

Article

A Simulation-Based Assessment of Energy Flow, Efficiency, and Emissions in a Battery Electric Vehicle

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Abstract

This study investigates the performance, energy flow, efficiency, and environmental impact of a C-segment battery electric vehicle (BEV). As BEVs are increasingly considered a sustainable alternative to conventional internal combustion engine vehicles, a detailed understanding of their energy utilization and operational emissions is essential. A MATLAB/Simulink-based vehicle model incorporating an 88.5 kWh battery pack, a 160 kW permanent magnet synchronous motor (PMSM), regenerative braking, and longitudinal vehicle dynamics was developed. The developed model was validated by comparing the simulated vehicle performance characteristics with the publicly available specifications and performance data of the reference TOGG T10F vehicle. The vehicle was evaluated under the WLTP Class 3 driving cycle, while the effects of aggressive and high-speed driving conditions were further investigated using the US06 and Artemis Motorway 150 cycles. The results indicate a net vehicle energy consumption of 136.4 Wh/km and a driving range of 623 km under WLTP conditions. The PMSM achieved average efficiencies of 93.4% in traction mode and 92.7% in regenerative braking mode, while the cumulative battery-to-wheel drivetrain efficiency reached 81.5%. In addition, approximately 19.9% of the consumed energy was recovered through regenerative braking. Vehicle emissions were also assessed using different electricity generation mixes based on the rated energy consumption, including charging losses, yielding operational emissions between 27.7 and 116.0 gCO₂e/km. The findings demonstrate that the developed model provides realistic performance predictions and confirm the potential of BEVs to achieve high efficiency and substantially lower emissions than conventional passenger vehicles.

Keywords: driving cycle analysis; electric vehicle; energy efficiency; emissions; vehicle performance



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1. Introduction

The transportation sector is one of the major contributors to global energy consumption and greenhouse gas emissions because the extensive use of internal combustion engine vehicles (ICEVs) in passenger mobility [1,2]. Increasing concerns about climate change, fossil fuel dependency, and urban air pollution have accelerated the transition toward low-emission and energy-efficient transportation technologies [3,4]. Among these alternatives, Electric Vehicles (EVs) have gained significant attention owing to their low emissions, high drivetrain efficiency, low noise, and compatibility with renewable electricity generation [5–8]. However, the real environmental and operational benefits of EVs depend both

on their electrified powertrain structure and on how efficiently the stored battery energy is converted into useful traction energy under different driving conditions [9,10].

Battery Electric Vehicle (BEV) performance cannot be fully explained by general indicators such as battery capacity, nominal driving range, or zero emissions alone. Battery energy demand, electric motor operating region, drivetrain losses, and the amount of energy recoverable through regenerative braking should be evaluated together under different driving conditions [11,12]. In particular, standard, aggressive, and high-speed driving cycles provide an important reference for comparatively examining the effects of vehicle speed, acceleration characteristics, and deceleration behavior on energy consumption and system efficiency [13]. Moreover, although EVs do not produce direct emissions during operation, their environmental contribution may vary depending on the electricity generation mix used for charging [14]. This highlights the need for a comprehensive assessment of energy flow, efficiency, and operational emission performance in BEVs. The following studies are reviewed to clarify the current research focus and to identify the remaining gaps in BEV energy consumption, efficiency, regenerative braking, and emission analyses.

Recent studies have investigated the performance and environmental impacts of different types of vehicles from various perspectives, including energy consumption, component-level efficiency, simulation-based performance evaluation, regenerative braking, and emission analysis. Accordingly, the reviewed studies are synthesized below by grouping them into these major research themes to provide a clearer understanding of current research trends.

Choi et al. (2025) analyzed the effects of vehicle weight, road gradient, speed, and acceleration on the energy consumption of a rear-wheel-drive BEV equipped with a 77.4 kWh battery and a 168 kW IPMSM using real-world driving tests under three driving cycles and MATLAB/Simulink simulations [15]. Feng et al. (2025) experimentally investigated the energy flow of a four-wheel-drive BEV equipped with a 60 kWh battery under NEDC, WLTC, and CLTC [16]. Energy consumption values and ranges were reported. Seo (2025) analyzed the energy consumption, driving range, and component-level energy flow of a BEV equipped with a 75 kWh battery and a 160 kW electric motor under urban and highway driving conditions [17]. The study used UDDS, HWFET, and US06 cycles. Grunditz and Thiringer (2016) investigated the energy consumption, driving range, and powertrain efficiency of a small BEV equipped with a 19.3 kWh battery under various official and Gothenburg real-world driving cycles [18]. Mavlonov et al. (2023) investigated the sensitivity of BEV energy consumption to electric motor size and efficiency map characteristics using BMW i3 and Kia Soul vehicle models [19]. The study compared BMW i3, Kia Soul, and YASA electric motors under the UDDS cycle. Collectively, these studies demonstrate that vehicle operating conditions, powertrain configuration, motor characteristics, and component-level energy flow are among the primary factors influencing BEV energy consumption and driving range.

Beyond component-oriented analyses, another group of studies has focused on developing and validating simulation models for predicting BEV performance under different operating conditions. Guo et al. (2024) proposed a novel framework for constructing and evaluating EV driving cycles based on energy consumption and emission analysis [20]. The study considered a BEV with a 38 kWh battery and developed a representative driving cycle. The proposed framework integrates a dual-chain Markov model with a double-layer evaluation model to improve the representativeness of EV driving cycles. Martyushev et al. (2023) mathematically modeled the energy consumption and driving range of a Nissan Leaf-based BEV equipped with a 24 kWh battery and an 80 kW synchronous motor under NEDC, WLTC, JC08, and US06 cycles using MATLAB/Simulink [21]. Xie et al.

(2020) investigated the energy consumption and driving range of a PBEV equipped with a single-pedal control system, a 54.63 kWh Li-ion battery, and a 60 kW PMSM using an experimentally calibrated simulation model [22]. The calibrated model was further used to evaluate the effects of regenerative braking strategies and single-pedal control on vehicle energy consumption and driving range under different driving conditions. Luin et al. (2019) developed a Vehicle-Specific Power-based microsimulation model to estimate EV energy consumption and calibrated it using Nissan Leaf data under driving cycles [23]. Kumar et al. (2023) developed a PMSM-based small EV model in MATLAB/Simulink and analyzed torque, power, and energy consumption under driving cycles [24]. Miri et al. (2021) developed a MATLAB/Simulink-based energy consumption and range estimation model for a BMW i3 60 Ah REx EV equipped with a 22 kWh Li-ion battery and a 125 kW electric motor [25]. The model also incorporated component efficiency maps, auxiliary power consumption, and a regenerative braking strategy, and was validated against published experimental energy consumption data. These studies highlight the importance of validated simulation frameworks for accurately estimating energy consumption, driving range, and overall vehicle performance under standardized driving cycles.

In addition to vehicle modeling and performance evaluation, recent research has increasingly concentrated on improving energy recovery during braking. Prasad and Gudipalli (2024) proposed an energy consumption reduction strategy for a front-wheel-drive EV based on ideal braking force distribution and regenerative braking force control [26]. The study considered a 15 kWh Li-ion battery and a 1.5 kW electric motor, reporting energy consumption values. The proposed control strategy was validated using MATLAB/Simulink and a real-time Opal-RT 5700 platform, demonstrating improved energy efficiency through optimized regenerative braking. Sandrini et al. (2022) developed a regenerative braking logic for front-wheel, rear-wheel, and all-wheel drive compact EVs to maximize energy recovery while preserving vehicle stability [27]. Ma and Sun (2020) proposed a regenerative braking energy recovery strategy based on ideal braking force distribution for a small four-wheel-drive EV [28]. The proposed strategy was validated under different braking conditions and demonstrated effective energy recovery while maintaining braking stability. Yin et al. (2023) proposed a regenerative braking strategy based on dynamic logic threshold control for a front-wheel-drive PEV [29]. The study evaluated driving cycles in MATLAB/Simulink, reporting braking energy recovery efficiencies. These studies indicate that advanced regenerative braking control strategies can significantly improve braking energy recovery while maintaining vehicle stability and enhancing overall vehicle efficiency.

Besides improving vehicle performance, another important research direction has focused on evaluating the environmental impacts of different types of vehicles under varying operating conditions and electricity generation mixes. Boruah et al. (2026) investigated the effects of region-specific EV driving cycles on energy consumption, driving range, and well-to-wheel emissions in two Indian cities using a Tata Tigor EV equipped with a 26 kWh battery [30]. Kannangara et al. (2021) developed an adaptable life cycle greenhouse gas emission assessment framework for ICEV, HEV, PHEV, BEV, and FCEV light-duty vehicles in the Canadian context [31]. The representative BEV was modeled with a 46 kWh battery, 16.85 kWh/100 km electricity consumption, and 295 km range. Sheng et al. (2021) compared the well-to-wheel energy consumption and greenhouse gas emissions of ICEVs, BEVs, PHEVs, and FCEVs in Australia and New Zealand [32]. The study also projected future vehicle uptake and well-to-wheel greenhouse gas emissions under different electricity generation mix scenarios until 2050. Woo et al. (2017) compared the well-to-wheel greenhouse gas emissions of BEVs and ICEVs from a global perspective by considering the electricity generation mix of 70 countries [33]. These studies demonstrate that the environmental performance of different vehicle technologies depends strongly on

electricity generation mixes, regional conditions, and life-cycle assessment assumptions rather than vehicle operation alone.

Although considerable progress has been achieved in each of these research areas, most previous studies have focused on individual aspects of different types of vehicles rather than integrating them into a comprehensive evaluation framework. In Table 1, it can be seen that most existing studies address energy consumption, efficiency, regenerative braking, or emissions individually, whereas integrated analyses combining all these aspects within a unified BEV modeling framework remain scarce.

Table 1. Literature comparison of energy consumption, efficiency, regenerative braking, and emission studies across different vehicle powertrain technologies.

Ref.	Vehicle Type	Driving Cycle	Energy Consumption	Efficiency Analysis	Regenerative Braking Analysis	Emission Analysis
[15]	Rear-wheel drive BEV	Real-world 61 km, UDDS, HWFET, US06, SC03, WLTP	✓	✓	✓	-
[16]	Four-wheel drive EV	NEDC, WLTC, CLTC, 120 km/h constant speed	✓	✓	✓	-
[17]	Mid-size BEV	UDDS, HWFET, US06	✓	✓	✓	-
[18]	Light-duty BEV	NEDC, FTP72, UDDS, HWFET, US06, SC03, JC08, WLTC, Artemis	✓	✓	✓	-
[19]	BMW i3 and Kia Soul BEVs	UDDS	✓	✓	-	-
[20]	BEV	RDC, WLTC, CLTC	✓	-	-	✓
[21]	BEV based on Nissan Leaf	NEDC, WLTC, JC08, US06	✓	-	-	-
[22]	PBEV with single-pedal control system	NEDC, WLTC, FTP75, CLTC, US06, JC08	✓	✓	✓	-
[23]	EV model calibrated using Nissan Leaf	UDDS, HWY, WLTP, NEDC	✓	-	✓	-
[24]	Small EV model	EUDC, NEDC	✓	-	-	-
[25]	BMW i3 Range Extender-based EV	NEDC, FTP-75, HWFET, SC03, US06	✓	✓	✓	-
[26]	Four-wheel drive EV	ECE R15, IDC	✓	✓	✓	-
[27]	BEVs	WLTC, US06	✓	-	✓	-
[28]	Small four-wheel drive EV	-	-	-	✓	-
[29]	PEV	NEDC, WLTC, FTP-72, FTP-75, CLTC-P, NYCC	✓	-	✓	-
[30]	Electric sedan Tata Tigor EV	H-ECDC, B-ECDC, MIDC, WLTC, NEDC	✓	-	-	✓
[31]	ICEV, HEV, PHEV, BEV and FCEV	-	✓	-	-	✓
[32]	ICEV, BEV, PHEV and FCEV	WLTC	✓	-	-	✓
[33]	BEV and ICEV	-	-	-	-	✓
This study	C-segment BEV based on TOGG T10F platform	WLTP Class 3, US06, Artemis Motorway 150	✓	✓	✓	✓

