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Full electric vehicle with multi-speed transmission: a review on dynamic and energetic performances improvements

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Abstract: Multi-speed transmissions (MSTs) can improve the dynamic and energetic performance of battery electric vehicles (BEVs), but their benefits depend on transmission architecture, gear ratios, shift strategy and losses.This paper presents a comprehensive literature review and quantitative database of published BEV MST studies, comparing dynamic and energetic performance with single-speed transmissions (SSTs). The results show that MSTs do not inherently provide simultaneous improvements.Walker et al. reported 19.4% faster 0–60 km/h acceleration, 47% higher top speed and 80% higher gradeability, but an 8–10% driving-range penalty. Conversely, Ruan et al. achieved 13.7% faster 0–60 km/h acceleration, 61.6% higher top speed, 25% higher gradeability and 30.9% lower energy consumption using a CVT. Tian et al. reported a 29.4% reduction in energy consumption with a 4-speed transmission and moderate dynamic improvements. The results highlight the importance of transmission efficiency, gear-ratio selection and shift strategy, rather than the number of ratios alone. A dynamic–energetic trade-off framework is proposed to support the multi-objective selection and optimization of MST configurations for BEVs.
Keywords: Battery electric vehicle; multi-speed transmission; dynamic performance; energetic performance; dynamic–energetic trade-off.

1. INTRODUCTION

The electrification of road transportation has become one of the principal technological approaches for reducing greenhouse gas emissions and decreasing dependence on fossil fuels. Despite continuous improvements in vehicle efficiency and the increasing adoption of alternative powertrain technologies, the transport sector remains one of the largest contributors to global greenhouse gas emissions, with road transport accounting for approximately 71.7% of total transport-related emissions and passenger cars representing nearly 60.6% of this contribution [2,3]. Within the EU, road transport represented approximatively 73% of all total GHG emissions in 2023, highlighting its dominant contribution to the sector's envirnomental impact. In the same year and in comparison with 2022, within the EU, a decrease of 1.6% in average CO_2 emissions was observed for the newly registered PC which has its explanation in the BEV registrations share, with an aproximative relative increase of +2% between these two years [4].

Consequently, increasingly stringent environmental regulations, such as the European Union's Fit for 55 package, have accelerated the transition towards zero-emission mobility by establishing ambitious targets for CO_2 emission reductions from new passenger vehicles and promoting the widespread deployment of battery electric vehicles (BEVs) [5].

Although vehicle electrification is expected to substantialy reduce the road transport emissions, projection published by European Environment Agency (EEA) indicate that additional improvements will be necessary. Figure 1 presents both historical evolution of GHG emissions and future projections by the transport mode, that are based on two scenarios: With Existing Measures (WEM) which assumes that the currently implemented regulations will remain in force, and With Additional Measures (WAM), which additionally considers planned decarbonization policies.

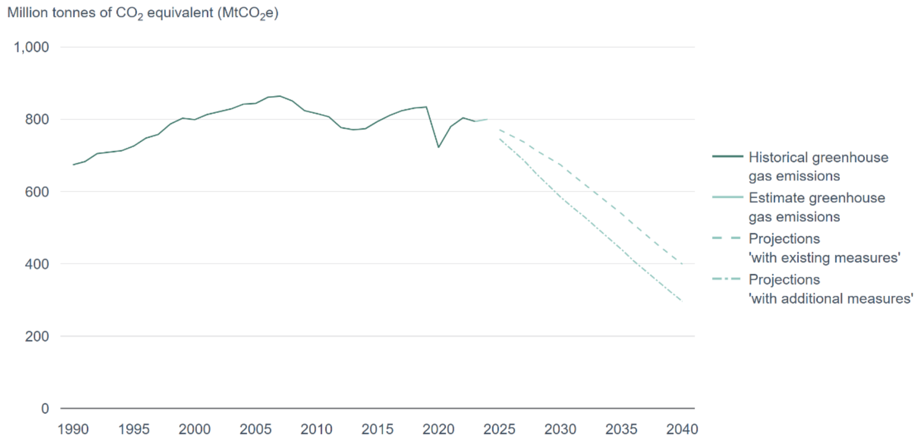


Figure 1. GHG emission of transport sector, historical evolution and future projections under WEM and WAM policy scenarios [4]

Among the available electrified powertrain technologies, battery electric vehicles (BEVs) have emerged as the most promising long-term solution for achieving sustainable road transportation. Their market penetration has increased significantly during recent years and is expected to continue growing rapidly, with projections indicating that BEVs could account for 67.9% of new passenger vehicle registrations by 2030, compared with only 9.8% in 2022 [6]. This rapid market expansion is already visible in the global automotive industry, as shown in figure 2. Europe electric vehicle sales surpassed 18 million units in 2025, representing an annual increase of approximately 20% compared with 2024 and accounting for nearly 25% of total passenger vehicle sales. Since 2021, annual EV sales have consistently increased by approximately 3.5 million units, resulting in electric vehicle representing nearly 5% of the global passenger car fleet [14, 15].

