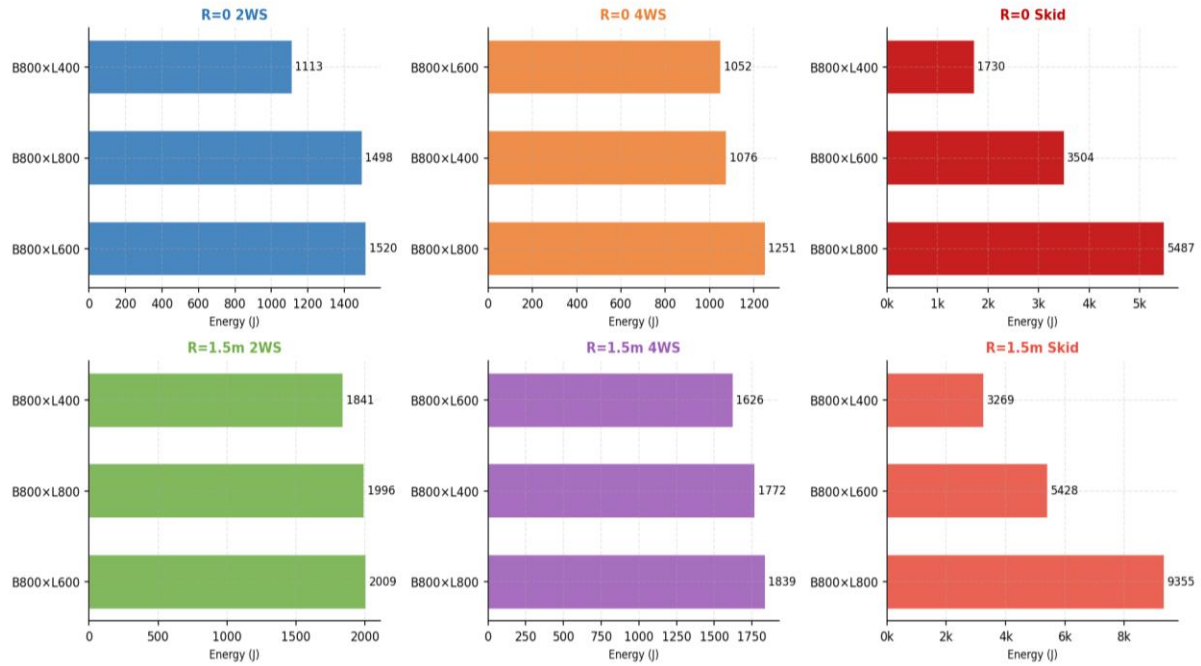


Figure 6-53 Energy ranking by steering mode - best to worst configuration.

6.3.2.3 Energy Trend Analysis - Effect of L and B/L

With $B=800$ mm fixed, L is the sole varying geometric parameter (equivalent to varying B/L). Figures 6-54 – 6-55 show energy trends vs L and B/L . The general pattern: energy increases with L in four modes ($R=0$ 2WS, $R=0$ Skid, $R=1.5m$ 2WS, $R=1.5m$ Skid), consistent with flat-terrain findings. Skid Steer is most sensitive to L (217% increase $R=0$ Skid; 186% $R=1.5m$ Skid, from $L=400$ to $L=800$). $R=1.5m$ 4WS is nearly L -insensitive (only 3.8% increase). In $R=0$ 4WS, $B800 \times L600$ slightly outperforms $B800 \times L400$ (2.3%), weakening the linear $R^2=0.647$ - a slope-specific geometric stability effect.

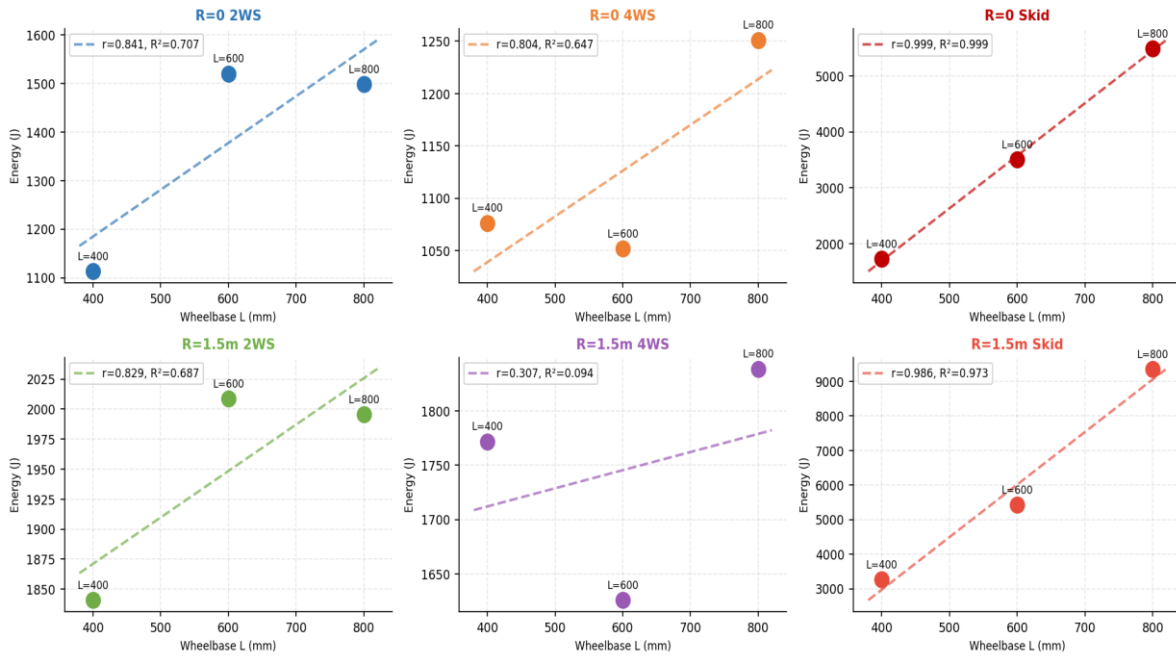


Figure 6-54 Energy vs wheelbase L for all steering modes ($B=800\text{mm}$; dashed = linear fit with r and R^2).

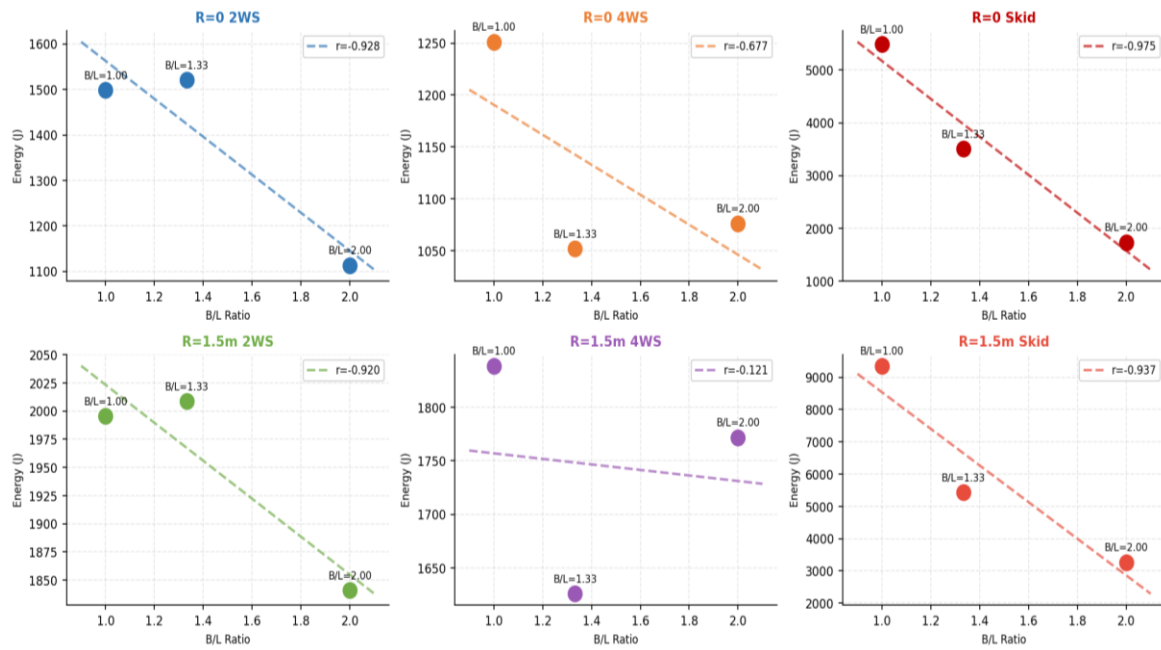


Figure 6-55 Energy vs B/L ratio for all modes (higher B/L = shorter wheelbase = generally lower energy).

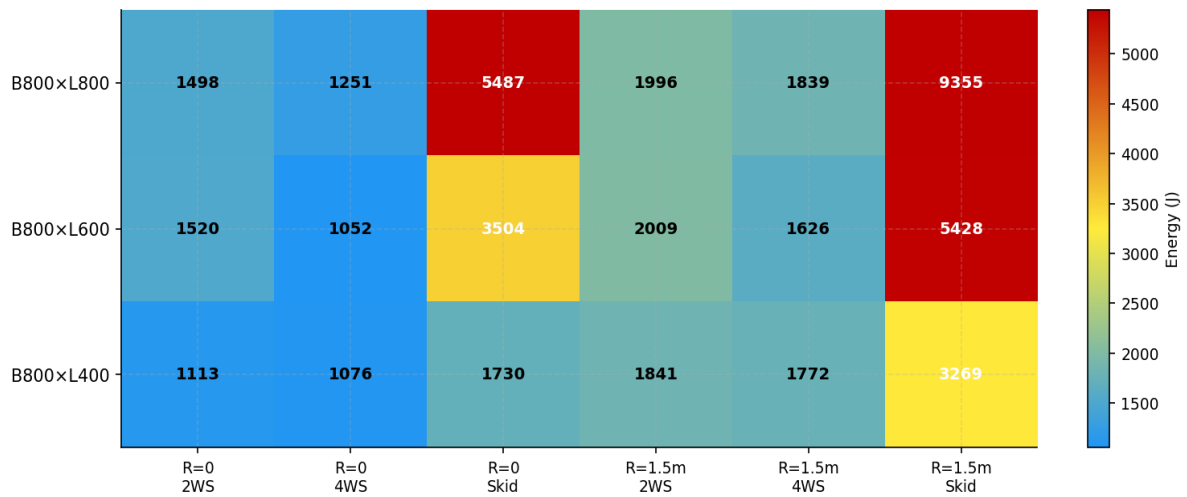


Figure 6-56 Slope energy heatmap ($B \times L$ design space; blue=efficient; colour capped at 90th percentile).