In this study, the MATLAB/Simscape EV Reference Application developed by MathWorks was adopted as the simulation framework and parameterized according to the publicly available technical specifications of the TOGG T10F Long Range RWD vehicle [34]. The resulting model was subsequently used to analyze the vehicle's energy flow, efficiency, regenerative braking performance, and operational emissions under different driving conditions. Since the modeled vehicle represents a mid-size passenger vehicle and its power-to-mass ratio exceeds 34 W/kg, the Worldwide Harmonized Light Vehicle Test Procedure (WLTP) Class 3 driving cycle was selected for the simulations. The developed vehicle model includes a lithium-ion battery pack, PMSM traction motor, fixed-ratio transmission system, regenerative braking system, and longitudinal vehicle dynamics. Unlike most existing MATLAB/Simulink-based BEV simulation studies, which primarily focus on

individual aspects such as energy consumption, driving range, regenerative braking, or emission analysis, the proposed model integrates detailed energy flow analysis, component-level efficiency evaluation, regenerative braking performance, sensitivity analysis under different driving cycles, and operational emission assessment within a single simulation framework. Therefore, the developed model provides a comprehensive evaluation of BEV performance based on a commercially available C-segment battery electric vehicle. In order to obtain realistic results, a commercially available C-segment EV configuration widely preferred in real-world applications was selected as the reference vehicle. All propulsion system components were modeled and sized according to the target vehicle specifications and operating requirements. As discussed in the literature review, most existing MATLAB/Simulink-based BEV studies primarily investigate vehicle energy consumption, driving range, regenerative braking, or emissions separately. Comprehensive simulation frameworks that simultaneously analyze energy flow, component-level efficiencies, regenerative braking performance, sensitivity to different driving cycles, and operational emissions remain limited. This study addresses this gap by integrating all these analyses into a unified MATLAB/Simulink model developed for a commercially available C-segment battery electric vehicle. Therefore, one of the major contributions of this study to the literature is the comprehensive analysis of energy conversion processes throughout the entire propulsion system. The main contributions of the study can be summarized as follows:

- Firstly, the electrical energy supplied by the battery, auxiliary system consumption, and net electrical energy transferred to the PMSM traction motor are determined throughout the driving cycle.
- The progression of energy through the propulsion system, from the battery to PMSM, drivetrain, and wheels, is analyzed step-by-step using simulation results.
- The energy flow is separately investigated for both traction and regenerative braking modes, and the operational efficiencies of the PMSM in motor and generator modes are calculated.
- The effects of different driving cycles, vehicle mass, and aerodynamic characteristics on energy consumption and driving range are evaluated through sensitivity analyses.
- Finally, the overall propulsion system efficiency from the battery to the wheels and the regenerative energy recovery efficiency from the wheels back to the battery are determined considering all drivetrain losses and battery energy variations during the driving cycle.

The obtained simulation results provide a detailed understanding of the energy distribution and efficiency characteristics throughout the propulsion system under different operating conditions. In addition, the power flow paths occurring during both traction and regenerative braking operations are analyzed and discussed in detail. Finally, by considering the emission intensities of different electricity generation mixes, the environmental impact and operational greenhouse gas emissions of the BEV are evaluated for various grid scenarios.

2. Materials and Methods

Vehicle dynamics describe the motion behavior of a vehicle under the influence of forces such as tire–road interaction, aerodynamic drag, and gravity. A solid understanding of these dynamics is crucial in the powertrain development stage because it helps determine the magnitude and characteristics of the loads that the propulsion system must withstand during operation. Moreover, vehicle dynamics play a significant role in assessing performance indicators including acceleration capability, driving efficiency, and overall energy usage. In this study, the analysis focuses on drivetrain loading conditions, the propulsion power required by the electric motor, the power supplied by the battery system, and the

overall energy consumption characteristics of the BEV. For simplification purposes, the vehicle structure is represented as a rigid body concentrated at a single mass point located at the center of gravity. Additionally, the analysis considers only longitudinal motion in the forward direction while assuming that lateral stability of the vehicle is preserved. The simulations are carried out in the MATLAB Simscape environment integrated with Simulink (R2021b, MathWorks, Natick, MA, USA), which provides an efficient framework for modeling and simulating physical systems. Furthermore, the simulation results were compared with the performance data of the reference TOGG T10F vehicle to evaluate the accuracy and validity of the developed model.

2.1. Vehicle Model

The force transferred between the tires and the road surface represents the primary tractive force acting on the vehicle. During downhill motion, gravitational force may contribute positively to vehicle propulsion, whereas it becomes a significant opposing force during uphill driving conditions. Additional resistance components include rolling resistance, aerodynamic drag, regenerative braking torque, and conventional mechanical braking forces. Vehicle acceleration occurs when the total tractive force exceeds the sum of all resistive forces; conversely, the vehicle decelerates when the resistive forces become dominant. Under constant-speed operation, the tractive and resistive forces must remain in equilibrium. Therefore, in order to estimate the propulsion power demand of the vehicle, the total tractive force (F_{tf}) required to overcome these resistive components should first be determined as expressed in Equation (1) [35].

$$F_{tf} = \frac{1}{2} \rho C_d A_F v^2 + W C_{rr} \cos \alpha + W \sin \alpha + aM \quad (1)$$

where ρ is the air density, C_d is the air drag coefficient, A_F is the frontal area, v is the velocity, W is the weight, C_{rr} is the rolling resistance coefficient, α is the slope of the road, a is acceleration and M is the mass with rotational inertia. The developed mathematical model is based on several simplifying assumptions. Vehicle motion is considered only in the longitudinal direction, while the road gradient is assumed to remain constant during the driving cycle. In addition, rolling resistance and effective wheel radius are treated as independent of vehicle speed, and effects such as tire slip and mechanical vibrations are neglected. Using this modeling approach, the propulsion power demand of the vehicle presented in Table 2 was calculated for vehicle speeds ranging from 0 to 200 km/h under both flat-road conditions and a 6% road gradient. The 6% slope value was selected because it represents the maximum highway grade commonly permitted in many countries, excluding certain low-speed or specially regulated road sections [36].

The maximum speed capability is a highly critical parameter for BEVs, and therefore this value is intentionally limited in many production models. During long-distance highway driving, continuous operation at high speeds has a significant negative impact on driving range. The primary reason for this is the substantial increase in vehicle power demand caused by aerodynamic drag force, which rises rapidly with increasing vehicle speed. Consequently, the battery is exposed to prolonged high-power demands that may exceed its nominal operating power, leading to accelerated battery depletion and a reduction in available capacity [37]. Furthermore, since most BEVs employ a single-speed reduction gearbox, the electric motor is required to operate at very high rotational speeds during high-speed driving conditions, which may also reduce motor efficiency and further increase overall energy consumption [38]. In addition to maximum cruising speed, vehicle performance is also characterized by acceleration capability and gradeability. Among these parameters, the performance factor (d_p) plays a significant role in evaluating the vehicle's

climbing capability on inclined roads. The performance factor can be determined using Equation (2) [39].

$$d_p = \frac{\tau_m i_g \eta_d / r - W C_{rr} - 1/2 \rho C_d A_f v^2}{W} \quad (2)$$

where τ_m is the motor torque, i_g is the reduction gear ratio, η_d is the drivetrain efficiency and r is the tire radius. In Equation (2), the first term in the numerator corresponds to the tractive force generated at the wheels by the motor torque, whereas the second and third terms represent the rolling resistance force and aerodynamic drag force, respectively. Using these force components, the gradeability of a vehicle operating on a steeply inclined road can be determined by Equation (3) [39].

$$\sin \alpha = \frac{d_p - C_{rr} \sqrt{1 - d_p^2 + C_{rr}^2}}{1 + C_{rr}^2} \quad (3)$$

where α is the slope angle. As indicated by Equations (2) and (3), the gradeability of a vehicle is directly related to the motor torque. Another important vehicle performance criterion is acceleration capability, which is commonly evaluated by the time required for the vehicle to accelerate from 0 to 100 km/h on a flat road surface. The acceleration (a) of the vehicle can be calculated using Equation (4) [39].

$$a = \frac{dv}{dt} \quad (4)$$

By rearranging Equation (4) in terms of time, Equation (5) is obtained, and integrating this expression over the velocity leads to Equation (6).

$$dt = \frac{1}{a} dv \quad (5)$$

$$t = \int \frac{1}{a} dv \quad (6)$$

According to Newton's second law of motion, the acceleration term can be redefined as given in Equation (7) [39].

$$a = \frac{\tau_m i_g \eta_d / r - W C_{rr} - 1/2 \rho C_d A_f v^2}{M} \quad (7)$$

Substituting the acceleration expression given in Equation (7) into Equation (6) results in Equation (8), which is used to calculate the 0–100 km/h acceleration time of the vehicle.

$$t = \int \frac{M}{\tau_m i_g \eta_d / r - W C_{rr} - 1/2 \rho C_d A_f v^2} dv \quad (8)$$

The developed simulation model is based on a longitudinal vehicle dynamics approach, in which the vehicle is represented as a rigid single-body system moving along the driving direction. For each simulation scenario, the road gradient is assumed to remain constant, while tire slip, suspension dynamics, and mechanical vibration effects are neglected. These assumptions reduce the computational complexity of the model and are considered appropriate for evaluating vehicle energy flow, energy consumption, powertrain efficiency, regenerative braking performance, and operational emissions under standard driving conditions. Although these simplifications may introduce small deviations during highly dynamic maneuvers or on low-adhesion road surfaces, they have a negligible influence on the objectives of the present study, which focuses on system-level energy and efficiency analysis rather than detailed vehicle handling or ride dynamics.

Table 2. Basic features of the designed vehicle, battery cell and battery pack.