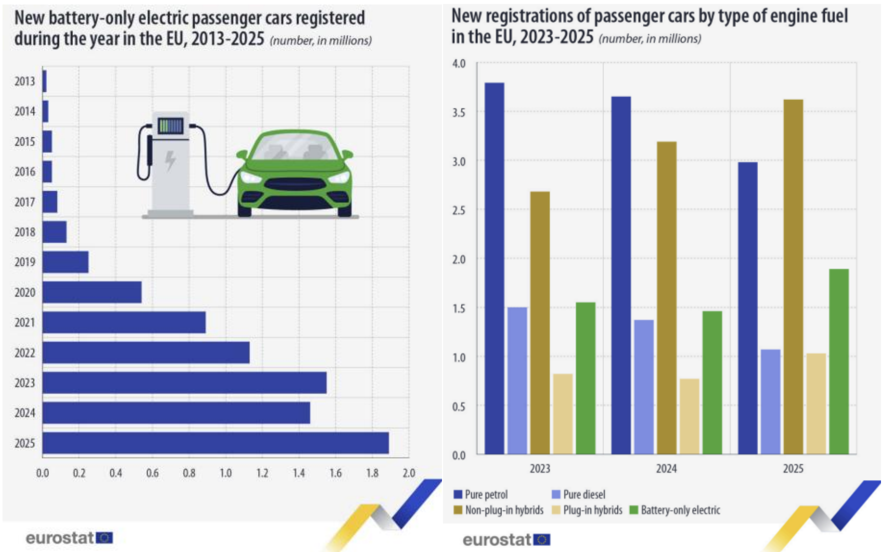


Figure 2. BEV new registrations number of passanger cars in EU [15]

For these reasons, research attention was shifted from the simple replacement of the ICE towards the improvements of the electric powertrain efficiency. Although modern electric machines can achieve peak efficiencies exceeding 90%, operating at peak efficiency only within a limited torque-speed regions [12]. Because most production BEVs employ a fixed single-speed transmission [9, 10], the electric machine is frequently forced to operate outside its most efficient regions under different driving scenarios, therefore resulting in an increased energy consumption and reduced driving range [12, 13, 19]. Consequently, the e-machine calibration settings must be capable of covering the widest possible range of driving scenarios, togheter with a gear ratio optimisation but with the trade-off compromising dynamic performances, as in the case of the paper that developed an eAxle Chevrolet Bolt EV PWT [11].

The performance of a battery electric vehicle is determined by the combined operation of its main drivetrain components, including the battery pack, power electronics, electric machine and mechanical transmission [1]. As the battery pack remains the largest, the heaviest, one of the most expensive components of a BEV, and with a big environmental impact due to the nature of materials used and the manufacturing process [7], improving drivetrain efficiency without increasing battery capacity has become an important engineering objective, as higher efficiency directly contributes to extended driving range, reduced energy consumption and the potential for battery or e-machines downsizing. As illustrated in figure 3 [14], the driving range of most available BEVs on the market already considerably exceeds the average daily driving distance, suggesting that future advances in electric powertrains should primarily focus on utilizing the available battery energy more efficiently rather than continuously increasing battery capacity.

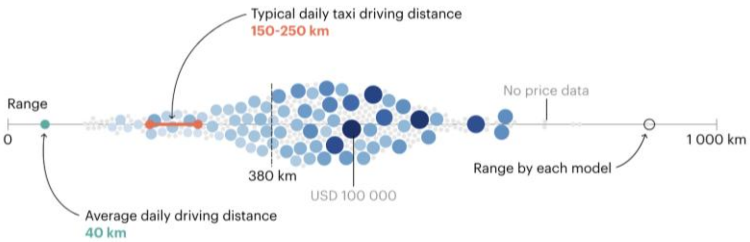


Figure 3. Distribution of the range of BEV models, 2025 [14]

At the same time, modern electric vehicles are expected to deliver rapid acceleration, high maximum speed, satisfactory gradeability and smooth driving characteristics while maintaining competitive manufacturing costs and high system reliability [8, 12, 19].

To address these limitations, increasing research attention has been devoted to the integration of multi-speed transmissions into battery electric vehicle powertrains [12, 19]. By providing multiple transmission ratios, the electric machine can operate closer to its optimal efficiency region over a broader range of vehicle speeds while simultaneously increasing wheel torque during vehicle launch and reducing motor rotational speed during high-speed operation [12, 13, 16, 17, 18, 19, 20].

Numerous studies have demonstrated that this approach has the potential to improve both dynamic and energetic vehicle performance through enhanced acceleration capability [12, 16, 17, 19] increased maximum speed and gradeability [12, 17, 19], lower energy consumption, higher drivetrain efficiency, improved regenerative braking effectiveness and extended driving range [12, 13, 16, 17, 18, 19, 20]. Furthermore, several investigations have shown that the efficiency gains achieved by multi-speed transmissions may enable a reduction in battery capacity while maintaining the same vehicle range, thereby contributing to lower vehicle mass and overall cost [16].

Although numerous studies have investigated different transmission architectures, gear ratio optimization methods and shift control strategies for battery electric vehicles, the reported improvements in dynamic and energetic performance vary considerably due to differences in vehicle category, drivetrain configuration, optimization methodology and driving cycle [12, 19]. Consequently, a quantitative comparison of the available literature is required to objectively assess the real benefits of multi-speed transmissions and identify the conditions under which they provide the greatest performance improvements.