In Figure 6-56, R=0 Skid Steer energy is consistently lower on slope than flat: B800xL800 slope 5,487 J vs flat 6,093 J (−9.9%); B800xL400 slope 1,730 J vs flat 2,216 J (−21.9%). This may be due to the Skid Steer energy is dominated by lateral wheel-ground friction torque, which is proportional to normal force $N = mg$ on flat but $N = mg \cdot \cos(18^\circ) \approx 0.951mg$ on slope - a ~5% reduction that directly reduces the resistive moment. Conversely, R=1.5m Skid Steer is higher on slope (requires extra work against gravity over the arc path). These two opposing slope responses share a common root: R=0 Skid is friction-dominated (reduced normal force saves energy on slope); R=1.5m Skid must additionally work against gravity along the arc (increases energy on slope).

6.3.2.4. Regression Formulae - Energy as a Function of L

With $B=800$ mm fixed, a linear regression $E = a + b \cdot L$ is fitted for each steering mode (L in mm, E in J). Table 6.7 summarises Pearson r , R^2 , and the recommended formula for each mode. Figure 6-57 shows actual vs predicted values. Note: with only 3 data points, R^2 has very limited statistical degrees of freedom; formulae indicate trend direction rather than precise prediction.

Table 6.7 Regression Formulae for All Six Modes (18° Slope; $B=800$ mm; L in mm, E in J)

Mode	$r(L)$	$R^2(L)$	Formula $E = a + b \cdot L$	$r(B/L)$	$R^2(B/L)$	B/L correlation	Dominant var
R=0 2WS	+0.841	0.707	$E = 800 + 0.963 \cdot L$	−0.928	0.861	Strong −ve	L
R=0 4WS	+0.804	0.647	$E = 864 + 0.437 \cdot L$	−0.677	0.458	Moderate	L

Mode	r(L)	R ² (L)	Formula $E = a + b \cdot L$	r(B/L)	R ² (B/L)	B/L correlation	Dominant var
R=0 Skid	+0.999	0.999	$E = -2062 + 9.39 \cdot L$	-0.975	0.951	Very strong -ve	L (dominant)
R=1.5m 2WS	+0.829	0.687	$E = 1716 + 0.387 \cdot L$	-0.920	0.846	Strong -ve	L
R=1.5m 4WS	+0.307	0.094	Near-constant ~1745 J	-0.121	0.015	Very weak	Neither
R=1.5m Skid	+0.986	0.973	$E = -3112 + 15.22 \cdot L$	-0.937	0.878	Very strong -ve	L (dominant)

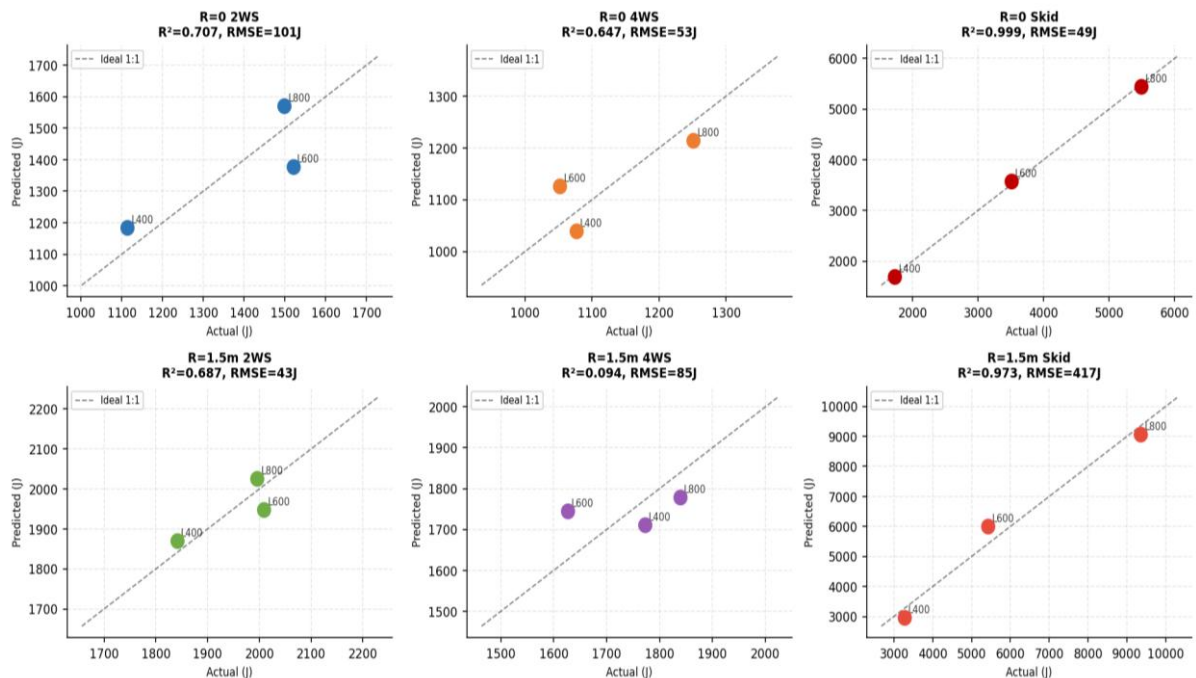


Figure 6-57 Linear regression actual vs predicted ($E = a + b \cdot L$) - all steering modes, 18° slope.

Key regression findings: (1) R=0 Skid Steer has the highest linear fit quality ($r=+0.9995$, $R^2=0.999$); formula $E=-2062+9.39 \cdot L$ nearly perfectly describes the data, with every 100 mm increase in L adding ~939 J. (2) R=1.5m Skid Steer is next ($r=+0.986$, $R^2=0.973$); formula $E=-3112+15.22 \cdot L$ has the steepest slope - ~1522 J per 100 mm of L increase, making it the most L-sensitive mode. (3) R=1.5m 4WS has the weakest correlation ($R^2=0.094$); energy varies only 1,626-1,839 J across configurations ($CV \approx 6.4\%$), confirming this mode is geometrically robust and L-insensitive. (4) R=0 4WS has $R^2=0.647$, lower than other wheel-controlled modes, due to the minor B800×L600 < B800×L400 ranking reversal on slope. (5) All b-coefficients

are positive, confirming the universal rule that shorter wheelbase improves slope energy efficiency.

6.3.2.4 Discussion

Slope results are broadly consistent with flat-terrain findings: short wheelbase ($L=400$ mm) maintains energy advantage in most modes; Skid Steer should be avoided in all configurations. However, several slope-specific differences emerge: first, $R=0$ 4WS best configuration shifts slightly from $B800 \times L400$ on flat to $B800 \times L600$ on slope (by 2.3%), attributed to slope stability geometry; second, $R=0$ Skid Steer is consistently lower-energy on slope than flat (−10 to −22%), a predictable effect of reduced normal force ($mg \cdot \cos\theta$) diminishing the resistive friction torque; third, $R=1.5m$ arc modes are substantially higher-energy on slope (+50-80%) due to the additional gravity work required over the arc path.

$R=1.5m$ 4WS exhibits the same geometric robustness on slope as on flat terrain: It shown in Figure 6.67, energy spans only 1,626-1,839 J across three configurations ($CV \approx 6.4\%$), with a linear correlation $r=+0.307$ vs L . This means that if a designer selects $R=1.5m$ 4WS as the primary steering strategy, wheelbase choice has negligible constraint on slope energy efficiency, freeing L selection to be driven by stability, payload, and other design criteria.

It is worth noting that on a slope, when the direction of travel is aligned with the gradient, the drive motors must consume additional energy to overcome the gravitational force component acting along the slope, to maintain the vehicle's positional stability.

6.3.2.5 Conclusions for Slope Run

This study analysed three $B=800$ mm full-width-track configurations on an 18° extreme slope. Main conclusions:

- (1) Optimal slope configuration: $B800 \times L600$ ($R=0$ 4WS, 1,051.68 J) and $B800 \times L400$ (1,076.21 J) are effectively co-optimal (2.3% gap). $B800 \times L800$ ranks third at 1,251.16 J.
- (2) Energy trend: except $R=1.5m$ 4WS, energy increases monotonically with L . Skid Steer is most sensitive to L (217% increase); $R=1.5m$ 4WS is least sensitive (3.8%).
- (3) Skid Steer slope behaviour: $R=0$ Skid is lower-energy on slope than flat (reduced normal force reduces friction torque); $R=1.5m$ Skid is higher-energy (arc path requires additional gravity work). Both effects are physically predictable.

(4) Regression formulae: $R=0$ Skid ($R^2=0.999$, $E=-2062+9.39 \cdot L$) and $R=1.5m$ Skid ($R^2=0.973$, $E=-3112+15.22 \cdot L$) have very high credibility; $R=1.5m$ 4WS has no meaningful regression ($R^2=0.094$); other wheel-controlled modes have moderate credibility ($R^2=0.647-0.707$).

6.4 Conclusion

This chapter has presented a comprehensive experimental investigation of energy consumption in a four-wheel-independent-steering agricultural robot platform across nine geometric configurations and six steering modes on both flat grass terrain and an 18° grass slope. The experimental campaign combined systematic individual configuration analysis with cross-configuration correlation and regression modelling, yielding a set of robust, quantitative findings with direct implications for the design and operation of energy-efficient agricultural robots in orchard and pastoral environments.