	Specification	Value
Vehicle	Mass	2154 kg
	Frontal area (A_F)	2.17 m ²
	Air drag coefficient (C_d)	0.24
	Effective tire radius (r)	0.335 m
	Fixed reduction gear ratio (i_g)	9.29
	Motor power	160 kW
	Maximum torque (0–4365 rpm)	350 Nm
	Maximum speed	177 km/h
Battery Cell	Manufacturer	Panasonic (Kadoma, Japan)
	Model	NCR18650BD
	Voltage (at SOC = 60%)	3.72 V
	Typical capacity (at 25 °C)	3.18 Ah
	Mass	49.5 g
	Volumetric energy density	630 Wh/L
	Gravimetric energy density	217 Wh/kg
Battery Pack	Number of cells (96 series × 78 parallel)	7488
	Number of modules	4
	Module cell number (24 series × 78 parallel)	1872
	Battery pack voltage (at SOC = 60%)	357 V
	Battery pack capacity	248 Ah
	Battery nominal energy	88.5 kWh
	Battery usable energy	85 kWh
	Max. Continuous Discharge Current	780 A

2.2. Electric Motor Model

The Permanent Magnet Synchronous Motor (PMSM) model used in this study was developed in the MATLAB/Simulink environment based on the PMSM block available in the Simscape Electrical library. The PMSM model represents the electrical and mechanical dynamics of a three-phase PMSM and enables accurate simulation of EV traction systems under different operating conditions. The model includes stator resistance, d-axis and q-axis inductances, permanent magnet flux linkage, rotor inertia, and electromagnetic torque dynamics. The voltages across the stator windings are expressed as Equation (9).

$$\begin{bmatrix} v_a \\ v_b \\ v_c \end{bmatrix} = \begin{bmatrix} R_s & 0 & 0 \\ 0 & R_s & 0 \\ 0 & 0 & R_s \end{bmatrix} \begin{bmatrix} i_a \\ i_b \\ i_c \end{bmatrix} + \begin{bmatrix} \frac{d\psi_a}{dt} \\ \frac{d\psi_b}{dt} \\ \frac{d\psi_c}{dt} \end{bmatrix} \quad (9)$$

where v , R , i and ψ represent the parameters associated with each stator phase winding, including voltage, resistance, current, and magnetic flux, respectively. The permanent magnets and the three-phase stator windings jointly contribute to the total magnetic flux linkage of each phase winding. Therefore, the total flux linkage can be expressed as given in Equation (10).

$$\begin{bmatrix} \psi_a \\ \psi_b \\ \psi_c \end{bmatrix} = \begin{bmatrix} L_{aa} & L_{ab} & L_{ac} \\ L_{ba} & L_{bb} & L_{bc} \\ L_{ca} & L_{cb} & L_{cc} \end{bmatrix} \begin{bmatrix} i_a \\ i_b \\ i_c \end{bmatrix} + \begin{bmatrix} \psi_{am} \\ \psi_{bm} \\ \psi_{cm} \end{bmatrix} \quad (10)$$

where L and ψ are the parameters of each phase of the stator windings; the self-inductances and the mutual inductances and the permanent magnet fluxes linking, respectively. The inductances in the stator windings are functions of the rotor electrical angle and defined by Equations (11)–(16).

$$L_{aa} = L_s + L_m \cos(2\theta_e) \quad (11)$$

$$L_{bb} = L_s + L_m \cos(2(\theta_e - 2\pi/3)) \quad (12)$$

$$L_{cc} = L_s + L_m \cos(2(\theta_e - 4\pi/3)) \quad (13)$$

$$L_{ab} = L_{ba} = -M_s + L_m \cos(2(\theta_e + \pi/6)) \quad (14)$$

$$L_{bc} = L_{cb} = -M_s + L_m \cos(2(\theta_e - \pi/2)) \quad (15)$$

$$L_{ac} = L_{ca} = -M_s + L_m \cos(2(\theta_e - 7\pi/6)) \quad (16)$$

where θ_e is the electrical angle of the rotor and M_s is the mutual inductance of the stator. By applying Park's transformation to the electrical equations of the machine, the torque expression can be transformed into a form that is independent of the rotor position angle. The Park transformation matrix (P) is defined as given in Equation (17).

$$P = 2/3 \begin{bmatrix} \cos \theta_e & \cos(\theta_e - 2\pi/3) & \cos(\theta_e - 4\pi/3) \\ -\sin \theta_e & -\sin(\theta_e - 2\pi/3) & -\sin(\theta_e - 4\pi/3) \\ 0.5 & 0.5 & 0.5 \end{bmatrix} \quad (17)$$

Applying Park's transformation to the stator winding voltages and currents converts the three-phase abc variables into the rotating dq0 reference frame, which is independent of the rotor position angle, as expressed in Equations (18) and (19).

$$\begin{bmatrix} v_d \\ v_q \\ v_0 \end{bmatrix} = P \begin{bmatrix} v_a \\ v_b \\ v_c \end{bmatrix} \quad (18)$$

$$\begin{bmatrix} i_d \\ i_q \\ i_0 \end{bmatrix} = P \begin{bmatrix} i_a \\ i_b \\ i_c \end{bmatrix} \quad (19)$$

Applying Park's transformation to the first two electrical equations results in the dq-axis dynamic equations that describe the operational behavior of the PMSM model, as presented in Equations (20)–(23).

$$v_d = R_s i_d + L_d \frac{di_d}{dt} - N\omega i_q L_q \quad (20)$$

$$v_q = R_s i_q + L_q \frac{di_q}{dt} + N\omega(i_d L_d + \psi_m) \quad (21)$$

$$v_0 = R_s i_0 + L_0 \frac{di_0}{dt} \quad (22)$$

$$\tau_m = \frac{3}{2} N (i_q (i_d L_d + \psi_m) - i_d i_q L_q) \quad (23)$$

where L is the inductance of d -axis, q -axis, and zero sequence, ω is the rotational speed of the rotor, and N is the number of permanent magnet pole pairs.

In this study, the vehicle parameters were determined based on the reference TOGG T10F BEV model, which belongs to the C-segment sedan class. The reference vehicle is equipped with a fully electric propulsion system delivering a maximum motor power of 160 kW and a peak torque of 350 Nm through a single-speed reduction gearbox [40]. A PMSM was selected as the traction motor due to its suitability for EV applications, offering advantages such as high efficiency, high power density, wide constant-power operating capability through flux-weakening operation, and low maintenance requirements resulting from its brushless rotor structure [41]. In addition, the technical characteristics of the reference vehicle, including motor performance and maximum vehicle speed, were considered during the modeling and simulation studies.

The PMSM operates in the constant-torque region below the rated speed and in the constant-power region above the rated speed. During low-speed operation, the motor

speed is controlled by increasing the applied voltage while maintaining a nearly constant magnetic flux. When the rated speed is reached, the motor terminal voltage attains its rated limit. Beyond this operating point, the voltage remains approximately constant, and flux weakening control is applied by reducing the magnetic flux inversely with speed. As a consequence, the motor torque decreases hyperbolically with increasing rotational speed, while the output power remains nearly constant within the high-speed operating region. Figure 1 presents the Torque–Speed and Power–Speed characteristics of the 160 kW PMSM employed in the reference TOGG T10F configured in this study.

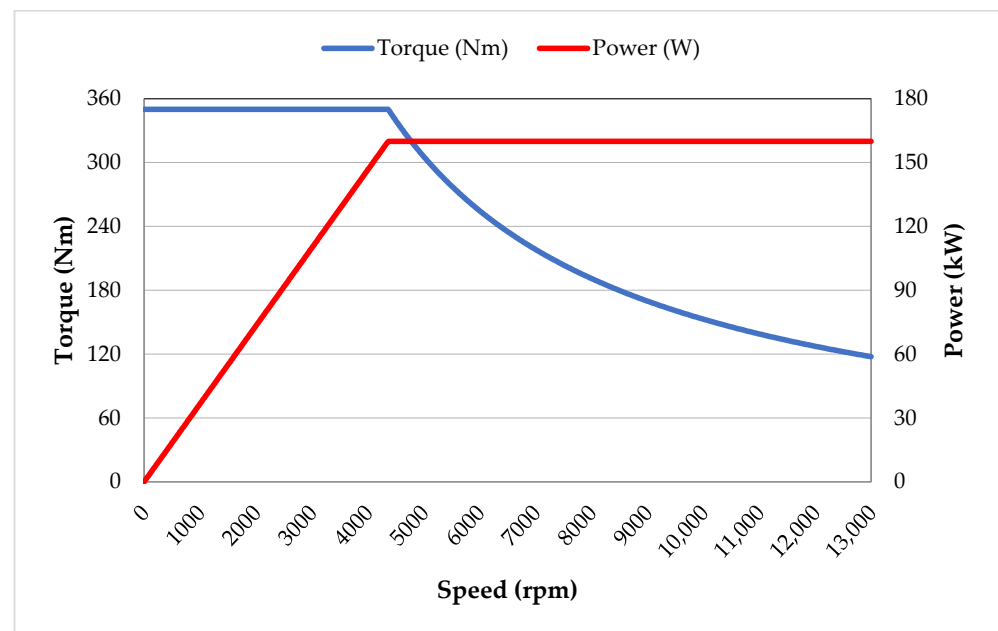


Figure 1. 160 kW PMSM Torque–Speed and Power–Speed characteristics.

As illustrated in the power–speed characteristic, the PMSM is capable of sustaining its maximum power output within the speed range of 4365 rpm to 13,000 rpm. A wide Constant Power Speed Ratio (CPSR) enables the motor to deliver high torque over an extended operating region, thereby enhancing vehicle acceleration and gradeability performance while also reducing the complexity of the transmission system [42]. The configured motor used in this study, the CPSR is calculated as $13,000/4365 = 2.978$. This value is sufficiently high to allow the vehicle to achieve high speeds without requiring a multi-speed transmission system.

In addition, when the maximum motor speed is limited to 13,000 rpm, the maximum vehicle speed (v_{max}) becomes approximately 177 km/h, as determined using Equation (24) [43].

$$v_{max} = \frac{\pi N_m r}{30 i_g} \times \frac{3600 \text{ s}}{1 \text{ h}} \times \frac{1 \text{ km}}{1000 \text{ m}} \quad (24)$$

where N_m is the revolutions per minute of the motor. To express the vehicle speed obtained from the first fractional expression in km/h instead of m/s, an appropriate unit conversion is applied. The variation in motor torque, required traction torque, and vehicle gradeability as a function of vehicle speed are presented in Figure 2.

As illustrated in Figure 2, the vehicle achieves a maximum gradeability of approximately 45% at standstill, and this capability gradually decreases as the vehicle speed increases. Based solely on the intersection of the available propulsion power and the road-load power demand, and disregarding the maximum motor-speed constraint, the vehicle could theoretically exceed 200 km/h on a flat road and attain approximately 191 km/h on

a 6% road gradient. However, these values represent theoretical road-load-based limits and are not achievable within the specified motor operating range. When the maximum motor speed is restricted to 13,000 rpm, the corresponding maximum vehicle speed is approximately 177 km/h. Furthermore, operation above the base motor speed occurs in the flux-weakening region, where the available motor torque progressively decreases with increasing speed. Based on the torque–speed characteristics of the selected traction motor, the calculated 0–100 km/h acceleration time of the vehicle is approximately 7.7 s. Overall, the obtained maximum speed, acceleration, and climbing performance characteristics are considered satisfactory for a C-segment vehicle and are also consistent with the performance characteristics of the reference TOGG T10F [44]. The acceleration value is slightly higher than the officially reported 7.5 s acceleration time of the reference TOGG T10F. The small difference can be attributed to modeling assumptions, including simplified longitudinal vehicle dynamics, constant drivetrain parameters, and the exclusion of certain vehicle control strategies such as launch control, traction control optimization, and torque management algorithms employed in the production vehicle.

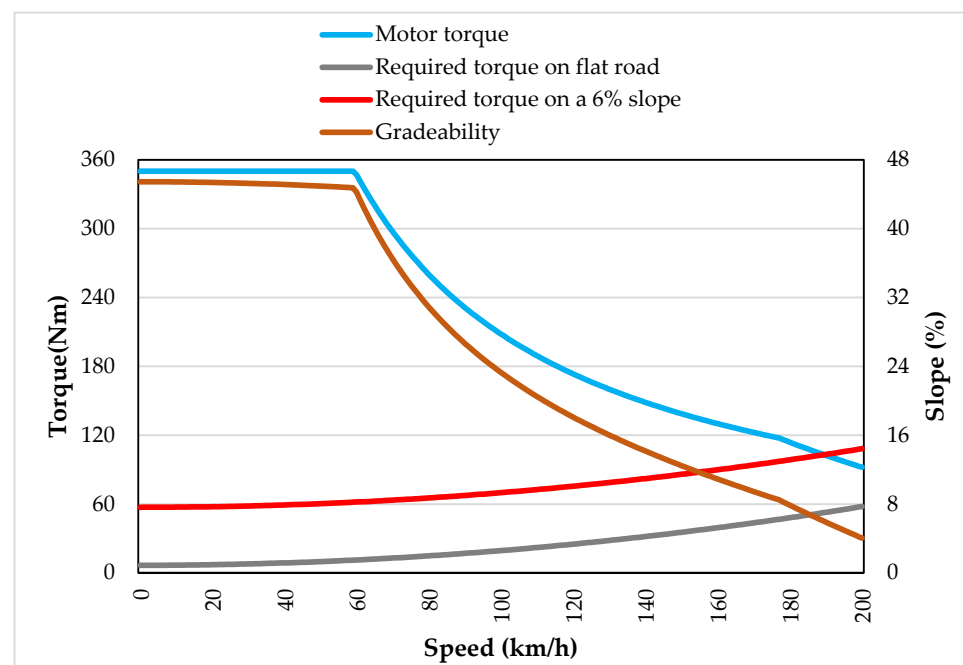


Figure 2. Motor torque, torque requirements and gradeability of the vehicle depending on speed.