Therefore, this paper presents a literature review on the influence of multi-speed transmissions on the dynamic and energetic performance of battery electric vehicles. Particular emphasis is placed on the quantitative comparison of the improvements reported in the literature regarding vehicle acceleration, maximum speed, gradeability, energy consumption, drivetrain efficiency, battery state of charge and driving range, while also analysing the influence of transmission topology, transmission efficiency, gear ratio optimization and shift strategies on the overall performance of the electric powertrain. More publications are systematically analysed through a unified database containing vehicle characteristics, electric machine specifications, transmission architectures, efficiency data, gear ratios, optimization methodologies, driving cycles and experimentally or numerically reported performance indicators. This systematic approach enables the identification of quantitative trends, performance limits and design trade-offs associated with different transmission architectures, providing objective guidance for future electric powertrain development. Particular attention is devoted to the compromise between dynamic and energetic performance, recognizing that transmission configurations maximizing vehicle dynamics do not necessarily maximize energy efficiency, and vice versa.

2. POTENTIAL BENEFITS OF MST ON BEV

The electric machine (EM) used in BEVs is capable of delivering high torque from very low rotational speeds. Its operating range can generally be divided into a constant-torque region, extending up to the base speed, and a constant-power or field-weakening region, in which torque decreases as rotational speed increases [23, 24, 25]. In PMSM-based traction systems, which are widely used in BEVs due to their high efficiency and controllability, the operating point is determined by the required vehicle torque and speed and is therefore distributed over a broad torque–speed range [23, 24].

However, the efficiency of the EM is not constant throughout this operating range. Efficiency maps and iso-efficiency contours show that the highest efficiency is generally concentrated within a limited region of torque and speed. As illustrated in figure 4 [26], for both Kia Soul and BMW i3, the efficiency maps demonstrate that high-efficiency operation is concentrated within a limited region of torque & speed operating envelope. For Kia Soul, efficiencies above 90% are mainly observed at moderate-to-high torque and speeds from approximately 2500 rpm upwards, whereas the BMW i3 exhibits a comparable high-efficiency region extending from approximately 3500-4000 rpm over a broad range of higher motor speeds. In both cases, the exact efficiency level remain strongly dependent on the torque operating point.

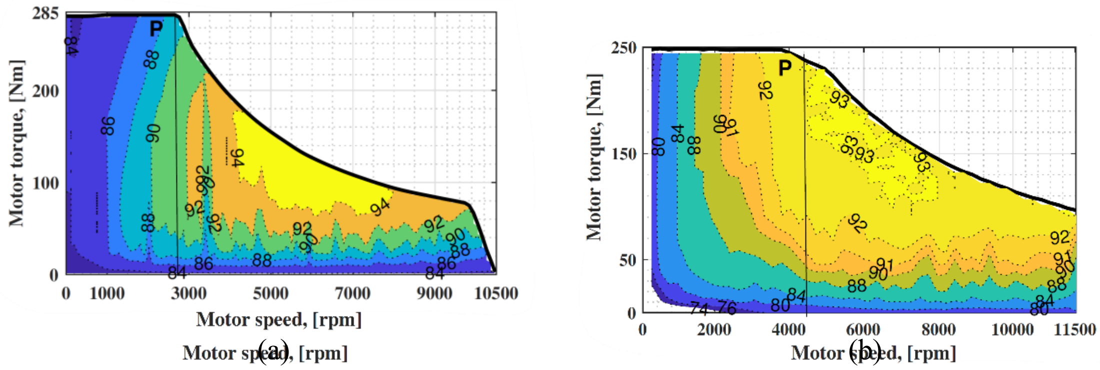
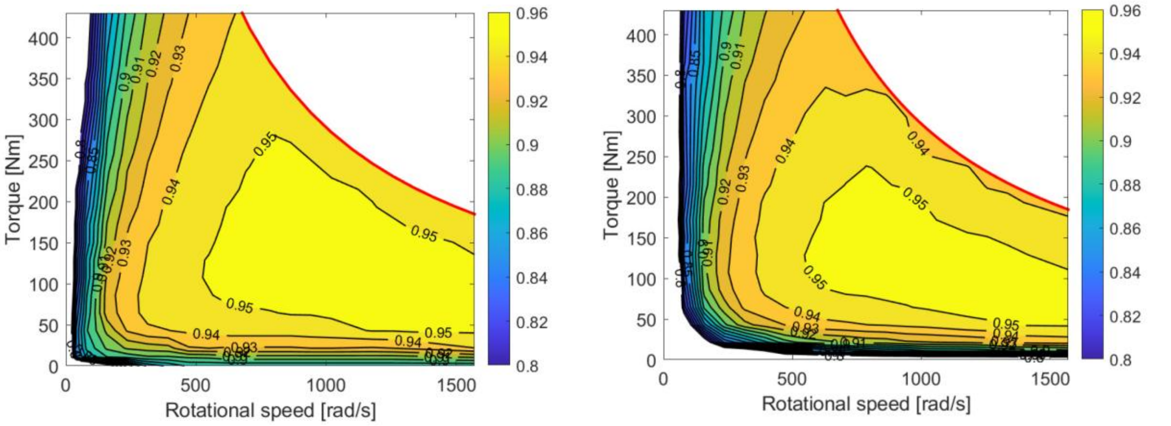


Figure 4. EM characteristic in motor mode: (a) Kia Soul ; (b) BMW i3 [26]

Also, by adding the inverter system power losses in order to obtain an electric drive efficiency map, the efficiency regions within the iso-efficiency curves are more limited, as showed in figure 5 [16]. Consequently, although the wide speed range of modern electric machines allows BEVs to operate with a single-speed transmission, the fixed gear ratio can force the EM to operate away from its high-efficiency region under certain driving conditions [12]. This represents one of the main limitations of the single-speed architecture from an energetic perspective.