The most significant and consistent finding of this study is the dominant role of wheelbase L as the primary geometric determinant of turning energy consumption. Across five of six steering modes on flat terrain, and across all modes on slope (with the exception of $R=1.5m$ four-wheel steering), shorter wheelbase configurations achieved substantially lower energy consumption, with the $L = 400$ mm group attaining 25-35% lower turning energy than $L = 800$ mm equivalents. This finding is statistically robust (Pearson $r = 0.80-0.90$ for most modes) and holds across both pivot and arc-path manoeuvres, as well as across flat and sloped terrain.

Among all tested configurations, the $B800 \times L400$ platform (track width 800 mm, wheelbase 400 mm, $B/L = 2.00$) consistently delivered the most outstanding performance throughout the experimental programme. On flat ground, $B800 \times L400$ ranked first or second in every steering mode, achieving the near-global minimum of 709.57 J via $R=0$ 4WS - only 2.62 J above the absolute minimum. On the 18° slope, $B800 \times L400$ maintained strong performance across all modes, and was the only configuration to complete all six steering modes successfully at both test conditions. Field observations during testing confirmed what the data quantitatively reflects: the $B800 \times L400$ configuration exhibited visibly superior agility, rapid and smooth manoeuvre execution, and excellent directional stability on both flat grass and the steep incline. On the slope in particular, the wide-track, short-wheelbase geometry produced impressively stable turning behaviour that distinguished it markedly from the longer-wheelbase configurations. These observations are consistent with what was directly observed during field testing - the $B800 \times L400$ configuration exhibited remarkable agility, and was the only

configuration in which the 1.5 m skid-steer turning radius approached the theoretical value ($R \approx 1.5$ m). This may be related to its high drive moment arm and low resistance moment arm.

Four-wheel steering (both $R=0$ and $R=1.5$ m variants) consistently outperformed two-wheel steering and skid steering across most configurations and terrain conditions, confirming its fundamental energy advantage through coordinated wheel geometry that minimises lateral slip. Skid Steer modes uniformly incurred the highest energy penalties across all configurations and terrains, with energy multiples of $3\text{--}25\times$ relative to the optimal mode. The L^2/B predictor variable was found to govern skid-steer energy with high accuracy ($r = +0.86\text{--}0.98$), and the catastrophic energy cost of skid-steer arc manoeuvres for long-wheelbase, narrow-track configurations ($B400\times L800$ $R=1.5$ m Skid: 23,847 J - a $34\times$ penalty over the global optimum) confirms that skid-steer arc manoeuvres should be strictly avoided in energy-sensitive applications.

Slope inclination introduced important modifications to the flat-terrain energy patterns. Arc-path manoeuvres ($R=1.5$ m) were most severely affected by slope, with average energy increases of 68–72% compared to flat-terrain values, primarily due to additional gravitational work against elevation change during the arc path. $R=0$ Skid Steer uniquely exhibited lower energy on slope than on flat terrain, a physically predictable consequence of the reduced normal force ($N = mg \cdot \cos\theta$) diminishing friction torque. The most significant slope-specific structural finding was the systematic rear-axle drive motor overloading during arc manoeuvres (rear/front energy ratio reaching up to $2.7\times$ in some modes), driven by rearward weight transfer on the incline. This characteristic is not observable from flat-terrain tests alone and has direct implications for rear-axle motor sizing and structural design of slope-capable platforms. It is also worth noting that on a slope, when the direction of travel is aligned with the gradient, the drive motors must consume additional energy to overcome the gravitational force component acting along the slope, in order to maintain the vehicle's positional stability.

The regression models developed in this study - including bilinear, L^2/B -based, and full quadratic formulations - provide practical tools for early-stage energy estimation. The highest fidelity was achieved for $R=0$ 2WS ($R^2 = 0.982$) and $R=0$ Skid Steer ($R^2 = 0.960$ bilinear), confirming that these modes have a cleanly geometric energy dependence. $R=1.5$ m 4WS was the notable exception, with near-constant energy across all configurations (coefficient of variation $\sim 14\%$), making it the most geometrically robust and terrain-adaptive mode in the study.

Several directions for future research emerge naturally from this work. First, the experimental dataset was obtained under specific soil moisture and grass conditions at one geographic location; testing across a wider range of terrain types (bare soil, wet grass, hard compacted surfaces) would improve the generalisability of the findings, particularly for the skid-steer and terramechanics-sensitive modes. Second, the energy models developed here are fundamentally kinematic and electrical; future work incorporating detailed tyre-terrain interaction modelling based on Bekker-type or Magic Formula frameworks could improve predictive accuracy, particularly for the slope condition where normal force and friction coefficient are terrain-dependent. Third, the rear-axle overloading phenomenon on slope warrants further mechanical investigation - specifically, the relationship between rear-axle load, motor thermal limits, and long-term mechanical durability in continuous slope operation. Finally, the integration of these energy findings into full-cycle path planning for orchard operations - explicitly accounting for turning energy, travel energy, and battery management - represents a direct and high-impact application of the results obtained here.

In conclusion, the experimental findings presented in this chapter establish a quantitative and physically grounded foundation for the energy-optimal geometric design and steering strategy selection of electric agricultural robot platforms. The clear recommendation for short-wheelbase, wide-track configurations (most notably B800×L400) using four-wheel steering for pivot manoeuvres is supported by consistent experimental evidence across both terrain types, and represents a significant practical contribution to the emerging field of energy-efficient autonomous agricultural vehicles.

7 Conclusion and Discussion

7.1 Summary of Research

This research was motivated by a clearly identified gap in the existing literature: despite an extensive body of mathematical modelling work predicting that steering strategy and vehicle geometry are major determinants of energy consumption in agricultural vehicles, no physically existing experimental platform had been reported that enables systematic, controlled comparison across adjustable geometric configurations and switchable steering strategies on realistic outdoor terrain. The present work addresses this gap end-to-end - from platform design and mechanical construction through to field experimentation, cross-configuration energy analysis, and regression modelling.

A fully electric, four-wheel-independent-steering agricultural robot platform was designed, built, and validated. The vehicle integrates four modular drive-steer units within a reconfigurable semi-monocoque chassis, supporting independent adjustment of both wheelbase L and track width B across three values each (400 mm, 600 mm, and 800 mm), yielding a 3×3 factorial matrix of nine geometric configurations. Six steering modes were implemented: zero-radius two-wheel point turn, zero-radius four-wheel point turn, zero-radius skid-steer point turn, and the corresponding three arc-path variants at $R = 1.5$ m. Eight-channel electrical energy logging provides continuous measurement of drive and steer motor current during all manoeuvres.

Controlled field experiments were conducted on both near-horizontal grass terrain (mean inclination 0.216°) and a steep grass slope (mean inclination 17.618°), encompassing a total of 54 valid flat-terrain workloads and 18 slope workloads. The experimental dataset provides the first directly comparable, multi-configuration, multi-steering-mode energy measurements obtained on a single physical platform under realistic agricultural terrain conditions.

7.2 Future Research Directions

Autonomous navigation and AI-driven path planning. The energy data collected here provide a quantitative basis for energy-aware path planning - enabling real-time selection of the steering mode and turning geometry that minimises energy expenditure for a given task. With indexed agricultural AI/robotics publications approaching 1,200 per year by 2024 (Preprints.org, 2025),

technologies such as SLAM, deep reinforcement learning for motion planning, and vision-based row detection are now mature for outdoor mobile deployment. Next-generation precision agriculture machinery is expected to embed advanced AI and autonomous decision-making suited to orchard and field environments (Springer Nature, 2025). Equipping the current platform with LiDAR, GPS/RTK, and onboard computing would enable autonomous row-end turning using the energy-optimal steering configuration identified here, directly linking the geometric findings of this research to practical autonomous operation.

Multi-functional integration and advanced sensing payloads. Reconfigurable multi-robot platforms - including SwarmFarm Robotics and coordinated aerial-ground fleets - demonstrate that high functional density is both feasible and operationally valuable in commercial agricultural settings (Computers and Electronics in Agriculture, 2025). The modular drive-steer architecture developed here is well suited to task-specific payloads: stereo and multispectral cameras for crop health monitoring, short-range LiDAR for obstacle avoidance and canopy profiling, robotic arms for targeted spraying or fruit picking, and soil/weather sensors for site-specific data collection. The energy characterisation completed here gives payload designers a precise locomotion energy budget, enabling informed allocation of remaining battery capacity to sensing and actuation - aligning with the emerging concept of the agricultural robot as a unified field intelligence platform.

Modular and standardised propulsion architecture. Reconfigurable systems such as the Hefty platform on Farm-NG's Amiga base exemplify modular design enabling multi-crop reconfigurability across sensing, computing, and mounting configurations (arXiv, 2025). Each drive-steer module developed here is already mechanically self-contained, electrically independent, and geometrically interchangeable. Defining standardised electrical interfaces, mounting footprints, and communication protocols would allow chassis designs varying in size, payload capacity, and row clearance to share the same proven drive units - substantially reducing development cost and time-to-deployment for future variants.

Adaptive auto-adjustable wheelbase and track width. This research establishes conclusively that geometric configuration has a quantifiable, predictable effect on energy consumption; the logical next step is closing the loop via linear actuators or lead-screw mechanisms in the wheel support beams and chassis rails, controlled by an onboard geometry management system. Such a system would allow the vehicle to narrow its track width in confined orchard rows, widen its stance on steep slopes, and shorten its wheelbase to minimise turning energy at row ends - all

autonomously and in real time. The regression models and optimal B/L ratios established here provide the control law directly: given terrain inclination, turning radius, and steering mode, the controller selects the configuration minimising predicted energy based on the derived formulae. Reconfigurable ground vehicles (RGVs) have already been demonstrated in row-crop navigation prototypes (Schmitz et al., 2022), confirming mechanical and control feasibility; the experimental dataset produced here provides the quantitative foundation for extending this to energy-driven real-time geometric reconfiguration in an orchard context.