The efficiency map of the PMSM traction motor used in this study is illustrated in Figure 3 [45,46]. The motor operates with high efficiency over a wide speed and torque range, with peak efficiency values exceeding 97% in the medium-speed and medium-torque operating region. At very low and very high torque demands, the efficiency decreases due to increased copper and switching losses.

2.3. Battery Model

The battery system considered in this study was modeled using the Datasheet Battery block in the MATLAB/Simulink Powertrain Blockset environment. A lithium-ion battery pack based on the Panasonic NCR18650BD cylindrical cell was selected as the energy storage system of the vehicle. The selected cell has a nominal voltage of 3.72 V at 60% SOC, a typical capacity of 3.18 Ah at 25 °C, and a maximum continuous discharge current capability suitable for high-power EV applications. The cell provides high gravimetric and volumetric energy densities of 217 Wh/kg and 630 Wh/L, respectively, making it appropriate for long-range BEVs. The battery pack configuration consists of 96 cells

connected in series (N_s) and 78 cells connected in parallel (N_p), resulting in a total of 7488 cells. The pack is divided into four modules, each containing 24 series-connected and 78 parallel-connected cells. Based on this configuration, the battery pack voltage reaches approximately 357 V at 60% SOC, while the total pack capacity becomes 248 Ah. The nominal battery energy is 88.5 kWh, with an available usable energy capacity of approximately 85 kWh. The battery pack is capable of supplying a maximum continuous discharge current of 780 A to satisfy the high power demand of the traction system during acceleration and high-speed driving conditions.

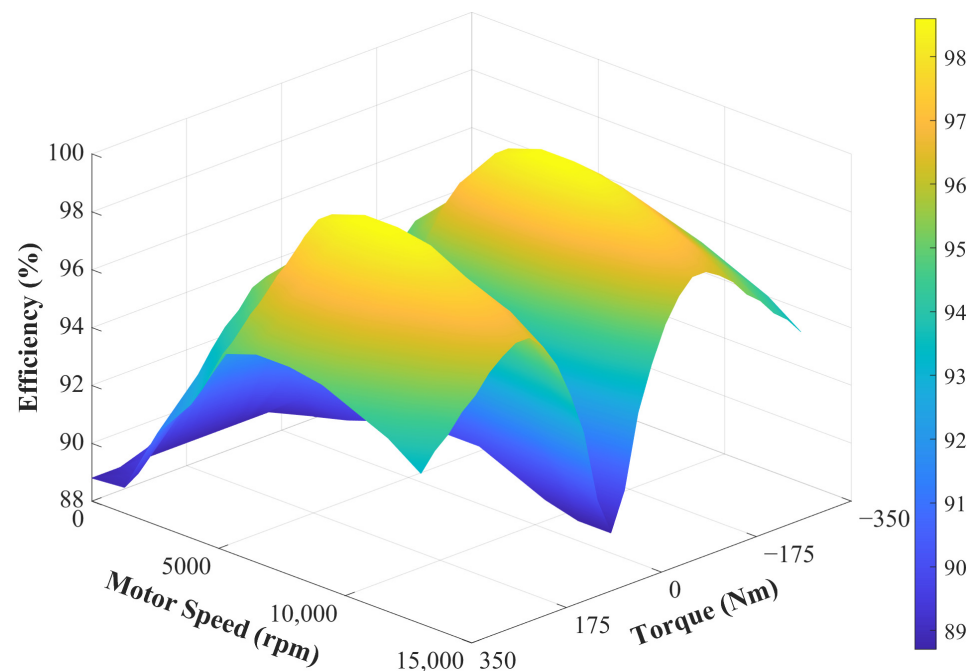


Figure 3. Efficiency map of the PMSM traction motor as a function of motor speed and torque.

In the simulation model, the open-circuit voltage (V_{oc}) and internal resistance (R_{int}) characteristics of the battery were defined as nonlinear functions of SOC and temperature (T) using lookup tables to represent the battery electrical characteristics. The WLTP Class 3 driving cycle was simulated under the standard ambient temperature of 25 °C; therefore, the battery temperature was assumed to remain constant throughout the simulation. Consequently, dynamic thermal behavior, heat generation, cooling conditions, and cell-temperature variations were not considered. Voltage drops caused by internal resistance and power losses under high-current operating conditions were incorporated into the electrical battery model.

$$V_{oc} = f(SOC) \quad (25)$$

$$R_{int} = f(T, SOC) \quad (26)$$

$$V_c = V_{oc} - I_c R_{int} \quad (27)$$

where V_c is the voltage and I_c is the current of the cell. Equations (28)–(30) are employed to calculate the battery pack output voltage (V_B), current (I_B), and power (P_B) characteristics.

$$V_B = V_c N_s \quad (28)$$

$$I_B = I_c N_p \quad (29)$$

$$P_B = V_B I_B \quad (30)$$

The battery power loss (P_L) is calculated using Equation (31), which represents the resistive losses caused by the internal resistance of the battery pack during charge and discharge operation.

$$P_L = N_s N_p I_c^2 R_{int} \quad (31)$$

The SOC was continuously updated through current integration during charge and discharge processes. Since SOC cannot be measured directly, it must be estimated using battery current, voltage, and temperature information together with an appropriate battery model (Figure 4). In this study, a hybrid SOC estimation structure combining Coulomb Counting (CC), Extended Kalman Filter (EKF), and Unscented Kalman Filter (UKF) algorithms was implemented in the MATLAB/Simulink environment in order to improve estimation accuracy under dynamic driving conditions. The CC method estimates SOC by integrating the battery current over time (Equation (32)).

$$SOC = SOC_0 - \frac{1}{C_{bat}} \int I_{bat} dt \quad (32)$$

where I_{bat} is the pack current, SOC_0 is the initial SOC and C_{bat} is the battery capacity. The EKF algorithm estimates SOC by linearizing the nonlinear battery model around the current operating point. The general nonlinear state-space equations used in the estimator are given as Equations (33) and (34).

$$x_k = f(x_{k-1}, u_k) + w_k \quad (33)$$

$$y_k = h(x_k, u_k) + v_k \quad (34)$$

where x_k is the system state vector, u_k is the input vector, y_k denotes the measurement vector, while w_k and v_k represent process and measurement noise, respectively. EKF offers low computational complexity and reliable real-time performance for automotive BMS applications [47]. However, its estimation accuracy may decrease under highly nonlinear operating conditions due to the linearization process based on Jacobian matrices. To overcome these limitations, the UKF algorithm was also implemented. Unlike EKF, UKF does not require linearization of the nonlinear battery model. Instead, it uses sigma-point (X_k) sampling and Unscented Transformation (UT) techniques to propagate nonlinear system states more accurately. UKF-based methods generally provide improved robustness and estimation accuracy under highly dynamic operating conditions [48]. The sigma-point generation process used in the UKF algorithm can be expressed as Equation (35).

$$X_k = \left[x_k, x_k + \sqrt{(n + \lambda)P_k}, x_k - \sqrt{(n + \lambda)P_k} \right] \quad (35)$$

where P_k is the covariance matrix, n represents the number of states, and λ is the scaling parameter of the UT. The propagated sigma points are then used to estimate the predicted system states and output measurements. In the developed BMS structure, CC, EKF, and UKF algorithms were operated simultaneously. The CC method provides rapid SOC tracking based on current integration, whereas EKF and UKF continuously correct estimation drift using voltage and temperature feedback. This hybrid estimation architecture significantly improves SOC estimation stability and robustness under varying load demands, regenerative braking conditions, and dynamic driving cycles [49].

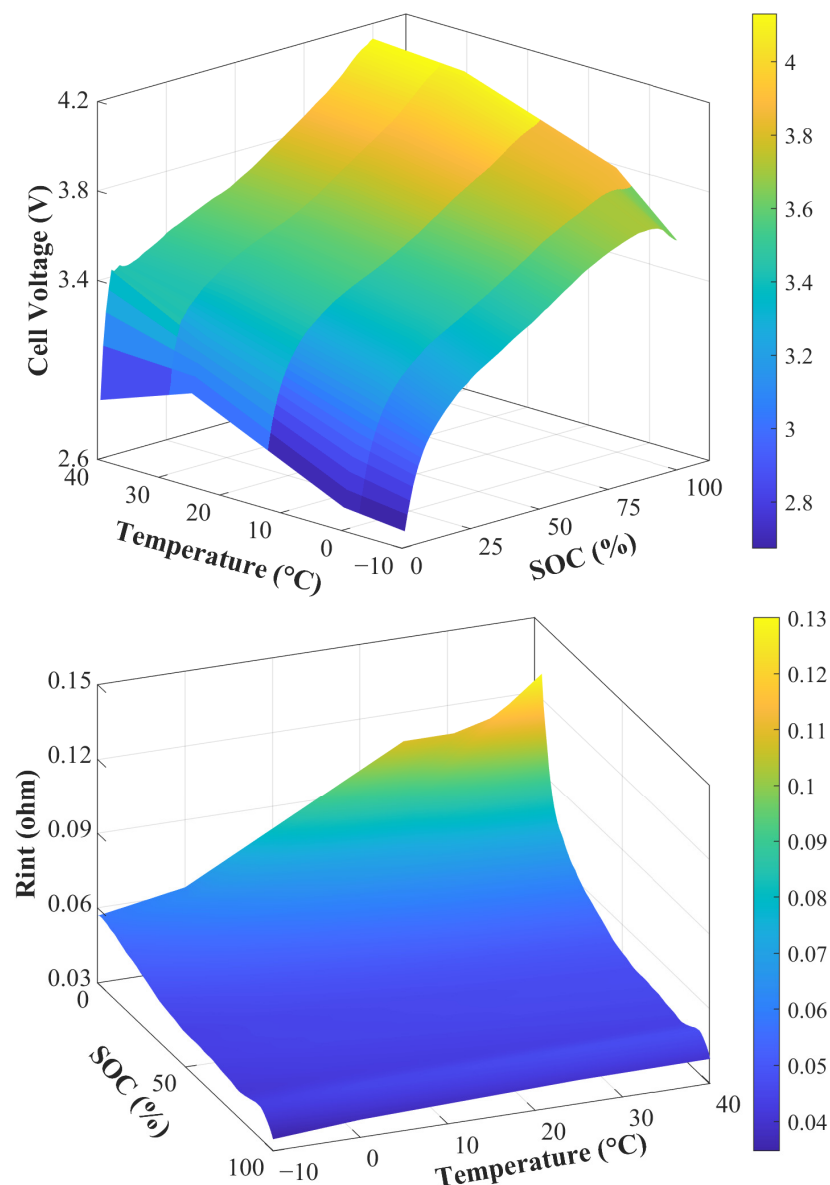


Figure 4. Variation of lithium-ion battery cell open-circuit voltage and internal resistance with respect to SOC and temperature.