(a) **(b)**

Figure 5. Efficiency maps of an Electric Machine (a) and Electric Drive (including inverter losses) (b) [16]

Since BEVs must accommodate highly variable torque and speed demands associated with urban, highway and hilly driving conditions, the motor can consequently operate with reduced efficiency over part of the driving range request [12]. A multi-speed transmission (MST) introduces additional degrees of freedom by adapting the relationship between vehicle speed and traction motor rotational speed. Therefore, lower transmission ratios can be used to reduce EM speed at higher vehicle velocities, such highway driving conditions, whereas higher ratios can increase the wheel torque available during vehicle launch and high-load operation. More importantly, from an energetic perfomance perspective, appropriate gear ratios selection can shift the EM operating points towards regions with higher efficiency over a wider range of driving conditions [12, 13, 16, 17, 18, 19, 20].

This concept can be illustrated by superposing the required EM operating points of a vehicle running under a certain mission profile on the EM torque-speed and efficiency inside map [13, 16, 17], as shown in figure 6 [16]. An ideal operating trajectory, corresponding to the locus of maximum efficiency found for the required isopower curves or power levels, can be considered as a reference for evaluating how effectively a transmission keeps the EM within its high-efficiency regions. The deviation between the actual operating points and this ideal trajectory provides a useful qualitative explanation of the potential energetic benefit associated with additional transmission ratios.

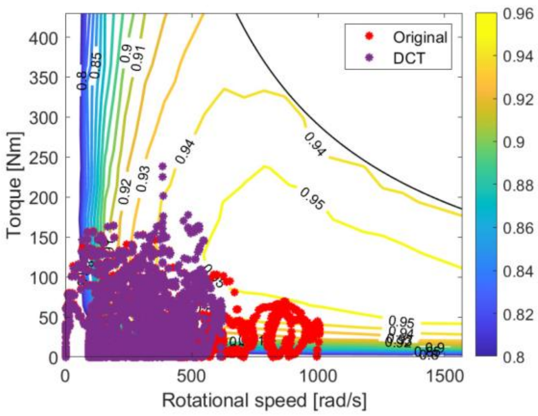


Figure 6. Operating points of a BEV with a SST (Original) vs 2-speed DCT on WLTC cycle [16]

Nevertheless, the implementation of MST also introduces additional mechanical complexity, transmission losses, control requirements, more mass and manufacturing costs [16, 19]. The overall benefits therefore depend not only on the transmission architecture itself but also on the optimization of gear ratios, shift strategies, electric machine operating characteristics and driving conditions [11, 12, 21, 22]. Since these parameters are strongly coupled, improvements in energetic performance may be accompanied by compromises in dynamic behaviour, making the design of multi-speed electric drivetrains a complex multi-objective optimization problem [19].

Moreover, the improvement in electric-machine efficiency does not directly translate into an overall drivetrain efficiency gain. The additional transmission elements introduce more load-dependent and speed-dependent losses such as gear-mesh, bearing, lubrication losses and shifting-related losses and torque delivery interuption, which must be considered togheter with the efficiency improvement achieved by forcing the electric machine operating points toward more efficiency-favourable regions. Therefore, the net energetic benefit of an MST is determined by the balance between the gain in EM efficiency and the additional losses generated by the newly added transmission. This trade-off makes transmission efficiency and power losses fundamental aspects in assesing the actualy benefits of multi-speed transmission in battery electric vehicles. Accordingly, the following sections examines the main sources of transmission losses and their influence on overall drivetrain efficiency for different type of integrated transmissions topologies.

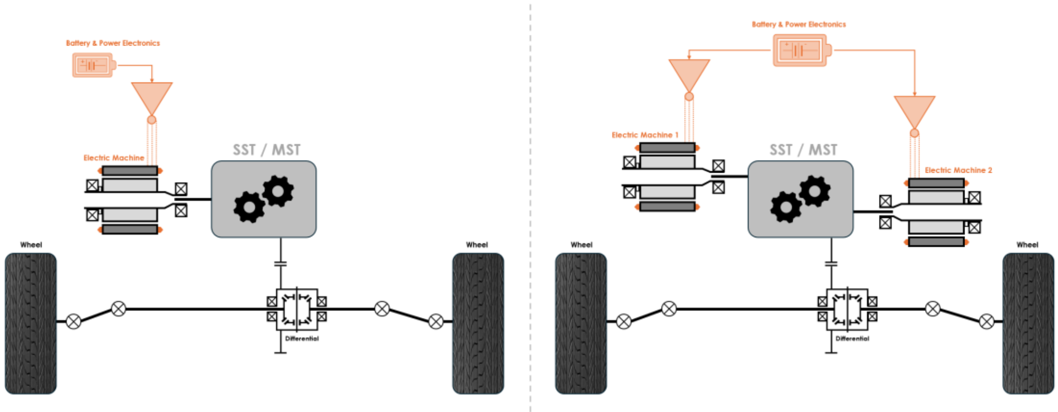
3. E-POWERTRAIN ARCHITECTURE AND MULTI-SPEED TRANSSMISIONS

In BEVs the transmission is positioned between the traction motor and the driven wheels, enabling torque multiplication and adapting the motor output to the vehichle operating required conditions. Based on the arrangement of the traction motor, electric powertrains can be generally classified into two categories [12]:

- centralized-drive (figure 7);
- distributed-drive configurations;

In a distributed-drive architecture, individual electric machines are directly associated with the vehicle wheels, with a reduction gear that is potentially incorporated between the traction motor and the wheel [12].