Scale-up to full-field operational platforms. The regression models and optimal B/L ratios identified here directly inform fixed-geometry decisions for larger production-intent vehicles. A natural progression is a 200 - 500 kg platform sized for orchard or vineyard operations in which the B800×L400 geometry and four-wheel steering architecture are structurally embedded. Operating over a full working day, such a vehicle would validate the predicted energy savings at operational scale under extended duty cycles, encompassing battery management, thermal behaviour, and motor durability considerations not assessable on the current prototype.

Advanced materials and manufacturing processes. Aluminium alloy 6061-T6 offers a $\sim 3\times$ strength-to-weight advantage over structural steel at comparable cost. Carbon fibre reinforced polymer (CFRP) composites - one-fifth the density of steel - can reduce primary structural mass by 30 - 50%, with demonstrated high specific energy absorption in vehicle structural applications and committed automotive adoption in future platforms (ResearchGate, 2020). Metal additive manufacturing via selective laser sintering of titanium and aluminium alloys enables topology-optimised, high-strength parts with complex internal geometries impractical by conventional machining. Advances in continuous carbon fibre 3D printing and recycled CFRP manufacturing are progressively reducing the cost barrier to composite structural components (3DPrint.com, 2023). Adopting these materials systematically - guided by FEA-validated structural models and lifecycle cost analysis - would reduce vehicle mass, improve energy efficiency, and extend operational range while maintaining the structural integrity required for demanding outdoor terrain.

The convergence of these six directions - autonomous AI-driven control, multi-functional payload integration, modular standardised propulsion, adaptive real-time geometric reconfiguration, operational scale-up, and advanced materials - points toward small-scale electric agricultural robots as intelligent, energy-efficient, and adaptable field agents capable

of sustained autonomous operation across orchard and pastoral agriculture. The experimental platform, methodology, and quantitative findings of this thesis provide a validated and transferable foundation for that transition.

7.3 Principal Findings

The experimental results yield a set of quantitative, physically consistent conclusions that collectively advance the understanding of geometry- and steering-related energy consumption in electric agricultural vehicles.

Wheelbase L is the dominant geometric parameter governing turning energy consumption. Across five of six steering modes on flat terrain, and all modes on slope terrain, short-wheelbase configurations ($L = 400$ mm) achieve 25-35% lower turning energy than long-wheelbase equivalents ($L = 800$ mm), with Pearson correlation coefficients of $r = 0.80$ - 0.90 for most modes. This finding is statistically robust and consistent across both pivot and arc-path manoeuvres, and across both terrain inclinations.

The globally optimal flat-terrain configurations are $B600 \times L400$ and $B800 \times L400$, both achieving minimum turning energies of approximately 707-710 J via zero-radius four-wheel steering - representing a 31% reduction relative to the worst-performing configuration ($B800 \times L800$ at 1,022 J). Among all tested platforms, $B800 \times L400$ ($B/L = 2.00$) consistently delivered the most outstanding energy performance and the broadest operational capability, ranking first or second across all six steering modes on flat terrain and maintaining strong performance on the 18° slope. Field observations confirmed what the quantitative data reflect: this configuration exhibited visibly superior agility, rapid and smooth manoeuvre execution, and exceptional directional stability on both flat grass and the steep incline - characteristics consistent with classical vehicle dynamics theory predicting that a high B/L ratio reduces the effective moment arm of turning resistance forces and improves lateral stability.

Four-wheel steering outperforms two-wheel and skid-steer strategies across the majority of configurations and terrains. Skid-steering modes uniformly incur the highest energy penalties (3 - $25\times$ relative to the optimal equivalent mode), and the L^2/B predictor variable captures their energy with high accuracy ($r = 0.86$ - 0.98). The arc four-wheel steering mode ($R = 1.5$ m) exhibits near-constant energy ($\sim 1,016$ J, $CV = 14\%$) across all nine geometric configurations, making it the most geometrically robust steering choice when operational flexibility is required.

On slope terrain, arc-path manoeuvres incur substantially higher energy (+38% to +98%) relative to flat-ground values due to gravitational work along the arc path, while zero-radius skid-steer energy is reduced by up to 22% owing to the decreased wheel-ground normal force. A systematic rearward shift in axle load distribution is identified during arc manoeuvres on slope, with rear-axle drive motors carrying up to $2.7\times$ the energy of front-axle motors - a mechanically significant finding that is not predictable from flat-terrain tests alone and has direct implications for rear-axle motor specification in slope-capable platforms.

Bivariate regression models of the form $E = f(B, L)$ are derived for all six steering modes, achieving $R^2 = 0.70-0.98$, providing practical engineering formulae that enable early-stage energy estimation from geometric parameters during platform design.

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Appendix A

Table A 1 Total Energy Consumption by Mode (with correction breakdown) - B800×L800 Platform

Mode	Dur.(s)	Drive E (J)	Steer E (J)	Corr.Drive (J)	Corr.Steer (J)	Total (J)	Drive%	Steer%
R=0 2WS	8.82	604.91	100.49	343.45	15.78	1,064.63	56.8%	9.4%
R=0 4WS	9.09	615.09	105.66	343.45	15.78	1,079.98	57.0%	9.8%
R=0 Skid Steer	21.57	5,567.09	0	343.45	0	6,092.74	94.1%	0%
R=1.5m 2WS	8.76	1,131.22	76.63	-	-	1,207.85	93.7%	6.3%
R=1.5m 4WS	8.52	932.17	89.44	-	-	1,021.62	91.2%	8.8%
R=1.5m Skid	20.19	7,715.76	0	-	-	7,879.10	100%	0%

Table A 2 Per-Motor Energy (J) for All Modes - B800×L800 Platform

Mode	LR-D	RR-D	RF-D	LF-D	LR-S	RR-S	RF-S	LF-S	Drive Tot.	Steer Tot.
R=0 2WS	185.47	226.15	77.44	115.84	12.94	27.24	37.90	22.40	604.91	100.49
R=0 4WS	304.45	122.59	70.52	117.54	22.83	36.84	31.09	14.89	615.09	105.66
R=0 Skid	2,002.62	969.70	1,721.36	873.41	0	0	0	0	5,567.09	0
R=1.5m 2WS	355.55	210.44	217.68	347.55	7.32	25.44	33.46	10.41	1,131.22	76.63
R=1.5m 4WS	276.12	192.96	177.08	286.02	16.06	35.44	31.42	6.52	932.17	89.44
R=1.5m Skid	4,036.14	377.43	891.93	2,410.27	0	0	0	0	7,715.76	0

Table A 3 Total Energy Consumption by Mode (with correction breakdown) - B600 x L600 Platform

Mode	Dur.(s)	Drive E (J)	Steer E (J)	Corr.Drive (J)	Corr.Steer (J)	Total (J)	Drive%	Steer%
R=0 2WS	7.29	501.65	100.01	351.10	22.06	974.82	51.5%	10.3%
R=0 4WS	8.73	641.56	123.94	351.10	22.06	1,138.66	56.3%	10.9%
R=0 Skid*	29.19	5,503.91	0	351.10	0	6,120.30 *	93.9%	0%
R=1.5m 2WS	7.92	1,111.07	84.18	-	-	1,195.25	93.0%	7.0%
R=1.5m 4WS	8.07	1,123.23	84.12	-	-	1,207.35	93.0%	7.0%
R=1.5m Skid	19.35	6,194.26	0	-	-	6,362.90	100%	0%

* R=0 Skid Steer failed - data recorded but manoeuvre not completed; excluded from efficiency comparisons.

Table A 4 Per-Motor Energy (J) for All Modes - B600 x L600 Platform

Mode	LR-D	RR-D	RF-D	LF-D	LR-S	RR-S	RF-S	LF-S	Drive Tot.	Steer Tot.
R=0 2WS	90.26	119.17	204.59	87.63	13.81	29.79	30.95	25.46	501.65	100.01
R=0 4WS	247.50	107.41	179.58	107.07	27.35	45.18	29.46	21.95	641.56	123.94
R=0 Skid*	1,426.44	1,349.38	1,284.68	1,443.41	0	0	0	0	5,503.91	0
R=1.5m 2WS	277.44	218.75	302.75	312.13	11.12	31.23	30.91	10.92	1,111.07	84.18
R=1.5m 4WS	265.17	223.94	324.85	309.27	16.13	35.04	23.69	9.26	1,123.23	84.12
R=1.5m Skid	3,434.04	219.39	762.83	1,778.00	0	0	0	0	6,194.26	0

Table A 5 Total Energy Consumption by Mode (with correction breakdown)- B400 x L400 Platform

Mode	Dur.(s)	Drive E (J)	Steer E (J)	Corr.Drive (J)	Corr.Steer (J)	Total (J)	Drive%	Steer%
R=0 2WS	8.49	366.56	115.30	301.69	24.33	807.88	45.4%	14.3%
R=0 4WS	8.70	393.43	131.85	301.69	24.33	851.30	46.2%	15.5%
R=0 Skid*	22.65	3,088.46	0	301.69	0	3,582.66 *	90.9%	0%
R=1.5m 2WS	8.40	1,032.30	70.70	-	-	1,103.00	93.6%	6.4%
R=1.5m 4WS	8.91	1,017.74	81.67	-	-	1,099.41	92.6%	7.4%
R=1.5m Skid	37.29	8,093.28	0	-	-	8,388.73	100%	0%

* R=0 Skid Steer failed - data recorded but manoeuvre not completed; excluded from efficiency comparisons.