Both the EKF and UKF employed an additive process-noise covariance matrix of $Q = 1 \times 10^{-10} \cdot I_2$ and an additive measurement-noise covariance of $R = 0.1$. The filters were initialized with the state vector $x_0 = [\text{SOC}_{\text{init}} + 0.05, 0]^T$ and the initial error covariance matrix $P_0 = \text{diag}(1 \times 10^{-5}, 1)$. Accordingly, the EKF and UKF estimators were intentionally initialized with a 5% SOC offset from the true initial SOC value to evaluate their convergence characteristics, while the polarization-voltage state was initialized to zero. For the UKF, the Unscented Transformation parameters were selected as $\alpha = 1$, $\beta = 2$, and $\kappa = 1$, resulting in the scaling parameter $\lambda = \alpha^2(n + \kappa) - n = 1$, where $n = 2$ denotes the number of state variables.

The Coulomb Counting (CC), EKF, and UKF algorithms operate in parallel within the developed BMS, and their estimated SOC values (SOCCC, SOCEKF, and SOCUKF) are retained independently rather than being combined into a single fused estimate. Furthermore, the voltage-based EKF and UKF estimators are enabled only when the BMS fault-status signal indicates normal operating conditions. This parallel estimation architec-

ture provides redundant and cross-validated SOC information for the vehicle-level control system, thereby improving the reliability of battery state monitoring.

The estimation performance of the proposed architecture was validated through the convergence behavior of the EKF and UKF estimators. Owing to the deliberately introduced 5% initial SOC offset, both filters successfully corrected the initial estimation error and converged to the reference SOC trajectory within approximately 0.3 s from the beginning of the WLTP driving cycle. After convergence, the steady-state estimation error remained below 0.25% SOC throughout the remainder of the driving cycle. These results demonstrate that the proposed parallel SOC estimation framework provides accurate and robust SOC tracking under dynamic operating conditions, including varying load demands and regenerative braking events.

To find fuel economy (FE) in gasoline equivalent (Lge/100 km), Equation (36) is used.

$$FE = \left(\frac{\Delta E_B}{E_{gas}d} \right) \times 100 \quad (36)$$

where E_{gas} is the gasoline energy density in kJ/L, d is the distance traveled, and ΔE_B is the amount of battery energy difference. The expression inside the parentheses in Equation (36) yields a unit of Lge/km. To express the result in the conventional fuel consumption format of Lge/100 km, the obtained value is multiplied by 100. The main specifications of the designed BEV, lithium-ion battery cell, and battery pack configuration used in this study are summarized in Table 2.

2.4. Auxiliary Systems

During the WLTP driving cycle, auxiliary devices that are not required by legislation must be switched off or deactivated during chassis dynamometer operation [50]. Therefore, this study primarily focuses on the traction system and battery energy consumption characteristics. The power consumption of secondary auxiliary devices, such as lighting, control units, sensors, and infotainment systems, was assumed to be constant at 300 W throughout the simulations [51,52]. Under these assumptions, the total energy consumption of the auxiliary devices during the 30 min WLTP driving cycle becomes approximately 150 Wh. Considering that the WLTP cycle distance is 23.3 km, the contribution of auxiliary devices to the vehicle energy consumption corresponds to approximately 6.4 Wh/km.

2.5. Braking Systems

In this study, a series regenerative braking strategy was adopted in order to maximize energy recovery during deceleration events. Under braking conditions, the demanded braking power is primarily supplied by regenerative braking through the PMSM traction motor, enabling the recovery of kinetic energy and recharging of the battery pack. In this study, unlike HEVs, the limiting factor for regenerative braking is not the battery charging capability but the generator-mode operating capability of the PMSM traction motor. Except under very high SOC conditions where the battery charging current must be restricted, the maximum battery charging power remains higher than the regenerative power that can be produced by the PMSM. Therefore, the regenerative braking power is mainly limited by the motor's generator-mode torque and power characteristics rather than the battery charging limit. The total braking power (P_{brk}) is equal to the sum of regenerative braking power ($P_{brk,rgn}$) and mechanical braking power ($P_{brk,mech}$) (Equation (37)).

$$P_{brk} = \tau_{brk}\omega_w = P_{brk,rgn} + P_{brk,mech} \quad (37)$$

where ω_w is the angular speed of the wheel and τ_{brk} is the brake torque. The $P_{brk,mech}$ is calculated using Equations (38) and (39) [53].

$$P_{brk,mech} = \tau_{disc}\omega_w \quad (38)$$

$$P_{brk,mech} = \frac{\mu_k P D_b \pi D_b R_m N}{4} \frac{2\pi N_w}{60} \quad (39)$$

where τ_{disc} is the disc torque, μ_k is the coefficient of kinetic friction, D_b is the brake actuator bore diameter, R_m is the mean radius of brake pad force application on brake rotor, N is the number of brake pads, and N_w is revolutions per minute of the wheel. The τ_m required to achieve $P_{brk,rgn}$ and mechanical power input of PMSM (P_{mech}) is shown by Equation (40).

$$\tau_m = \frac{P_{brk,rgn}\eta_d}{\frac{2\pi N_m}{60}} = \frac{P_{mech}}{\frac{2\pi N_m}{60}} \quad (40)$$

The electrical power generated by the PMSM (P_{elc}) in generator mode is calculated using Equation (41).

$$P_{elc} = P_{mech}\eta_m \leq P_B k_{chrg} \quad (41)$$

where k_{chrg} is the charging factor at any SOC. The k_{chrg} coefficient varies according to the battery SOC level and is defined as 1 up to 85% SOC, 0.7 at 90% SOC, 0.35 at 95% SOC, and 0 at 100% SOC. In this way, the regenerative braking capability is gradually reduced at high SOC levels in order to prevent battery overcharging and protect battery health. Consequently, τ_{disc} that must be supplied by the mechanical braking system is determined using Equation (42).

$$\tau_{disc} = \tau_{brk} - \tau_m i_g \quad (42)$$

where $\tau_m i_g$ denotes the motor braking torque reflected to the wheel through the transmission gear ratio.

3. Simulation

A driving cycle represents the variation in vehicle speed with respect to time and is commonly used to evaluate vehicle performance characteristics such as energy consumption, driving range, and exhaust emissions [54]. Driving cycles play a critical role in vehicle design and performance analysis because they directly influence vehicle energy demand, component sizing, operating efficiency, and overall system cost [55]. In this study, the WLTP driving cycle was selected because it provides more realistic driving conditions compared to previous standard test cycles and better reflects real-world vehicle operation [56]. Since the modeled BEV has a maximum motor power of 160 kW and a total vehicle mass of 2154 kg, the corresponding power-to-mass ratio is calculated as 74.28 W/kg. As this value exceeds 34 W/kg, the WLTP Class 3 driving cycle was adopted for the simulations. The WLTP Class 3 cycle consists of four operating phases, namely low-, medium-, high-, and extra-high-speed sections, each representing different driving environments and traffic conditions. The main characteristics of these stages are summarized in Table 3. The use of the WLTP Class 3 cycle enables a more realistic evaluation of vehicle energy consumption, regenerative braking performance, and battery usage characteristics under dynamic driving conditions.

The battery SOC decreased from an initial value of 61.8% to 58.17% at the end of the 23.3 km WLTP driving cycle (Figure 5). This SOC reduction corresponds to an energy consumption of approximately 3174 Wh. Accordingly, the vehicle energy consumption was calculated as 136.4 Wh/km. Based on the gasoline-equivalent energy conversion, this value corresponds to approximately 1.53 Lge/100 km. Considering the vehicle's usable battery

energy capacity of 85 kWh, the estimated driving range based on the WLTP Class 3 driving cycle is approximately 623 km. The vehicle's total battery capacity of 88.5 kWh corresponds approximately to the energy content of 10 L of gasoline, while the usable battery energy of 85 kWh is equivalent to nearly 9.5 L of gasoline. These values clearly demonstrate one of the major disadvantages of BEVs, namely the significantly lower energy density of batteries compared to conventional liquid fuels. However, despite this limitation, BEVs are able to compensate for a large portion of this disadvantage due to their substantially higher drivetrain efficiency compared to internal combustion engine vehicles.

Table 3. Characteristics of WLTP Class 3.

Characteristic	Unit	Low	Medium	High	Extra-High	Total
Distance	m	3095	4756	7158	8254	23,262
Total time	s	589	433	455	323	1800
Standing time	s	156	48	31	7	242
Average speed (Trip)	km/h	18.9	39.5	56.6	92	46.5
Average driving speed	km/h	25.7	44.5	60.8	94.0	53.8
Maximum speed	km/h	56.5	76.6	97.4	131.3	

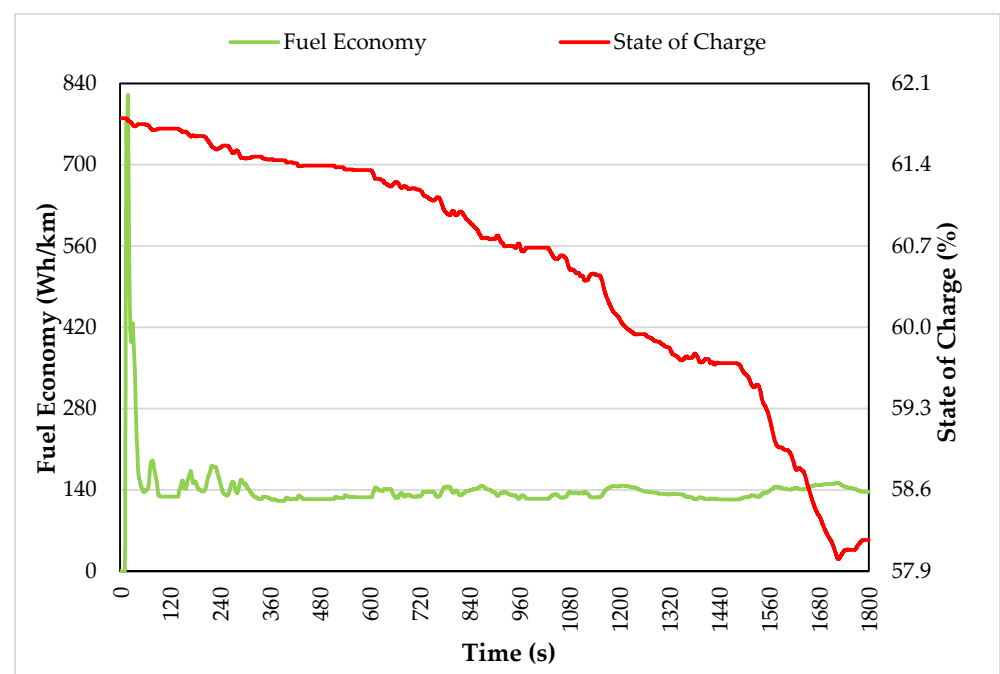


Figure 5. Variation in cumulative vehicle energy consumption and battery SOC throughout the WLTP Class 3 driving cycle.

Figure 6 demonstrates that the actual vehicle speed closely follows the target WLTP speed profile, confirming the effectiveness of the implemented longitudinal vehicle controller. The middle figure illustrates the variation in PMSM rotational speed throughout the cycle, while the lower figure presents the corresponding motor torque demand. Positive torque values represent traction operation during acceleration, whereas negative torque values indicate regenerative braking operation during deceleration periods.