Within the scope of this review, the centralized-drive configuration is considered as the primary architecture. This layout is conceptually similar to that of the conventional powertrain of ICEVs, with the internal combution engine replaced by an electric machine and the fuel tank recplaed by a battery pack. Depending on the drivetrain configuration, one (figure 7 a) or two traction motors (figure 7 b) may be employed. As showed in figure 7, the transmission, togheter with the differential or final-drive unit, is located between the electric machine(s) and the driven wheels [12]. This transmission could be either a single speed transmission, which is widedly spread today on BEVs on market, or a multi-speed transmission which is less comon on commercial EVs.



(a) **(b)**

Figure 7. E-powertrain architecture: Centralized-drive configuration with one (a) or two (b) traction motors

Transmission systems can be classified according to the method used to vary the transmission ratio and by the gear shifting mechanisms that are used. From the ratio variation perspective, transmission can be divided into *discrete ratio transmissions*, in which the ratio changes between predefined values, and *continuously variable transmissions*, which provide a continuous variation between minimum and maximum ratios without the interuption of torque transfer. Based on the gear shifting mechanism, discrete transmissions can be manually operated or automated through electromechanical or electrohydraulic actuators. In automotive applications, the main architectures relevant to electrified powertrains include single speed transmissions (SST), manual transmissions (MT), automated manual transmissions (AMT), dual-clutch transmissions (DCT), planetary automatic transmissions (PAT) and continuosly variable transmissions (CVT) [19].

Single speed transmission (SST) provides simplicity, low mass and low mechanical losses, but it doesn't offer an on-board ratio adaptation, therefore making the electric machine responsible for covering the entire vehicle operating range. The architecture of a BEV's powertrain with this transmission and the transmission internal architecture are showed in figure 8 (a).

Manual transmissions (MT) require direct driver intervention for clutch operation and gear selection. Although these transmissions offer relatively simple construction, low cost and good reliability, their manual operation makes them poorly suited for BEVs, where automatic ratio selection is preferable for maintaining appropiate motor operating conditions [19, 27]. In figure 8 (b) an MT clasical and simplified topology is presented within an e-powertrain.

Automated manual transmissions (AMT) retain the basic mechanical architecture of an MT while replacing driver's manual gear selection with automated electromechanical or electrohydraulic actuation controled by predefined calibration of the TCU. Their relatively simple construction, competitive efficiency and lower cost make them a potential solution for BEVs. Furthermore, the conventional clutch required in ICEV applications may be eliminated thanks to the electric machine that can provide a precise torque and/or speed control during gear shifting [19, 28]. A simplified topology of this transmission is presented in figure 8 (c).

Dual-clutch transmissions (DCT) employ two clutches and two input shafts, with the gears divided between the two shafts. This arrangement allows the next ratio to be preselected and enables rapid gear changes with minimal torque delivery interruptions. Their efficiency depends strongly on clutch design: wet clutches introduce additional oil-related losses, whereas dry clutches may present greater cooling challenges. DCTs therefore provide smooth operation and efficiecnt shifting, but at the expense of increased complexity, cost and maintenance requirements compared with simpler architectures [19, 29]. A simplified topology of this transmission is presented in figure 8 (d).

Continuosly variable transmissions (CVT) use variable-diameter conical pulleys connected by a belt or chain to provide a continuous ratio variation. This allows the EM a to operate closer to its high-efficiency region over a wider range of vehicle operating conditions. In conventional CVT applications, a torque converter is used to provide torque multiplication and facilitate vehicle launch and idle. However, due to the capability of electrice machines to deliver high torque from standstill, togheter with the absence of engine-idling requirements, the torque converter can eliminated in BEV applications [12, 13], thereby reducing mechanical losses and system complexity. Nevertheless, the efficiency of a belt-driven CVT is affected by frictional power losses, pulley clamping forces, belt deformation and potential slip, particularly when operating at small effective pulley diameters. Consequently, the usable ratio range of the pulley-belt mechanism is constrained by efficiency and durability considerations. A reduction gear can therefore be integrated downstream of the CVT to extend the overall transmission ratio range without requiring excessively small pulley diameters, allowing the CVT to operate within a more favourable efficiency range. Furthermore, replacing conventional hydraulic actuation with electromechanical actuation may reduce auxiliary losses and improve the overall efficiency of the CVT-based electric drivetrain [12, 13, 19]. A simplified topology of this transmission is presented in figure 8 (e).