Table A 6 Per-Motor Energy (J) for All Modes - B400 x L400 Platform

Mode	LR-D	RR-D	RF-D	LF-D	LR-S	RR-S	RF-S	LF-S	Drive Tot.	Steer Tot.
R=0 2WS	85.17	133.19	46.18	102.02	16.39	32.30	41.14	25.47	366.56	115.30
R=0 4WS	187.09	101.92	21.81	82.61	32.89	40.98	36.69	21.29	393.43	131.85
R=0 Skid*	795.57	779.22	649.65	864.02	0	0	0	0	3,088.46	0
R=1.5m 2WS	362.22	209.03	111.86	349.19	12.48	30.65	21.35	6.22	1,032.30	70.70
R=1.5m 4WS	294.27	256.83	159.01	307.63	15.13	33.61	26.70	6.23	1,017.74	81.67
R=1.5m Skid	4,515.95	258.41	1,357.22	1,961.70	0	0	0	0	8,093.28	0

Table A 7 Total Energy Consumption by Mode (with correction breakdown) - B400 x L800 Platform

Mode	Dur.(s)	Drive E (J)	Steer E (J)	Corr.Drive (J)	Corr.Steer (J)	Total (J)	Drive%	Steer%
R=0 2WS	8.16	463.39	101.41	364.48	21.48	950.76	48.7%	10.7%
R=0 4WS	9.84	724.21	150.32	364.48	21.48	1,260.49	57.5%	11.9%
R=0 Skid*	21.39	4,284.56	0	364.48	0	4,843.28 *	92.1%	0%
R=1.5m 2WS	8.67	1,119.42	87.71	-	-	1,207.13	92.7%	7.3%
R=1.5m 4WS	7.17	926.22	64.37	-	-	990.58	93.5%	6.5%
R=1.5m Skid	65.31	23,345.07	0	-	-	23,847.19	100%	0%

* R=0 Skid Steer failed - data recorded but manoeuvre not completed; excluded from efficiency comparisons.

Table A 8 Per-Motor Energy (J) for All Modes - B400 x L800 Platform

Mode	LR-D	RR-D	RF-D	LF-D	LR-S	RR-S	RF-S	LF-S	Drive Tot.	Steer Tot.
R=0 2WS	117.40	120.72	71.60	153.68	12.98	33.63	33.25	21.55	463.39	101.41
R=0 4WS	358.74	204.45	42.20	118.82	33.03	46.79	41.02	29.48	724.21	150.32
R=0 Skid*	1,405.09	799.95	1,258.99	820.52	0	0	0	0	4,284.56	0
R=1.5m 2WS	334.60	244.59	180.51	359.72	8.27	33.63	34.22	11.58	1,119.42	87.71
R=1.5m 4WS	243.15	249.41	161.11	272.54	12.73	28.95	17.97	4.71	926.22	64.37
R=1.5m Skid	11,995.51	1,202.26	3,199.60	6,947.69	0	0	0	0	23,345.07	0

Table A 9 Total Energy Consumption by Mode (with correction breakdown) - B400 x L600 Platform

Mode	Dur.(s)	Drive E (J)	Steer E (J)	Corr.Drive (J)	Corr.Steer (J)	Total (J)	Drive%	Steer%
R=0 2WS	7.41	397.49	133.18	324.66	39.74	895.07	44.4%	14.9%
R=0 4WS	7.68	558.00	147.42	324.66	39.74	1,069.82	52.2%	13.8%
R=0 Skid*	21.39	4,181.77	0	324.66	0	4,837.41 *	92.4%	0%
R=1.5m 2WS	9.39	1,054.09	132.10	-	-	1,186.18	88.9%	11.1%
R=1.5m 4WS	8.25	916.94	118.77	-	-	1,035.71	88.5%	11.5%
R=1.5m Skid	24.09	3,086.08	0	-	-	3,374.03	100%	0%

* R=0 Skid Steer failed - data recorded but manoeuvre not completed; excluded from efficiency comparisons.

Table A 10 Per-Motor Energy (J) for All Modes - B400 x L600 Platform

Mode	LR-D	RR-D	RF-D	LF-D	LR-S	RR-S	RF-S	LF-S	Drive Tot.	Steer Tot.
R=0 2WS	99.58	124.91	53.51	119.50	16.14	58.99	28.13	29.93	397.49	133.18
R=0 4WS	290.39	139.01	30.39	98.21	25.05	63.99	34.98	23.39	558.00	147.42
R=0 Skid*	1,711.22	538.15	1,485.17	447.23	0	0	0	0	4,181.77	0
R=1.5m 2WS	275.66	297.95	156.16	324.32	12.62	66.19	31.45	21.83	1,054.09	132.10
R=1.5m 4WS	238.26	228.45	155.28	294.95	14.13	63.38	26.16	15.10	916.94	118.77
R=1.5m Skid	873.55	957.06	534.85	720.62	0	0	0	0	3,086.08	0

Table A 11 Total Energy Consumption by Mode (with correction breakdown) - B600 x L800 Platform

Mode	Dur.(s)	Drive E (J)	Steer E (J)	Corr.Drive (J)	Corr.Steer (J)	Total (J)	Drive%	Steer%
R=0 2WS	8.97	535.77	107.74	346.31	19.78	1,009.60	53.1%	10.7%
R=0 4WS	7.65	534.42	108.39	346.31	19.78	1,008.90	53.0%	10.7%
R=0 Skid*	20.37	2,950.40	0	346.31	0	3,485.24 *	89.5%	0%
R=1.5m 2WS	9.75	1,247.00	97.41	-	-	1,344.40	92.8%	7.2%
R=1.5m 4WS	7.68	943.52	76.59	-	-	1,020.11	92.5%	7.5%
R=1.5m Skid	32.79	12,024.89	0	-	-	12,276.05	100.0%	2.0%

* R=0 Skid Steer failed - data recorded but manoeuvre not completed; excluded from efficiency comparisons.

Table A 12 Per-Motor Energy (J) for All Modes - B600 x L800 Platform

Mode	LR-D	RR-D	RF-D	LF-D	LR-S	RR-S	RF-S	LF-S	Drive Tot.	Steer Tot.
R=0 2WS	156.47	149.66	75.42	154.22	12.07	31.59	42.13	21.95	535.77	107.74
R=0 4WS	304.69	115.79	37.50	76.45	20.67	33.99	33.86	19.87	534.42	108.39
R=0 Skid*	810.61	718.96	704.50	716.33	0	0	0	0	2,950.40	0
R=1.5m 2WS	335.63	277.71	244.10	389.56	10.46	32.06	39.74	15.15	1,247.00	97.41
R=1.5m 4WS	244.30	221.36	185.36	292.49	11.47	29.40	27.61	8.10	943.52	76.59
R=1.5m Skid	5,914.63	602.73	1,609.71	3,897.82	0	0	0	0	12,024.89	0

Table A 13 Total Energy Consumption by Mode (with correction breakdown) - B800 x L400 Platform

Mode	Dur.(s)	Drive E (J)	Steer E (J)	Corr.Drive (J)	Corr.Steer (J)	Total (J)	Drive%	Steer%
R=0 2WS	6.42	341.59	72.70	343.21	21.25	778.74	43.9%	9.3%
R=0 4WS	6.51	276.22	68.90	343.21	21.25	709.57	38.9%	9.7%
R=0 Skid	8.55	1,786.03	0	343.21	0	2,215.98	83.6%	0%
R=1.5m 2WS	8.70	963.84	70.61	-	-	1,034.45	93.2%	6.8%
R=1.5m 4WS	7.32	834.72	60.15	-	-	894.87	93.3%	6.7%
R=1.5m Skid	9.21	2,360.24	0	-	-	2,426.99	100.0%	0%

Table A 14 Per-Motor Energy (J) for All Modes - B800×L400 Platform

Mode	LR-D	RR-D	RF-D	LF-D	LR-S	RR-S	RF-S	LF-S	Drive Tot.	Steer Tot.
R=0 2WS	162.36	58.70	21.64	98.89	9.02	20.95	26.79	15.93	341.59	72.70
R=0 4WS	92.03	101.08	24.10	59.01	13.84	25.37	21.18	8.50	276.22	68.90
R=0 Skid	763.64	218.59	616.97	186.83	0	0	0	0	1,786.03	0
R=1.5m 2WS	163.51	343.00	261.79	195.54	8.52	31.84	19.70	10.55	963.84	70.61
R=1.5m 4WS	142.14	305.15	222.12	165.32	9.48	29.34	15.96	5.37	834.72	60.15
R=1.5m Skid	1,430.65	59.83	278.23	591.53	0	0	0	0	2,360.24	0

Table A 15 Total Energy Consumption by Mode (with correction breakdown) - B400 x L400 Platform

Mode	Dur.(s)	Drive E (J)	Steer E (J)	Corr.Drive (J)	Corr.Steer (J)	Total (J)	Drive%	Steer%
R=0 2WS	8.91	386.22	72.93	367.99	18.14	845.27	45.7%	8.6%
R=0 4WS	6.45	249.17	71.66	367.99	18.14	706.95	35.2%	10.1%
R=0 Skid	14.52	2,725.89	0	367.99	0	3,205.70	87.9%	0%
R=1.5m 2WS	8.46	987.54	56.92	-	-	1,044.46	94.6%	5.4%
R=1.5m 4WS	8.25	891.85	59.20	-	-	951.05	93.8%	6.2%
R=1.5m Skid	13.08	3,215.92	0	-	-	3,297.09	100%	0%

Table A 16 Per-Motor Energy (J) for All Modes - B600 x L400 Platform

Mode	LR-D	RR-D	RF-D	LF-D	LR-S	RR-S	RF-S	LF-S	Drive Tot.	Steer Tot.
R=0 2WS	125.04	137.76	32.66	90.75	9.62	18.43	29.59	15.30	386.22	72.93
R=0 4WS	88.95	72.27	25.04	62.91	14.47	21.24	25.00	10.94	249.17	71.66
R=0 Skid	641.55	818.36	484.84	781.14	0	0	0	0	2,725.89	0
R=1.5m 2WS	341.22	247.27	122.62	276.44	6.72	21.08	22.48	6.64	987.54	56.92
R=1.5m 4WS	294.08	242.93	120.63	234.21	11.07	21.53	22.04	4.57	891.85	59.20
R=1.5m Skid	1,864.90	83.42	470.68	796.93	0	0	0	0	3,215.92	0