The maximum motor torque demand of 146 Nm occurs at the 1155th second of the driving cycle. This operating point does not correspond solely to the instant of maximum vehicle speed or maximum acceleration. Instead, the highest torque demand occurs at a vehicle speed of 25.1 km/h under an acceleration of approximately 1.53 m/s^2 , which is very close to the maximum acceleration value of 1.58 m/s^2 observed at 14.6 km/h. Considering that the vehicle mass exceeds 2 tons, it is inevitable that acceleration demand becomes a

more dominant factor than vehicle speed in determining instantaneous traction torque requirements. The maximum vehicle speed of 131.3 km/h occurs at the 1724th second of the driving cycle. At this instant, since there is no acceleration demand, the required motor torque remains relatively low at only 24.8 Nm, while the corresponding motor speed reaches 9653 rpm. When the maximum torque demand occurs, the motor output power is calculated as 27.2 kW, whereas at the maximum speed point it is 25.1 kW. Therefore, neither the maximum torque condition nor the maximum speed condition corresponds to the maximum motor power operating point.

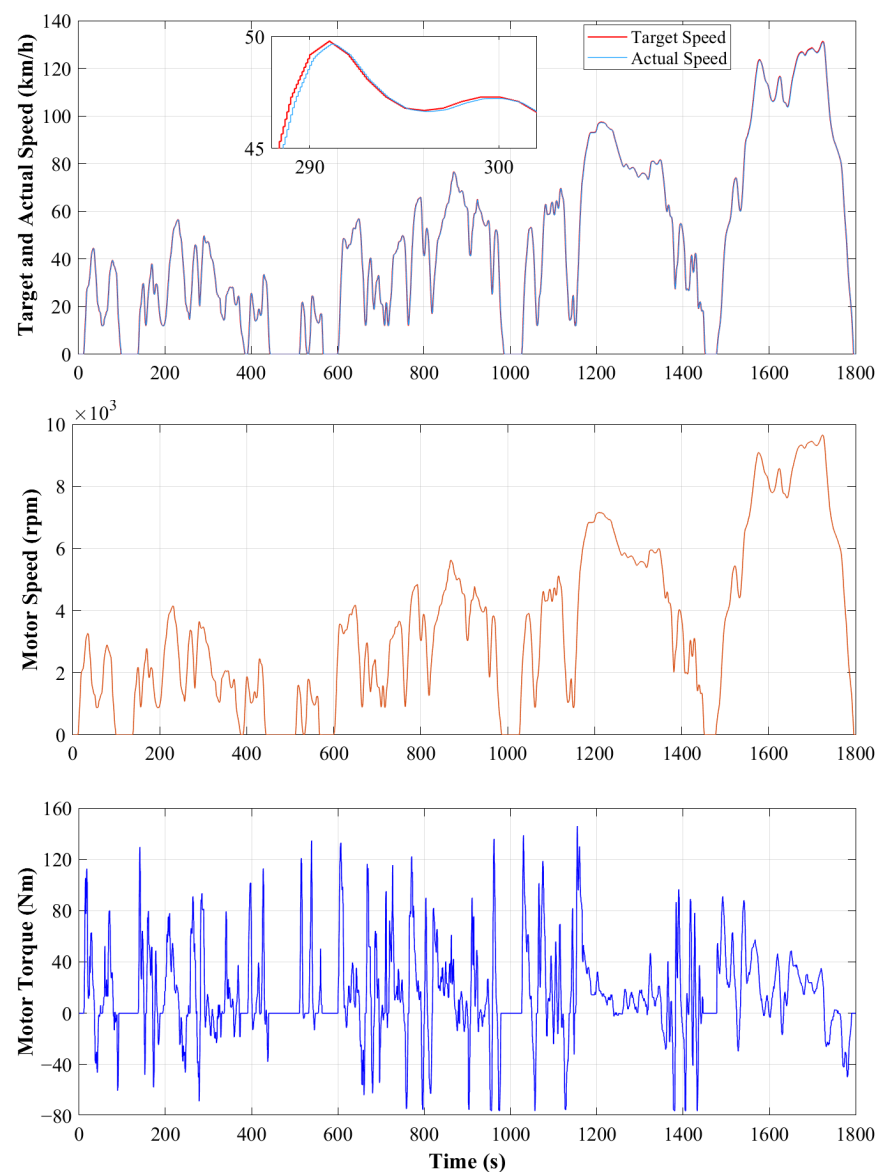


Figure 6. Vehicle speed tracking, motor speed, and motor torque profiles during the WLTP Class 3 driving cycle.

In addition to the WLTP Class 3 driving cycle, the simulations were also performed using the US06 and ArtMot150 driving cycles in order to evaluate the vehicle under different operating conditions. The US06 cycle represents an aggressive driving profile characterized by rapid acceleration and deceleration events, high transient power demand, and elevated driving dynamics.

It was originally developed within the Supplemental Federal Test Procedure to better reflect aggressive real-world driving behavior and includes a maximum vehicle speed

of approximately 129 km/h together with high acceleration rates. Furthermore, the Art-Mot150 cycle was employed to analyze vehicle performance under sustained high-speed driving conditions typical of intercity and highway transportation. This cycle includes long-duration high-speed operation, relatively fewer stop-and-go events, and vehicle speeds reaching up to 150 km/h. Therefore, it is particularly suitable for evaluating high-speed energy consumption, battery discharge behavior, motor efficiency, and thermal loading conditions in BEVs.

4. Results and Discussion

Figure 7 illustrates the mechanical output power of the PMSM traction motor, while the middle figure presents the corresponding electrical input/output power of the motor drive system. The lower figure shows the battery power variation throughout the driving cycle. Positive power values indicate traction operation, whereas negative values correspond to regenerative braking conditions. It can be observed that battery power demand is generally higher than PMSM electrical power during traction due to inverter, drivetrain, and auxiliary system losses. Similarly, during regenerative braking events, only a portion of the recovered electrical energy can be transferred back to the battery because of conversion and battery charging losses.

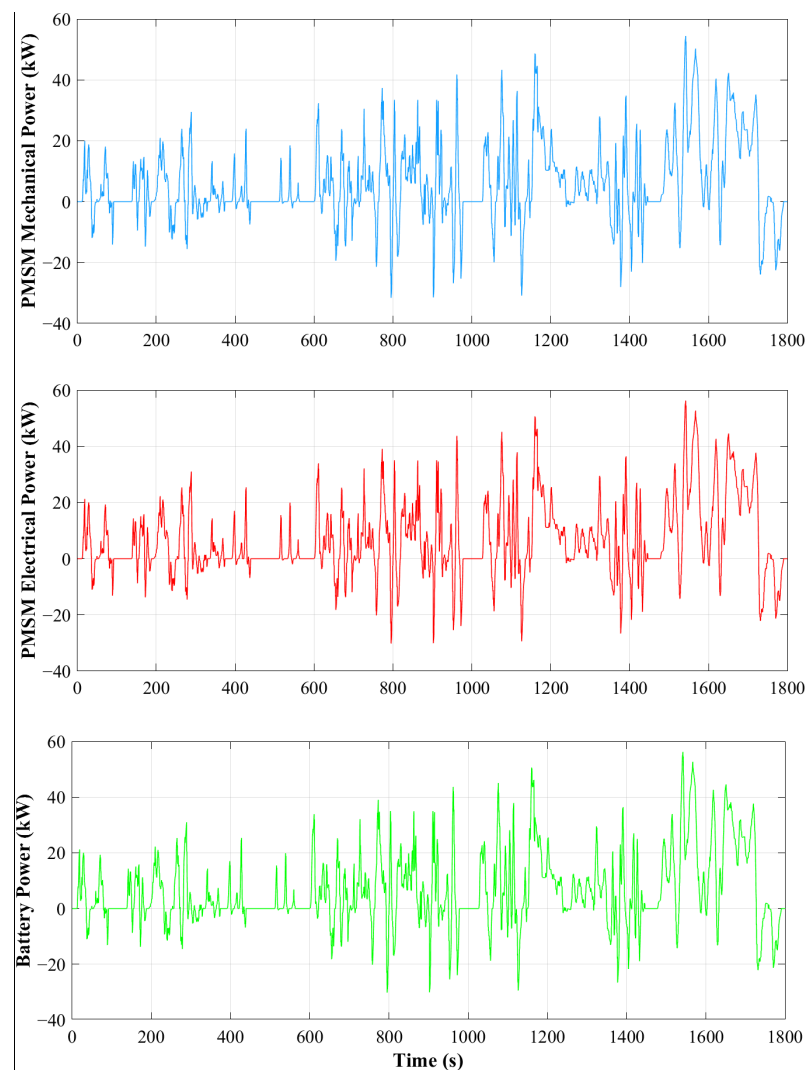


Figure 7. PMSM mechanical power, PMSM electrical power, and battery power profiles obtained during the WLTP Class 3 driving cycle.

At the 1542nd second of the WLTP Class 3 driving cycle, the vehicle reached its maximum traction power demand. Under this operating condition, the PMSM delivered an output mechanical power of 54.4 kW while drawing 56.3 kW electrical power, corresponding to an efficiency of approximately 96.7%. At the same instant, the total battery power demand reached 57.8 kW, indicating that nearly 1.5 kW of additional power was consumed by battery and converter losses and auxiliary systems. On the other hand, the maximum regenerative braking event occurred at the 795th second of the cycle. During this operating point, the PMSM operated in generator mode with an input mechanical power of 31.6 kW. With a generation efficiency of approximately 95.6%, nearly 30.2 kW of electrical power was generated by the PMSM. Of this recovered energy, approximately 29.8 kW was successfully transferred back to the battery pack.

To evaluate the robustness of the developed vehicle model and investigate the influence of key vehicle parameters on energy consumption, the simulations were repeated under different driving cycles representing varying acceleration characteristics and maximum cruising speeds, namely WLTP Class 3, US06, and ArtMot150. In addition, a sensitivity analysis was performed by increasing the vehicle mass by 10% (Case 1) and the aerodynamic parameters ($C_D \times A_f$) by 10% (Case 2), since both parameters directly affect traction power demand and vehicle energy consumption. The resulting energy consumption values and their corresponding variations obtained from the simulations are summarized in Table 4.

Table 4. The vehicle energy consumption values under different driving cycles.

Driving Cycle	FE Wh/km	% ΔQ_{cons}	FE Case 1	% ΔQ_{cons}	FE Case 2	% ΔQ_{cons}
WLTP	136.4	0.0	144.6	6.0	141.1	3.4
US06	170.1	24.7	179.2	5.3	177.1	4.1
Artemis150	177.3	29.9	184.2	3.9	187.2	5.6

The average vehicle speeds of the WLTP Class 3, US06, and ArtMot150 driving cycles are approximately 53.8 km/h, 79.6 km/h, and 99.6 km/h, respectively, indicating that the ArtMot150 cycle represents the most demanding high-speed driving condition among the evaluated test cycles [57]. The vehicle energy consumption increases with the aggressiveness and average speed of the driving cycle. The lowest energy consumption value was obtained in the WLTP cycle with 136.4 Wh/km, whereas the highest consumption was observed in the ArtMot150 cycle with 177.3 Wh/km. While the estimated WLTP driving range was calculated as approximately 623 km, this value decreased to nearly 500 km under the US06 cycle and further dropped to approximately 479 km under the ArtMot150 cycle. Compared to the WLTP cycle, the energy consumption increased by approximately 24.7% in the US06 cycle and 29.9% in the ArtMot150 cycle. Although the ArtMot150 cycle has a significantly higher average speed than the US06 cycle, the relatively small difference in energy consumption between these two cycles indicates that average speed alone is not the only factor affecting vehicle energy demand. The US06 cycle includes more aggressive acceleration and deceleration events representative of real-world dynamic driving conditions. These rapid transient operating conditions increase traction power demand and reduce the effectiveness of regenerative braking due to conversion losses, thereby contributing to higher energy consumption. The sensitivity analysis further demonstrates that the influence of vehicle parameters on energy consumption strongly depends on driving characteristics. In the WLTP cycle, a 10% increase in vehicle mass (Case 1) caused the highest increase in energy consumption, approximately 6.0%, whereas the impact of aerodynamic changes (Case 2) remained limited to 3.4%. In contrast, under the high-speed ArtMot150 cycle, aerodynamic characteristics became more dominant, leading to a 5.6% increase in energy consumption, while the effect of mass increase decreased to 3.9%. For the US06 cycle, the

effects of mass and aerodynamic changes became relatively close, with increases of 5.3% and 4.1%, respectively. These results indicate that vehicle mass has a more significant effect on energy consumption under relatively low- and medium-speed driving conditions characterized by frequent acceleration events, whereas aerodynamic properties become increasingly critical during sustained high-speed driving operation.