Other transmission concepts have also been investigated for electrified powertrains, although they are less frequently considered in BEV studies. Inverse Automated Manual Transmissions (I-AMT) represent a modified AMT architecture in which a clutch replaces one of the synchronizing mechanisms, reducing torque interruption during gear changes and potentially enabling smoother shifting through advanced control strategies. Infinitely Variable Transmissions (IVT) combine a CVT with a planetary gear arrangement to achieve an extremely wide ratio range, including a theoretical zero-speed ratio. However, because electric machines can deliver full torque from standstill, their advantages over conventional CVTs in BEV applications remain limited. Magnetic Gear Transmissions (MGT) transfer torque through magnetic coupling rather than mechanical tooth engagement, eliminating physical contact, lubrication requirements and wear. Despite these advantages, their relatively low efficiency and high permanent-magnet content currently limit their practical applicability in electric vehicles. Planetary Automatic Transmissions (PAT) employ planetary gearsets and automated shifting mechanisms to provide multiple discrete gear ratios and represent an alternative automatic transmission architecture for electrified powertrains [19].

The selection of the transmission topology determines not only the mechanical layout and operating flexibility of the BEV powertrain, but also the associated energy losses and transmission efficiency. Therefore, beyond the qualitative comparison of the different architectures, their energetic impact must be assessed by considering the mechanisms responsible for power losses within the transmission. The following subsection focuses on transmission efficiency and the main sources of power losses associated with the different transmission concepts used in BEV applications and the ways of improving it and making it more suitable for these applications.

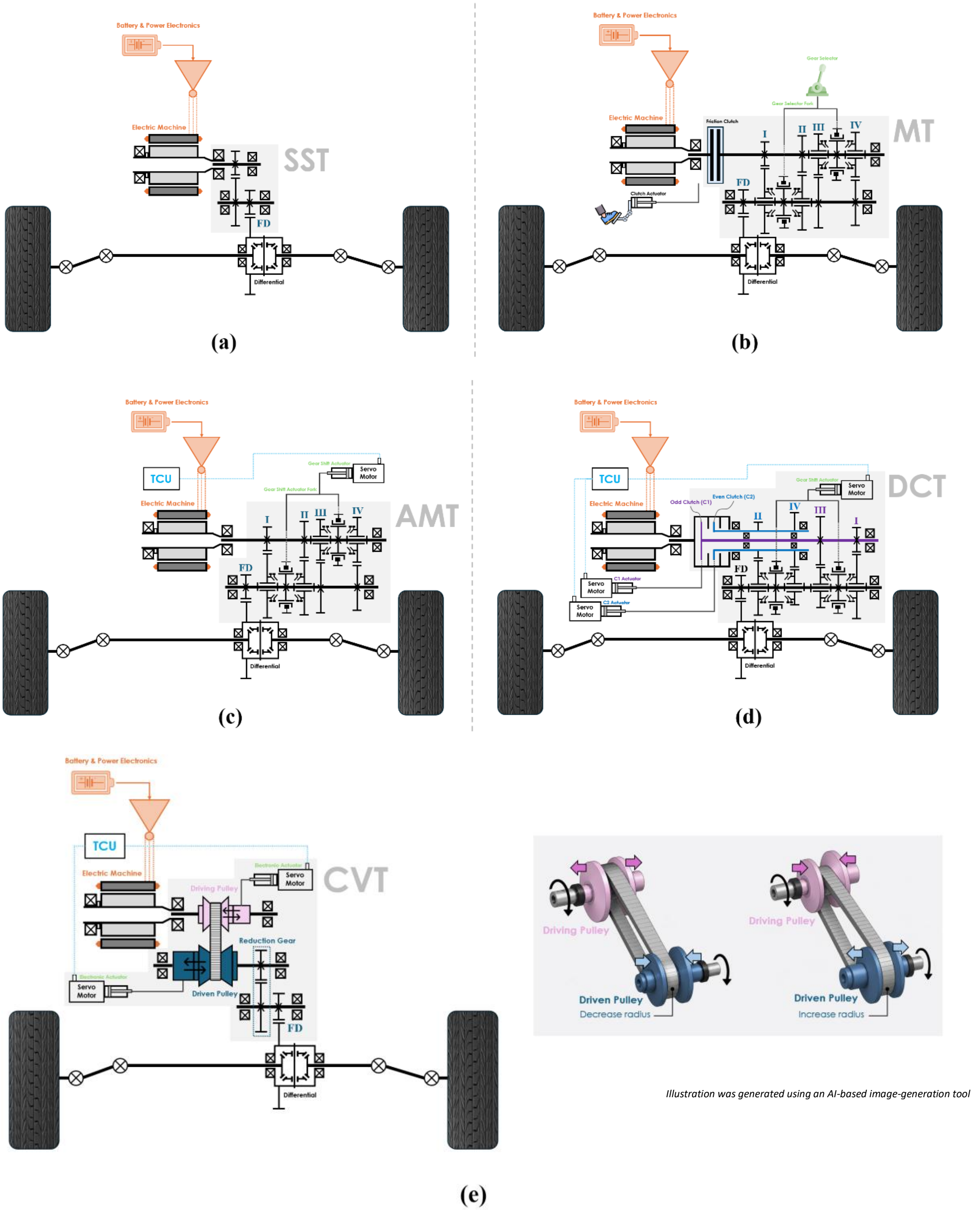


Figure 8. BEV e-powertrain architectures with different transmission topologies: (a) SST, (b) MT, (c) AMT, (d) DCT, (e) CVT

4. TRANSMISSION POWER LOSSES AND EFFICIENCY

Among the various torque-conversion and transmission mechanisms, cylindrical gear systems provide a favorable compromise between transmission ratio, efficiency and cost [30]. However, transmission efficiency is not constant and depends on operating conditions such as torque, rotational speed, temperature, lubrication, and component characteristics [19, 31].