Table A 17 Per-Motor Energy (J) for All Modes - B800×L600 Platform

Mode	LR-D	RR-D	RF-D	LF-D	LR-S	RR-S	RF-S	LF-S	Drive Tot.	Steer Tot.
R=0 2WS	216.47	176.19	34.58	70.98	28.52	44.31	30.09	17.29	498.22	120.21
R=0 4WS	207.48	100.24	25.48	68.47	34.44	49.72	26.24	11.87	401.67	122.27
R=0 Skid	1,170.48	414.56	935.02	447.89	0	0	0	0	2,967.95	0
R=1.5m 2WS	463.89	199.22	160.51	301.79	29.67	52.31	23.63	5.99	1,125.41	111.60
R=1.5m 4WS	347.92	219.35	174.36	312.64	34.36	60.54	24.18	5.46	1,054.27	124.54
R=1.5m Skid	2,216.26	172.75	490.01	1,165.94	0	0	0	0	4,044.96	0

Table A 18 Top Three Configurations for Each Steering Mode (valid data only)

Mode	#1 Config	Energy (J)	#2 Config	Energy (J)	#3 Config	Energy (J)
R=0 2WS	B800×L400	778.74	B400×L400	807.88	B600×L400	845.27
R=0 4WS	B600×L400	706.95	B800×L400	709.57	B800×L600	842.39
R=0 Skid*	B800×L400	2,215.98	B600×L400	3,205.70	B800×L600	3,450.16
R=1.5m 2WS	B800×L400	1,034.45	B600×L400	1,044.46	B400×L400	1,103.00
R=1.5m 4WS	B800×L400	894.87	B600×L400	951.05	B400×L800	990.58
R=1.5m Skid	B800×L400	2,426.99	B600×L400	3,297.09	B400×L600	3,374.03

Table A 19 Per-Motor Energy (J) for All Modes - B800×L400 Platform (18° Slope)

Mode	LR-D	RR-D	RF-D	LF-D	LR-S	RR-S	RF-S	LF-S	Drive Tot.	Steer Tot.
R=0 2WS	165.31	164.21	144.67	162.62	9.03	52.60	36.23	14.03	636.81	111.89
R=0 4WS	135.98	177.73	106.12	166.23	18.19	67.82	29.40	10.24	586.06	125.65
R=0 Skid Steer	510.73	154.30	456.73	155.23	12.68	36.67	34.39	4.47	1,276.99	88.21
R=1.5m 2WS	406.79	741.59	382.34	208.97	10.47	57.97	23.58	9.19	1,739.69	101.21
R=1.5m 4WS	421.66	721.89	335.58	195.86	11.77	56.81	19.99	8.40	1,674.99	96.97
R=1.5m Skid	515.62	1,811.37	531.40	331.97	8.86	41.05	22.02	6.33	3,190.36	78.26

Table A 20 Total Energy Consumption by Steering Mode - B800×L600 Platform (18° Slope)

Mode	Dur.(s)	Drive E (J)	Steer E (J)	Correction (J)	Total (J)	Drive%	Steer%
R=0 2WS	15.45	966.28	237.13	+316.89	1,520.30	63.6%	15.6%
R=0 4WS	9.00	611.09	123.70	+316.89	1,051.68	58.1%	11.8%
R=0 Skid Steer	19.23	3,043.95	0	+316.89	3,504.40	91%	0%
R=1.5m 2WS	9.36	1,921.46	87.31	-	2,008.77	95.7%	4.3%
R=1.5m 4WS	7.92	1,541.99	83.95	-	1,625.94	94.8%	5.2%
R=1.5m Skid	20.58	5,298.39	0	-	5,427.66	100%	0%

Table A 21 Total Energy Consumption by Steering Mode - B800×L800 Platform (18° Slope)

Mode	Dur.(s)	Drive E (J)	Steer E (J)	Correction (J)	Total (J)	Drive%	Steer%
R=0 2WS	10.23	1,029.49	109.58	+359.23	1,498.31	68.7%	7.3%
R=0 4WS	10.47	770.52	121.41	+359.23	1,251.16	61.6%	9.7%
R=0 Skid Steer	21.81	4,964.80	0	+359.23	5,486.59	93.5%	0%
R=1.5m 2WS	7.56	1,936.08	59.51	-	1,995.59	97.0%	3.0%
R=1.5m 4WS	8.34	1,758.20	80.54	-	1,838.74	95.6%	4.4%
R=1.5m Skid	33.36	9,118.14	0	-	9,354.61	100%	0%

Table A 22 Per-Motor Energy (J) for All Modes - B800×L800 Platform (18° Slope)

Mode	LR-D	RR-D	RF-D	LF-D	LR-S	RR-S	RF-S	LF-S	Drive Tot.	Steer Tot.
R=0 2WS	267.52	210.21	272.48	279.27	12.52	32.37	46.63	18.07	1,029.49	109.58
R=0 4WS	252.04	209.87	106.79	201.82	24.20	42.17	41.42	13.64	770.52	121.41
R=0 Skid Steer	1,454.72	920.73	1,535.69	1,053.65	0	0	0	0	4,964.80	0
R=1.5m 2WS	441.89	646.55	505.00	342.64	5.98	19.75	23.53	10.25	1,936.08	59.51
R=1.5m 4WS	431.41	582.03	435.15	309.60	12.82	30.43	26.15	11.14	1,758.20	80.54
R=1.5m Skid	1,148.14	4,406.71	2,386.82	1,176.47	0	0	0	0	9,118.14	0

Table A 23 Slope Overall Energy Ranking - Best Mode per Configuration

Rank	Config	L(mm)	B/L	Best E (J)	Mode
1	B800×L600	600	1.33	1,051.68	R=0 4WS
2	B800×L400	400	2.00	1,076.21	R=0 4WS
3	B800×L800	800	1.00	1,251.16	R=0 4WS

Table A 24 Best Configuration per Steering Mode - 18° Slope (Top Three)

Mode	#1	E (J)	#2	E (J)	#3	E (J)
R=0 2WS	B800×L400	1,113.20	B800×L800	1,498.31	B800×L600	1,520.30
R=0 4WS	B800×L600	1,051.68	B800×L400	1,076.21	B800×L800	1,251.16
R=0 Skid	B800×L400	1,729.70	B800×L600	3,504.40	B800×L800	5,486.59
R=1.5m 2WS	B800×L400	1,840.90	B800×L800	1,995.59	B800×L600	2,008.77
R=1.5m 4WS	B800×L600	1,625.94	B800×L400	1,771.96	B800×L800	1,838.74
R=1.5m Skid	B800×L400	3,268.62	B800×L600	5,427.66	B800×L800	9,354.61

Appendix B

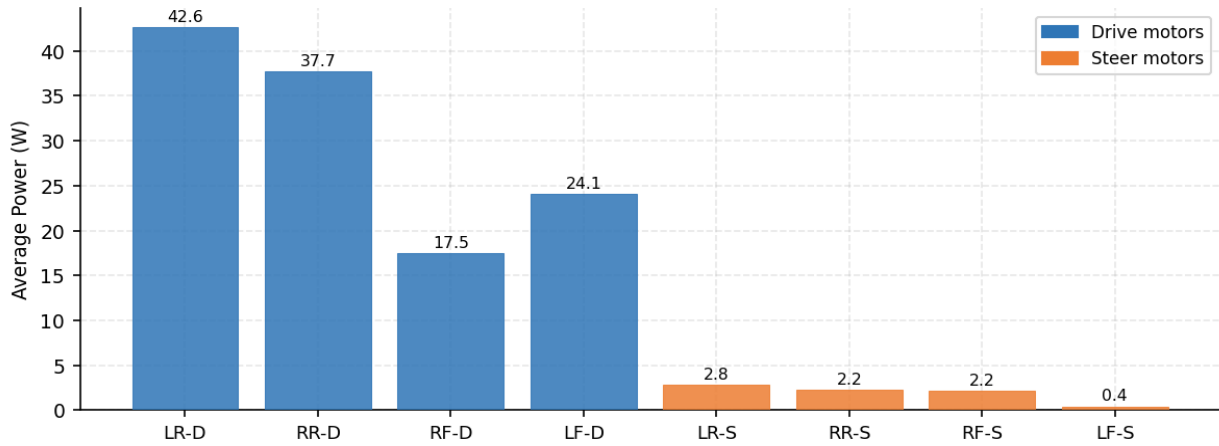


Figure B 1 Straight-drive average power per motor - B600 x L600 platform

During straight-line travel, the overall average power is 129.58 W, of which drive motors consume 121.92 W (94.1%) and steer motors 7.66 W (5.9%). The rear-axle pair consumes 65.9% of total drive power, indicating a rearward longitudinal weight bias under steady-state rolling conditions. Lateral load distribution is well balanced (left: 54.7%, right: 45.3%).