The distribution of motor operating points in terms of motor speed and torque under different driving cycles is illustrated in Figure 8. In the WLTP cycle, most operating points are concentrated in the medium-speed and medium-torque region where the PMSM operates with high efficiency approximately 93%. The US06 cycle exhibits a wider operating range with more frequent transient torque demands due to aggressive acceleration and deceleration events.

In the Artemis Motorway 150 cycle, the operating points shift toward higher motor speed regions because of sustained high-speed driving conditions. The results demonstrate that the PMSM predominantly operates within high-efficiency regions for all evaluated driving cycles, contributing significantly to the overall energy efficiency of the BEV. The energy flow characteristics of the proposed BEV were analyzed separately for both traction and regenerative braking operating modes, and the obtained results are summarized in Tables 5 and 6. The analysis provides detailed insight into the energy conversion processes occurring between the battery, PMSM, driveline, and wheels during vehicle operation. During driving mode, the battery supplied a total energy of 14,449 kJ to the propulsion system. In the first stage, a portion of this energy was consumed by auxiliary systems and power electronic conversion losses, resulting in 14,295 kJ electrical energy delivered to the PMSM input. Therefore, the efficiency of the battery-to-motor electrical transfer stage was calculated as 98.9%. In the second stage, the PMSM converted the supplied electrical energy into 13,353 kJ mechanical output energy with an efficiency of 93.4%. Finally, the mechanical energy was transmitted through the driveline to the wheels, and 11,782 kJ useful traction energy reached the wheels with a driveline efficiency of 88.2%. Consequently, the cumulative efficiency from the battery output to the wheel traction energy during driving operation was calculated as 81.5%. These results indicate that the PMSM traction motor operates with very high efficiency under traction conditions. In addition, although the total battery energy is limited compared to conventional liquid fuels, the high cumulative drivetrain efficiency enables BEVs to utilize the available stored energy much more effectively than ICEVs.

Table 5. The energy flow and efficiencies in traction mode (excluding the 12 V auxiliary electrical system).

	Battery	IPMSM Input	IPMSM Output	Wheel
Energy (kJ)	14,449	14,295	13,353	11,782
Efficiency (%)	100	98.9	93.4	88.2
Cum. Eff. (%)	100	98.9	92.4	81.5

Table 6. The energy flow and efficiencies in traction mode (including the 12 V auxiliary electrical system).

	Wheel	IPMSM Input	IPMSM Output	Battery
Energy (kJ)	4347.3	3119.8	2893.6	2876.6
Efficiency (%)	100	71.8	92.7	99.4
Cum. Eff. (%)	100	71.8	66.6	66.2

The regenerative braking energy flow was also analyzed in detail. During braking operation, a total of 4347.3 kJ kinetic energy was available at the wheels. However, not all of this energy could be recovered through regenerative braking, since a portion of the braking energy was consumed in overcoming rolling resistance, aerodynamic drag, and other road load forces acting on the vehicle during deceleration. In addition, another

portion of the energy was dissipated as heat through the conventional friction braking system. Consequently, only 3119.8 kJ mechanical energy could be transferred to the PMSM through the driveline during regenerative braking operation. Therefore, the efficiency of the wheel-to-motor energy transfer stage was calculated as 71.8%. The PMSM operated in generator mode and converted the incoming mechanical energy into 2893.6 kJ electrical energy with a generator efficiency of 92.7%. In the final stage, 2876.6 kJ of this regenerated electrical energy was transferred back to the battery pack with a battery charging efficiency of 99.4%. As a result, the cumulative regenerative braking efficiency from the wheels to the battery was found to be 66.2%.

The results clearly demonstrate that the PMSM exhibits similarly high efficiencies in both motor and generator operating modes. While the PMSM efficiency was calculated as 93.4% during traction operation, it remained at 92.7% during regenerative braking operation. Although a portion of the braking energy is inevitably lost through friction braking, electrical losses, and drivetrain losses, a substantial amount of kinetic energy can still be recovered and reused by the propulsion system. Compared to HEVs, regenerative braking capability in this study is mainly limited by the generator-mode operating capability of the PMSM rather than the battery charging limit. Therefore, the regenerative braking system can effectively recover high amounts of braking energy during deceleration events. The recovered electrical energy is then stored in the battery and reused during subsequent traction operation, thereby directly improving both vehicle energy efficiency and driving range. Also, the results indicate that the energy recovered through regenerative braking becomes increasingly important under aggressive driving conditions characterized by frequent acceleration and deceleration events, such as the US06 driving cycle. In such operating conditions, regenerative braking significantly reduces net battery energy consumption by converting a large portion of the vehicle's kinetic energy back into usable electrical energy. According to the obtained results, the amount of energy recovered through regenerative braking corresponds to approximately 19.9% of the total energy supplied by the battery throughout the driving cycle. Moreover, when both traction-mode and regenerative-mode efficiencies are considered together, the overall round-trip energy conversion efficiency from the battery to the wheels and subsequently back to the battery through the entire propulsion system was calculated as approximately 54%. It should be noted that the reported regenerative energy recovery ratio and wheel-to-battery efficiency were obtained from the developed model under the assumptions adopted in this study. Accordingly, these values should be interpreted as simulation-based performance indicators rather than absolute real-world performance metrics. In practical applications, regenerative braking performance may be influenced by additional factors, including battery aging, thermal effects, sensor uncertainties, controller implementation, drivetrain losses, tire-road interaction, and varying environmental and operating conditions. Therefore, the reported values provide a reference for comparative simulation analysis within the adopted modeling framework.

The designed vehicle model in this study was developed by taking the TOGG T10F V1 RWD Long Range version as the reference vehicle. Accordingly, the simulated vehicle was configured with a 160 kW PMSM traction motor, 350 Nm maximum torque, 88.5 kWh battery capacity, 177 km/h maximum speed, and a total vehicle mass of 2154 kg, all adopted from the TOGG T10F V1 RWD Long Range specifications. Table 7 compares the simulation results obtained in this study with the published specifications of the TOGG T10F Long Range RWD model. As can be observed, the developed MATLAB/Simscape-based vehicle model shows very close agreement with the official TOGG T10F performance data in terms of WLTP driving range and energy consumption characteristics. Both the simulated vehicle and the TOGG T10F provide an estimated WLTP range of 623 km. The official rated energy consumption of the TOGG T10F is reported as approximately 171 Wh/km,

while the corresponding value obtained in this study is 159.1 Wh/km. It should be noted that the rated consumption values published by manufacturers generally include charging losses. In this study, the rated energy consumption value was calculated by considering the average charging efficiency of 85.7% reported in the literature for the ten best-selling EVs in Türkiye. Therefore, charging losses were also included in the rated consumption calculations in order to obtain results comparable with officially published manufacturer data [58]. In contrast, the “Vehicle Consumption” values represent the net battery energy consumed directly by the vehicle propulsion system and onboard auxiliary systems. Therefore, the simulated vehicle consumption value of 136.4 Wh/km is highly consistent with the published TOGG T10F vehicle consumption value of approximately 136 Wh/km. Under high-speed highway driving conditions, the developed model yielded an energy consumption of 177.3 Wh/km, which is also very close to the approximately 185 Wh/km highway consumption value reported for the TOGG T10F [59]. These results confirm that the developed vehicle model can realistically represent the energy consumption behavior and range performance of a modern long-range C-segment BEV under both standardized and high-speed driving conditions.

Table 7. Comparison of energy consumption and driving range results between the designed vehicle model and TOGG T10F.

	WLTP Range (km)	Rated Cons. (Wh/km)	Vehicle Cons. (Wh/km)	Highway Cons. (Wh/km)
This study	623	159.1	136.4	177.3
TOGG T10F	623	171	136	185

The increasing transition toward BEVs is primarily driven by their potential to provide more environmentally friendly and sustainable transportation compared to conventional ICEVs. Therefore, evaluating these vehicles in terms of greenhouse gas emissions is of great importance. Although BEVs are generally described as zero-emission vehicles, this characteristic is valid only during vehicle operation, since no direct tailpipe emissions are produced while driving. However, the environmental impact of BEVs also depends significantly on the primary energy sources used in electricity generation for battery charging. Consequently, the carbon intensity of the electrical grid supplying the charging energy must be considered when evaluating the overall emissions of BEVs. In this study, vehicle emissions were calculated based on the electricity consumption values obtained from the simulations. Different electricity generation scenarios for the year 2024 were considered, including the electricity grid mixes of Türkiye, the European Union (EU), the United States (USA), China, India, and the global average electricity generation mix [60]. The resulting emission values are presented in Table 8. It should be noted that the present analysis estimates only the operational indirect CO₂ emissions associated with electricity generation during vehicle operation. Emissions related to battery manufacturing, vehicle production, charging infrastructure, maintenance, and end-of-life processes are outside the scope of this study. Therefore, the presented results should not be interpreted as a full LCA, but rather as an operational emissions assessment based on different electricity generation mixes. CO₂e (carbon dioxide equivalent) is used throughout this study because the adopted grid emission factors are reported in terms of CO₂-equivalent emissions. CO₂e represents the combined global warming impact of all greenhouse gases expressed as an equivalent amount of CO₂ based on their respective global warming potentials, whereas CO₂ refers only to carbon dioxide emissions.