Transmission power losses can generally be divided into load-dependent losses and load-independent, or speed-dependent losses, as illustrated in figure 9 [31].

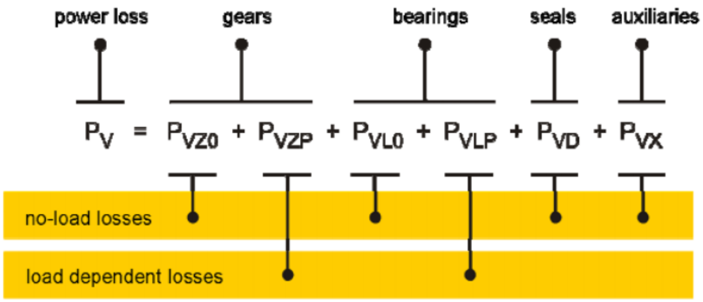


Figure 9. Transmission mechanisms power losses and load dependency [31]

Gear power losses primarily result from tooth-contact friction and are therefore strongly influenced by transmitted load, contact conditions, gear geometry, alignment, lubrication. Additional losses arise from lubricant churning, splashing and compression and windage of oil mist or air, which are mainly related to rotation speed and lubrication system design and conditions [19, 31].

Bearings power losses include load-dependent losses as well as speed-dependent losses associated with lubrication churning and internal friction. Their magnitude is affected by bearing type and size, applied load, lubrication method, lubricant properties and temperature [31].

Further losses originate from dynamic sealings, due to friction between the sealing elements and rotating shafts, as well as from synchronizers and clutches, where frictional interactions occur during operation of gear shifting. In transmission equipped with wet multi-plate clutches, additional hydraulic and viscous losses will occur. Torque converters, if present, introduce further hydrodynamic losses, while auxiliary systems, such as mechanically driven oil pumps and hydraulic actuation systems, consume additional power [19, 31].

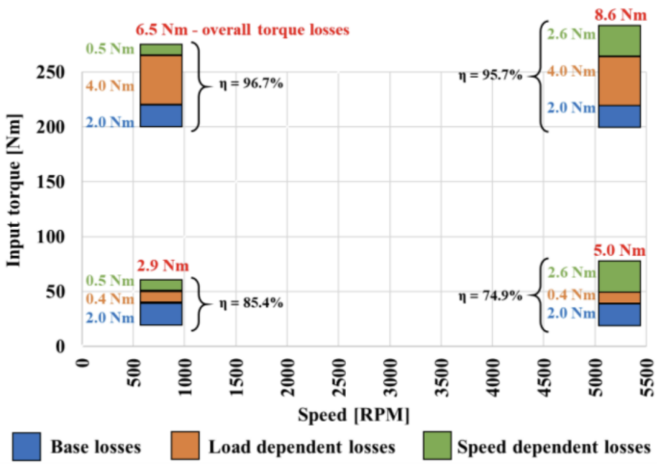


Figure 10. Transmission efficiency, base, load and speed dependent losses in Nm [19]

Figure 10 [19], illustrates the efficiency of a representative gearbox as a function of input torque and rotational speed at a constant temperature, in four different such operating points. Maintaining a fixed temperature isolates the influence of the operating point by preventing changes in lubricant viscosity and its associated adhesion and cohesion properties from affecting the transmission efficiency or torque losses, that are measured in this case. In same figure, the efficiency of the gearbox is lower in lower load conditions in both low and high rotational speed, and higher for greater loads. A significant share of transmission losses are speed-dependent losses and are mainly associated with the mechanical resistance of rotating components, such as bearings, dynamic sealings, or as well as fluid-related effects. The latter include lubricant churning by immersed gears and oil splashing/agitation generated by the differential ring gear, which can act as a mechanical pump. In contrast, the main contribution to overall power losses originates from load-dependent losses, which are mainly given by the friction during gear meshing, which can be also seen in figure 11 [19].

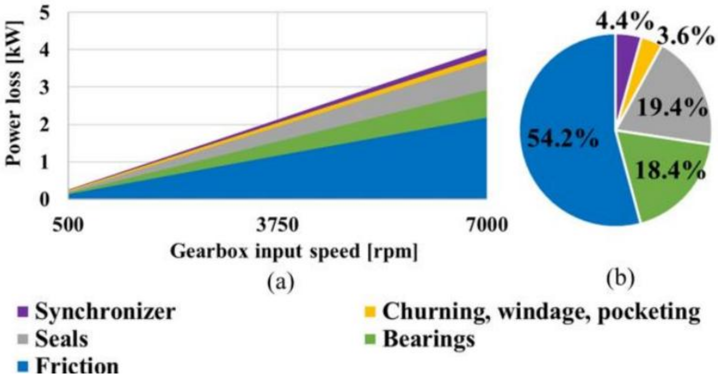


Figure 11. Power loss distribution of a 6-speed MT [19]

Furthermore, according to [19, 22], power losses of a 6-speed MT are significant, reaching approximately 4kW at 7000 rpm in 4th gear, for a 50% load and 80°C oil temperature, as shown in figure 11. In contrast, more efficient transmissions or SSTs with lower losses, such as BorgWarner eGearDrive, may exhibit losses of approximately 1kW at the same rotation speed and operating conditions. This comparison highlights the relevance of transmission losses in the overall efficiency of the electric powertrain, particulary at high rotational speeds.