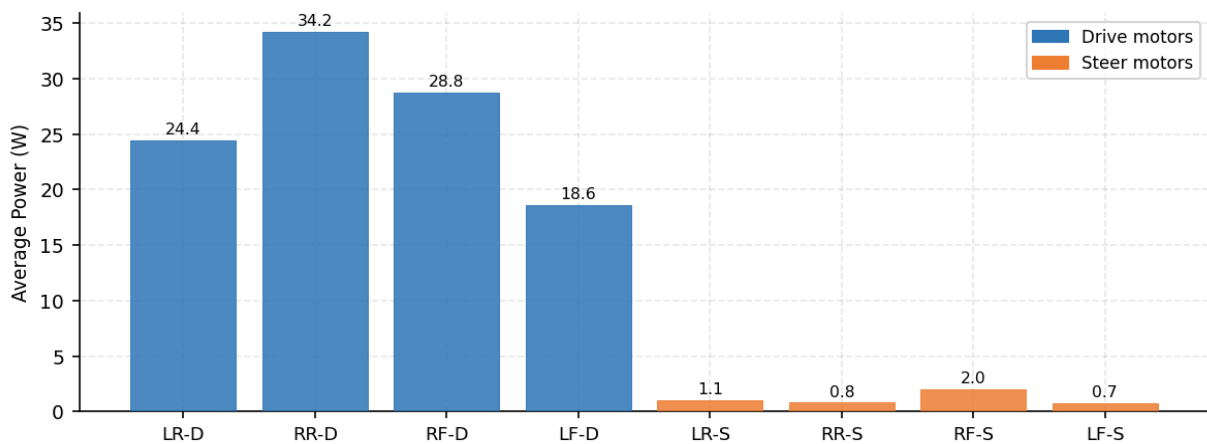


Figure B 2 Straight-drive average power per motor - B800 x L600 platform

During straight-line travel, the overall average power is 110.58 W, of which drive motors consume 105.97 W (95.8%) and steer motors 4.62 W (4.2%). The rear-axle pair consumes 55.3% of total drive power, indicating a rearward longitudinal weight bias under steady-state rolling conditions. A lateral imbalance is also observed: the right-side pair draws 59.4% of drive power, which may indicate a lateral weight distribution asymmetry worth noting for future structural refinement.

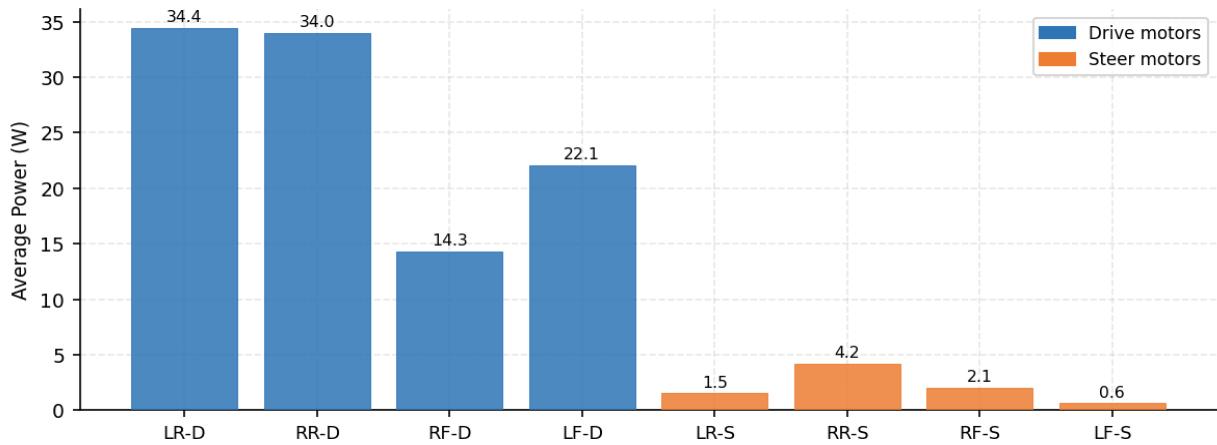


Figure B 3 Straight-drive average power per motor - B400 x L400 platform

During straight-line travel shown in Figure B 3, the overall average power is 113.21 W, of which drive motors consume 104.76 W (92.5%) and steer motors 8.45 W (7.5%). The rear-axle pair consumes 65.3% of the total drive power, which is substantially higher than that of the front axle. This may be attributed to a rearward weight bias of the vehicle, as the battery compartment, which represents a significant proportion of the vehicle mass, is located towards the rear. The increased normal load on the rear wheels may generate greater available frictional force, enabling the rear motors to contribute a larger proportion of the driving torque.

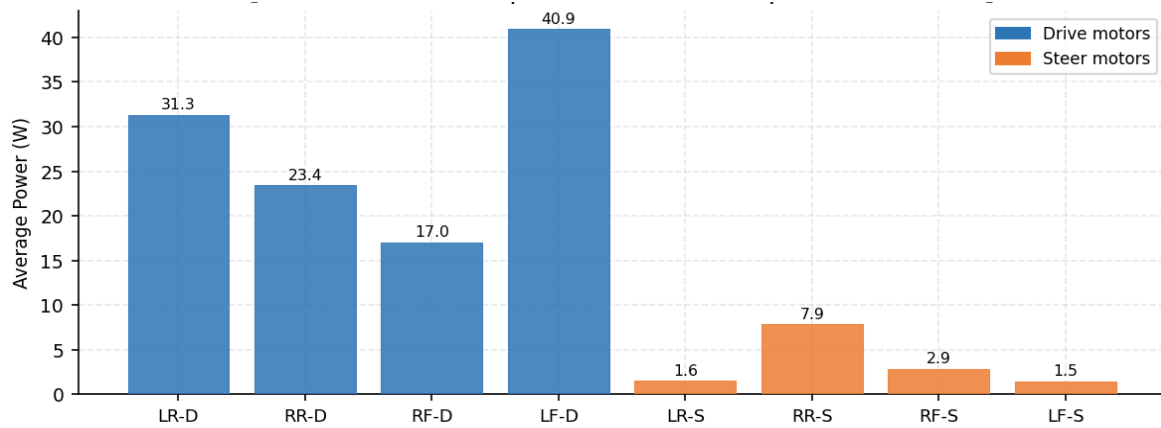


Figure B 4 Straight-drive average power per motor - B400 x L600 platform

During straight-line travel, the overall average power is 126.53 W, of which drive motors consume 112.73 W (89.1%) and steer motors 13.80 W (10.9%). The rear-axle pair consumes 56.5% of total drive power, indicating a rearward longitudinal weight bias under steady-state rolling conditions. (rear: 56.5%, front: 43.5%).

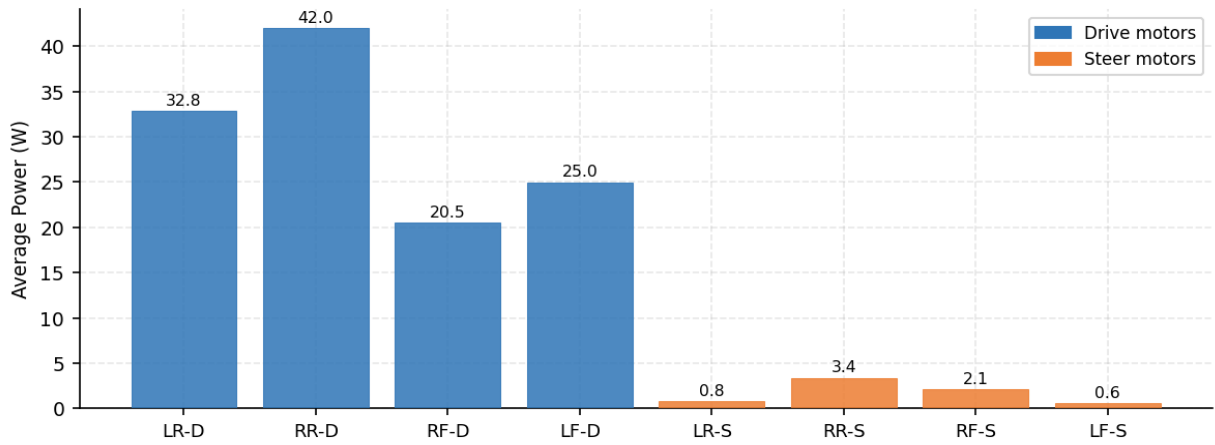


Figure B 5 Straight-drive average power per motor - B600 x L800 platform

During straight-line travel, the overall average power is 127.14 W, of which drive motors consume 120.27 W (94.6%) and steer motors 6.87 W (5.4%). The rear-axle pair consumes 62.2% of total drive power, indicating a rearward longitudinal weight bias under steady-state rolling conditions. Lateral load distribution is well balanced (left: 48.1%, right: 51.9%).

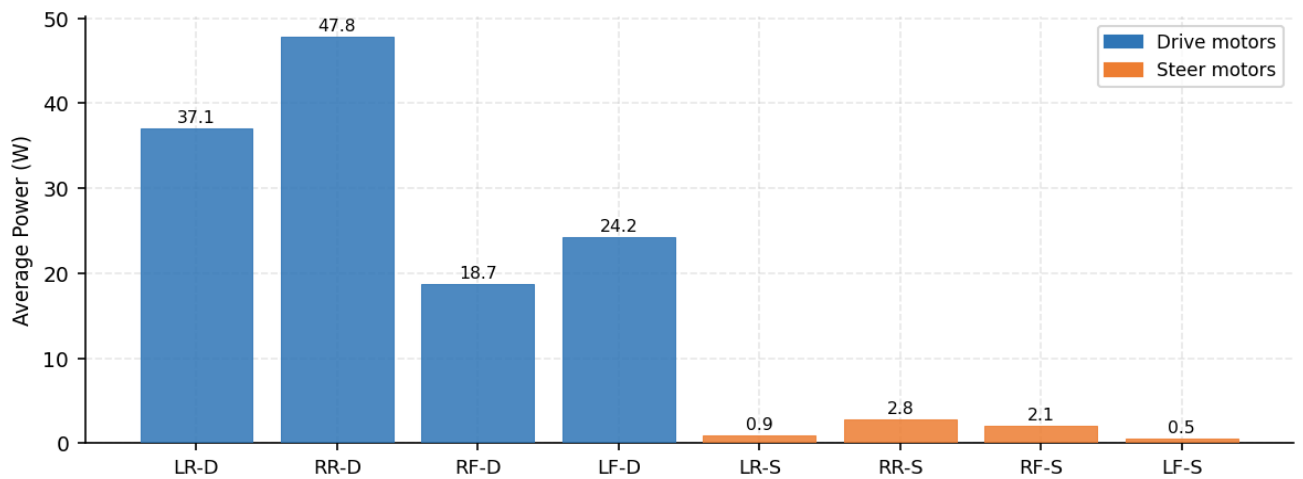


Figure B 6 Straight-drive average power per motor - B600 x L400 platform

During straight-line travel, the overall average power is 134.09 W, of which drive motors consume 127.79 W (95.3%) and steer motors 6.30 W (4.7%). The rear-axle pair consumes 66.4% of total drive power, indicating a rearward longitudinal weight bias under steady-state rolling conditions. Lateral load distribution is close (left: 48.0%, right: 52.0%).