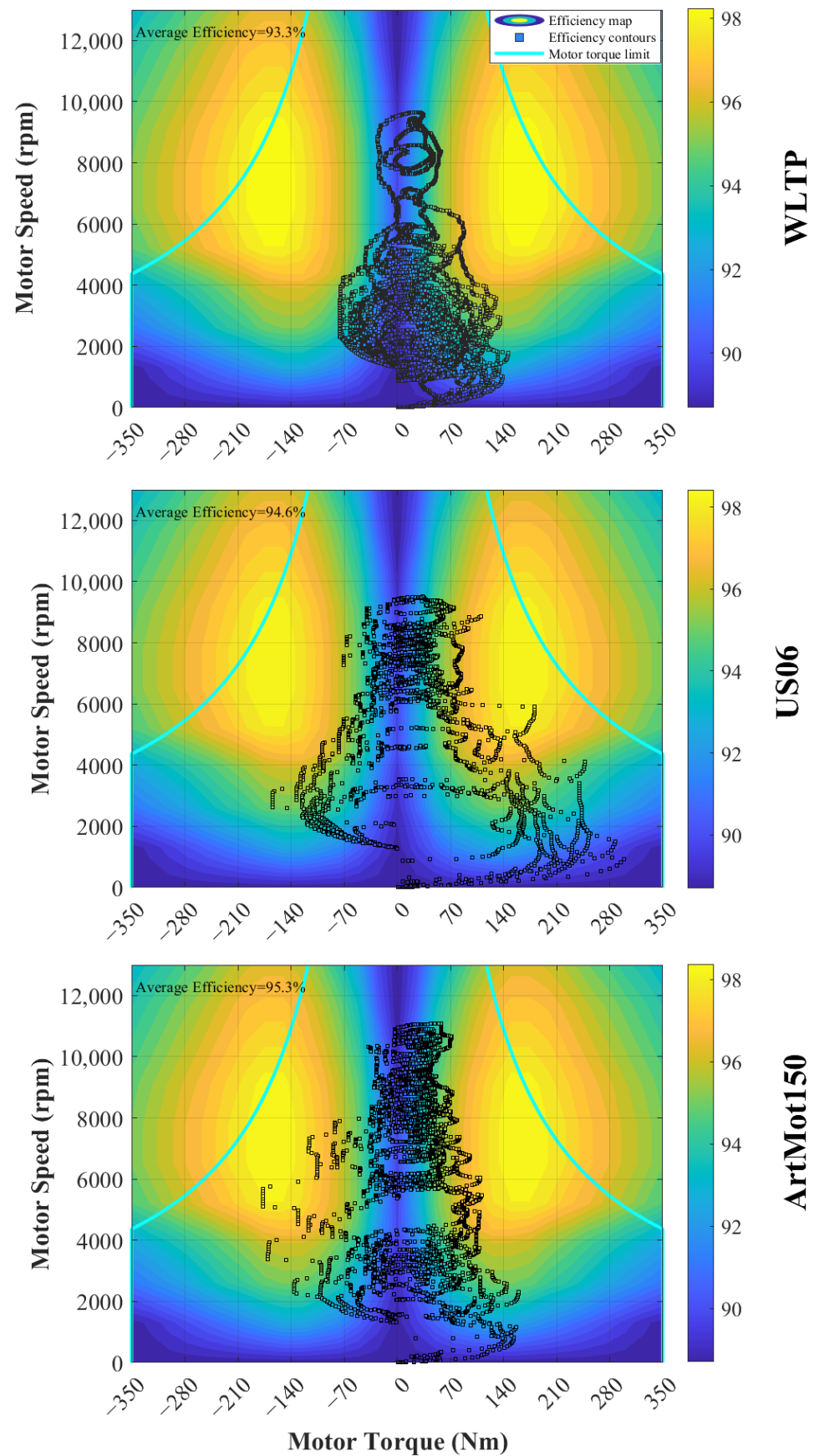


Figure 8. Distribution of PMSM operating points over the efficiency map under WLTP Class 3, US06, and ArtMot150 driving cycles.

Table 8. Comparative operational CO₂e emission analysis of the designed BEV for different electricity mixes based on the rated energy consumption.

	Emission Intensity (gCO ₂ e/kWh) (kgCO ₂ e/kWh)	BEV Emission (gCO ₂ e/km)
EU	174	27.7
USA	321	51.1
Türkiye	405	64.4
China	563	89.6
India	729	116.0
World	445	70.8

The vehicle emission values were calculated by multiplying the simulated vehicle rated consumption with the corresponding grid emission intensity values. The emission intensity data were obtained from the IEA Electricity 2025 report. As shown in the table, the environmental performance of the BEV strongly depends on the carbon intensity of the electricity used for charging. Among the evaluated scenarios, the European Union electricity mix provides the lowest emission value with 27.7 gCO₂e/km due to its relatively high share of renewable and low-carbon electricity generation sources. In contrast, India exhibits the highest BEV emission value of 116.0 gCO₂e/km because of its heavily fossil-fuel-dependent electricity generation structure. The calculated emission value for Türkiye is 64.4 gCO₂e/km, which remains significantly lower than conventional ICEVs despite the relatively carbon-intensive electricity mix. Similarly, the global average electricity mix results in an emission level of 70.8 gCO₂e/km. China and the USA produce intermediate results with 89.6 gCO₂e/km and 51.1 gCO₂e/km, respectively. These findings clearly demonstrate that BEVs can substantially reduce transportation-related greenhouse gas emissions even when charged using relatively carbon-intensive electricity grids. Furthermore, the results indicate that increasing the share of renewable energy sources in electricity generation can significantly improve the environmental benefits of BEVs.

A typical passenger vehicle had average operational CO₂ emissions of approximately 107.9 gCO₂/km in the European Union in 2024, whereas the corresponding value in the United States was approximately 248.6 gCO₂/km in 2023 [61,62]. Compared with these values, the BEV analyzed in this study demonstrates significantly lower greenhouse gas emissions under all evaluated electricity generation scenarios, particularly when charged using low-carbon electricity mixes.

5. Conclusions

This study presented the design, modeling, and performance analysis of a BEV based on a MATLAB/Simscape environment using the TOGG T10F Long Range RWD as the reference vehicle. The developed model included longitudinal vehicle dynamics, a PMSM traction motor, lithium-ion battery pack, regenerative braking system, and drivetrain efficiency analysis under different driving conditions. The simulation results demonstrated that the proposed BEV configuration provides both high energy efficiency and long driving range performance comparable to modern commercial EVs. According to the WLTP Class 3 driving cycle, the vehicle energy consumption was calculated as 136.4 Wh/km, corresponding to approximately 1.53 Lge/100 km gasoline-equivalent consumption. Using the 85 kWh usable battery capacity, the estimated WLTP driving range was found to be approximately 623 km, which is highly consistent with the official TOGG T10F specifications. The influence of different driving characteristics on vehicle performance was also investigated using WLTP Class 3, US06, and Artemis Motorway 150 driving cycles. The results indicated that aggressive acceleration behavior and sustained high-speed driving significantly increase vehicle energy consumption and reduce driving range. While the WLTP-based range was approximately 623 km, this value decreased to nearly 500 km under the US06 cycle and approximately 479 km under the ArtMot150 cycle. Furthermore, sensitivity analyses

revealed that vehicle mass has a stronger influence on energy consumption under low- and medium-speed driving conditions, whereas aerodynamic characteristics become increasingly dominant at higher cruising speeds. During traction operation, the PMSM efficiency was calculated as 93.4%, while the cumulative drivetrain efficiency from the battery to the wheels reached 81.5%. In regenerative braking mode, the PMSM generator efficiency remained at 92.7%, and the cumulative wheel-to-battery regenerative efficiency was found to be 66.2%. The environmental impact of the vehicle was also evaluated under different electricity generation scenarios for 2024. Depending on the electricity grid emission intensity, the calculated BEV emissions varied between 27.7 gCO₂e/km and 116.0 gCO₂e/km. The results clearly demonstrated that the environmental benefits of BEVs strongly depend on the carbon intensity of electricity generation and can be further improved with increasing renewable energy penetration in electrical grids. This study has several limitations that should be acknowledged. First, the developed MATLAB/Simscape model has not yet been experimentally validated and was parameterized using publicly available technical specifications of the reference TOGG T10F vehicle. In addition, simplified auxiliary power loads and battery thermal behavior were adopted to maintain computational efficiency, while battery degradation mechanisms, including capacity fade and internal resistance growth, were not considered. Furthermore, the emissions analysis was limited to the estimation of operational indirect CO₂ emissions associated with electricity generation and does not represent a comprehensive life cycle assessment, as emissions related to battery manufacturing, vehicle production, charging infrastructure, maintenance, and end-of-life processes were excluded.

Future work may focus on:

- Advanced optimization and validation of battery and energy management systems: Improving BMS algorithms, regenerative braking control strategies, and power distribution methods to further enhance vehicle efficiency, battery lifetime, and driving range under different operating conditions. Future studies will also include experimental validation of the proposed simulation model.
- Integration of next-generation BEV technologies and advanced battery modeling: Investigating the effects of advanced battery chemistries, high-efficiency PMSM designs, wide-bandgap power electronics, and lightweight vehicle materials on the overall performance, efficiency, and sustainability of BEVs. Future work will also incorporate detailed battery thermal behavior and battery degradation mechanisms, such as capacity fade and internal resistance growth, to improve model fidelity.
- Evaluation under real-world operating conditions and comprehensive environmental assessment: Extending the study using additional driving cycles, different traffic conditions, road slopes, environmental temperatures, and real-world driving data to evaluate the robustness, adaptability, and energy performance of the proposed BEV system. In addition, future research will extend the environmental assessment by performing a comprehensive LCA that includes battery manufacturing, vehicle production, charging infrastructure, maintenance, and end-of-life processes.

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Abbreviations

The following abbreviations are used in this manuscript:

ArtMot150	Artemis Motorway 150 Driving Cycle
BEV	Battery Electric Vehicle
BMS	Battery Management System
B-ECDC	Bengaluru Electric Car Driving Cycle
CC	Coulomb Counting
CLTC	China Light-Duty Vehicle Test Cycle
CO ₂	Carbon Dioxide
CSPR	Constant Power Speed Ratio
ECE-R15	Economic Commission for Europe Regulation 15
EMS	Energy Management System
EV	Electric Vehicle
EKF	Extended Kalman Filter
EU	European Union
EUDC	Extra-Urban Driving Cycle
FE	Fuel Economy
FCEV	Fuel Cell Electric Vehicle
FTP	Federal Test Procedure
H-ECDC	Hyderabad Electric Car Driving Cycle
HEV	Hybrid Electric Vehicle
HWFET	Highway Fuel Economy Test
HWY	Highway Cycle
ICEV	Internal Combustion Engine Vehicle
IDC	Indian Driving Cycle
IPMSM	Interior Permanent Magnet Synchronous Motor
JC08	Japan Cycle 08
MIDC	Modified Indian Driving Cycle
NYCC	New York City Cycle
PBEV	Pure Battery Electric Vehicle
PEV	Pure Electric Vehicle
PHEV	Plug-in Hybrid Electric Vehicle
PMSM	Permanent Magnet Synchronous Motor
NEDC	New European Driving Cycle
SOC	State of Charge
UDDS	Urban Dynamometer Driving Schedule
UKF	Unscented Kalman Filter
US06	United States 06 Supplemental Federal Test Procedure
USA	United States of America
WLTC	Worldwide Harmonized Light Vehicles Test Cycle
WLTP	Worldwide Harmonized Light Vehicle Test Procedure
Units and Symbols	
a	Acceleration.
A	Ampere
A_F	Frontal area
Ah	Ampere hour
C_d	Air drag coefficient

C_{rr}	Rolling resistance coefficient
d	Distance
D_b	Brake actuator bore diameter
d_p	Performance factor
E_{gas}	Energy density of gasoline in kJ/L
F	Faraday constant in coulombs per mole
F_{tf}	Total tractive force on wheels
$F_t(t)$	Total traction forces
$F_r(t)$	Total resistive forces
gCO_2	Gram carbon dioxide
gCO_2e/km	Equivalent gram of carbon dioxide per kilometer
i_g	Reduction gear ratio
k_{chrg}	Charging factor
km/h	Kilometer per hour
kW	Kilo watt
kWh	Kilo watt hour
M	Mass with rotational inertia
n	Number of transferred electrons
N	Number of brake pads
Nm	Newton meter
N_m	Rotations per minute of the motor
N_w	Rotate per minute of the wheel
P	Pressure
P_b	Battery power
P_{brk}	Brake power
$P_{brk,mech}$	Mechanical brake power
$P_{brk,rgn}$	Regenerative brake power
P_{elc}	Electrical power output of PMSM
P_{mech}	Mechanical power input of PMSM
P_m	Motor power
p_s	Partial pressure at the surface
r	Tire radius
R_m	Mean radius of brake pad force
t	Time
T	Temperature
v	Velocity
v_{max}	Maximum speed
V	Volt
V_c	Cell voltage
V_{oc}	Open-circuit voltage
W	Weight
Wh	Watt hour
α	Slope angle
ΔE_B	Energy difference in the battery
η_d	Drivetrain efficiency
η_m	Motor efficiency
μ_k	Coefficient of kinetic friction
ρ	Air density
τ_{brk}	Brake torque
τ_{disc}	Disk torque
τ_m	Motor Torque
ω_w	Angular speed of the wheel

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