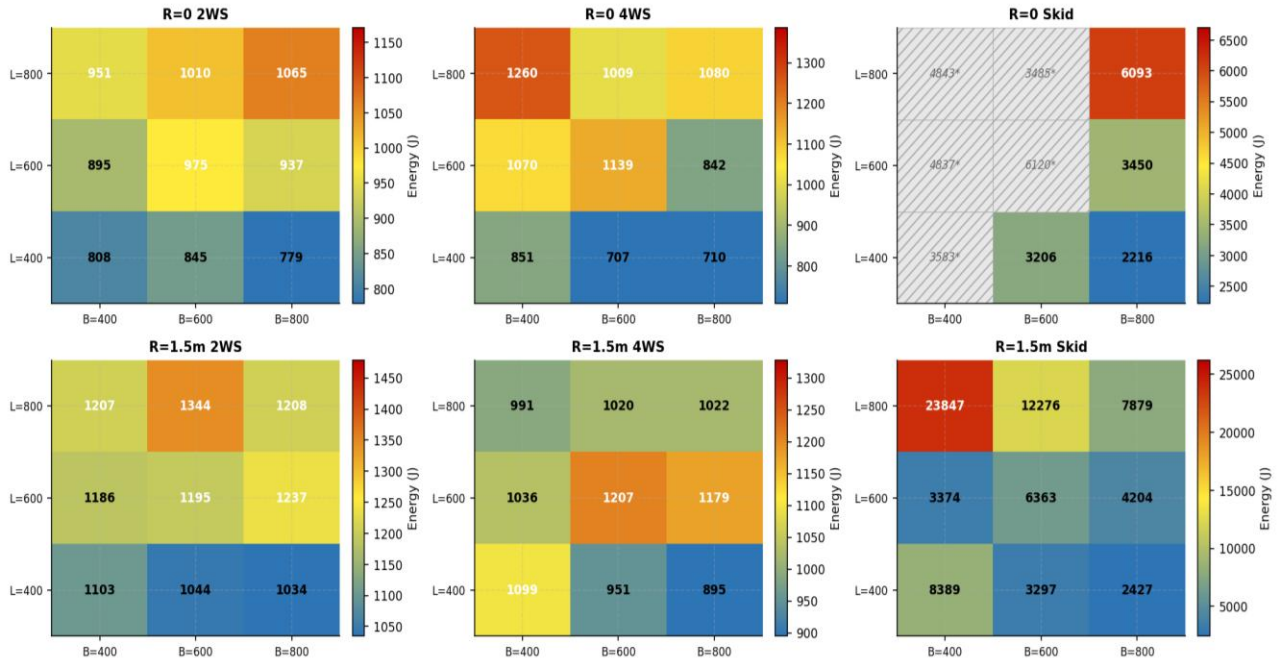


Figure B 7 Energy heatmap across the 3×3 $B \times L$ design space for each steering mode. Colour scale from blue (low energy, efficient) to red (high energy, inefficient). Grey hatching = failed $R=0$ Skid Steer tests.

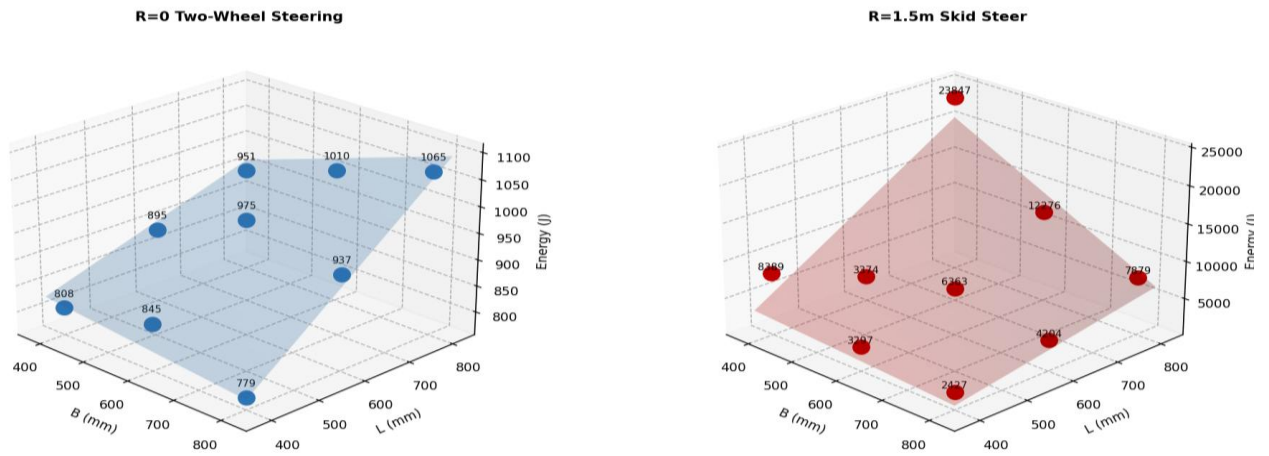


Figure B 8 Three-dimensional energy surface in $B \times L$ parameter space for $R=0$ 2WS (left) and $R=1.5m$ Skid Steer (right). Data points (spheres) are overlaid on the fitted surface (translucent). The $R=1.5m$ Skid Steer surface rises sharply as L increases and B decreases.

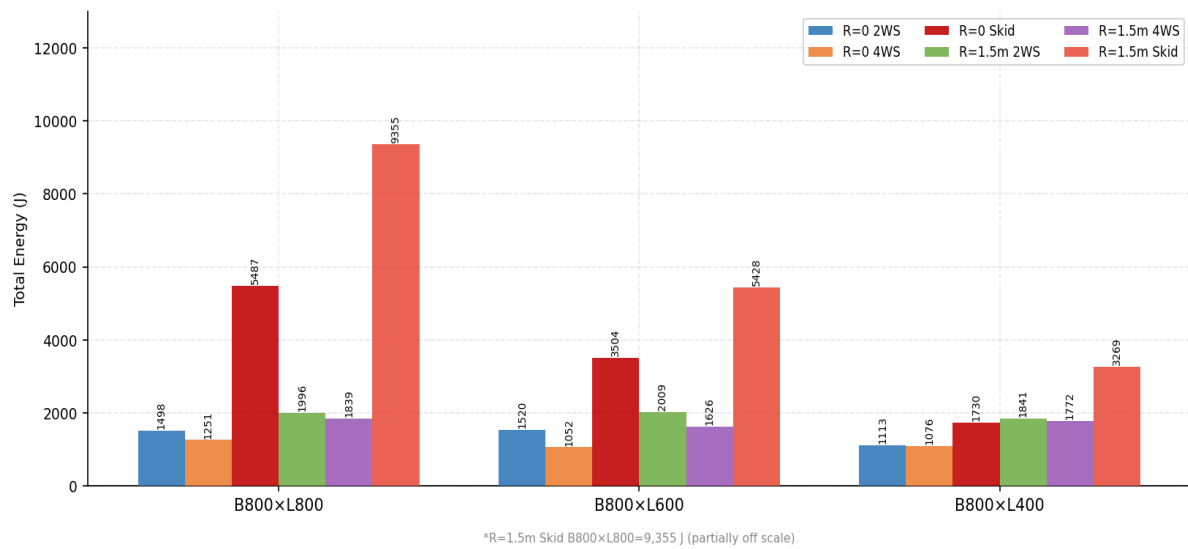


Figure B 9 Full slope energy overview - all configurations and modes ($B=800\text{mm}$; L varies). $R=1.5\text{m}$ Skid $B800 \times L800=9,355\text{ J}$ partially off scale.

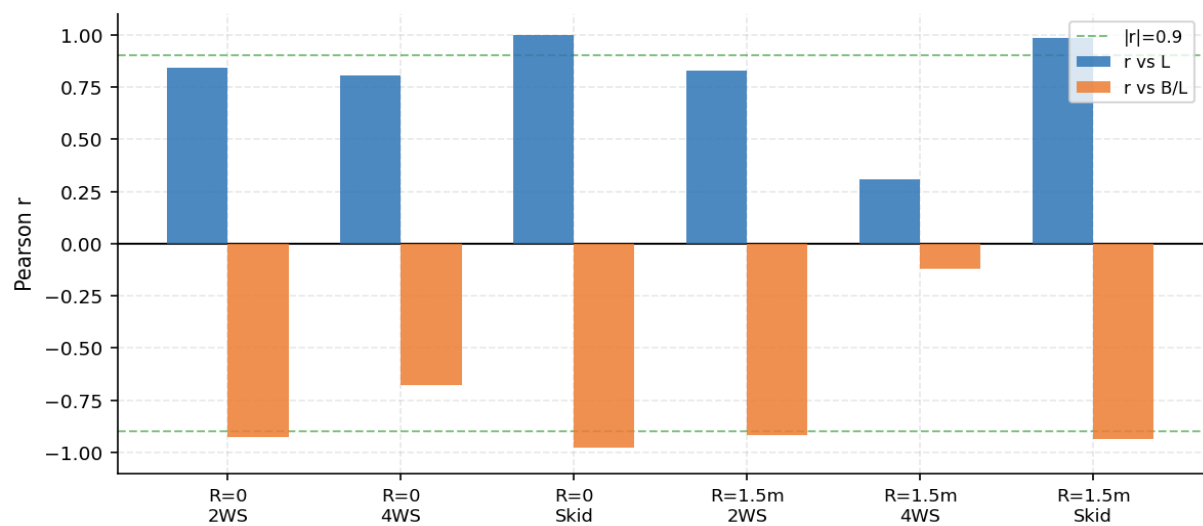


Figure B 10 Pearson correlation comparison - energy vs L and B/L . Blue= $r(L)$, orange= $r(B/L)$; green dashed= $|r|=0.9$ threshold.