In DCTs, the clutch assemblies represent a major source of power losses. According to [19], clutch related losses account for approximately up to 75% of the total transmission losses. Therefore, reducing these losses is an important consideration for the application of DCTs in BEVs. According to [13], wet cltuch friction can results in an efficiency reduction of approximately 2-3%.

Unlike conventional transmissions, DCTs perform gear shifting through alternate engagement of the two clutches rathen than through synchronizers [13]. In a 3-speed DCT (figure 12 b), the first and third gears are arranged on one shaft while the second gear is located on the other shaft; for a 4-speed DCT (figure 8 d), gears one and three are assigned to one shaft and gears two and four to the other. For 2-speed DCT (figure 12 a), the absence of idler gears and synchronizers is particularly advantageous, as ratio changes are achived solely by switching between the two clutches associated with the fixed transmission ratios [13].

The transition from a single-speed to a two- or multi-speed transmission generally introduces additional efficiency losses, estimated at approximately 4–5%, mainly due to internal friction, viscous effects and clutch-related losses. For transmissions with more than two ratios, however, the incremental loss associated with each additional gear is comparatively smaller, typically around 2–3%.

Dry-clutch DCTs offer higher efficiency than wet-clutch configurations (Figure 13), but their lower torque capacity limits their applicability to lower input power drivetrains. Their main efficiency advantage results from the absence of an oil cooling circuit, thereby avoiding the churning losses characteristics of wet-clutches. Nevertheless, recent developments in wet-clutch technology, including on-demand cooling and partially dry ("damp") operating conditions, have reduced parasitic losses. Improved efficiency, with future wet-clutch DCTs expected to approach the lower efficiency range of dry-clutch designs [32].

Theoretical analyses and experimental bench tests reported in [33] showed that a 2-speed DCT can achieve an average efficiency of approximately 95%. Efficiency increases with input torque but decreases at higher rotations speeds. Viscous shear in the wet clutch represents the main contributor to resisting torque, followed by oil churning and windage losses, while gear mech losses have a comparatively minor influence. The resisting torque is lower in second gear than in first gear, whereas churning and windage losses are higher. The study did not account for temperature effects, which were identified as a subject for further investigation [33].

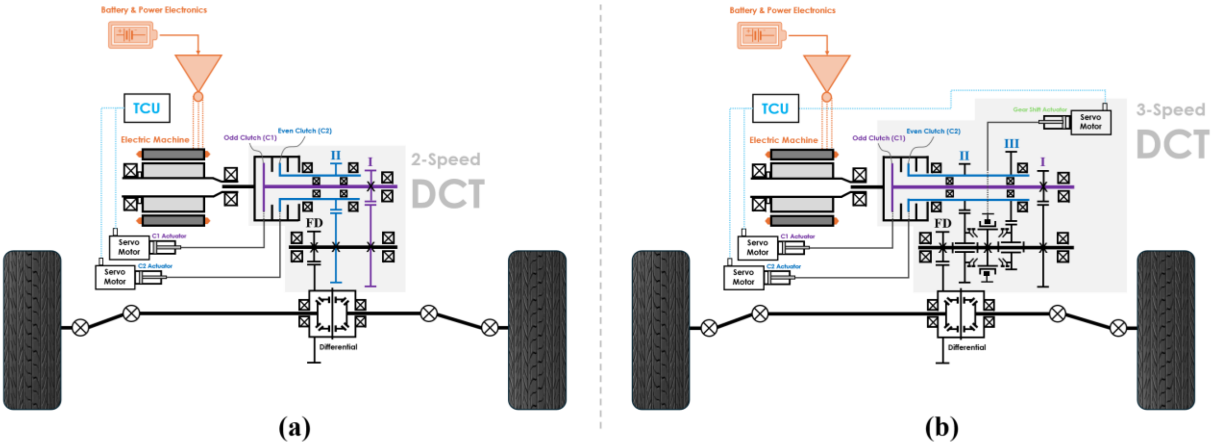


Figure 12. BEV e-powertrain architecture with DCT: (a) 2-speed DCT; (b) 3-speed DCT

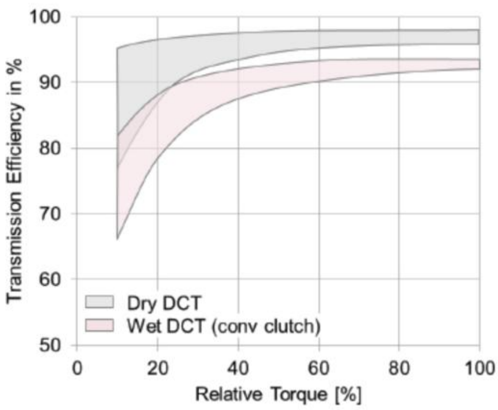


Figure 13. Efficiency comparison of a DCT with wet-clutch vs DCT with dry-clutch [29]

CVTs generally exhibit lower efficiency, which may limit their suitability for BEVs applications. According to [30], an average efficiency of 87% can be achieved, with the main losses attributed to the hydraulic power required to maintain the clamping pressure necessary for maintain selected ratio and especially for variation of it [34, 35]. These losses increase significantly with input torque because higher clamping forces are required, as illustrated in figure 14 [13]. Although the magnitude of the losses depends on the specific transmission design, an additional "over-clamping" pressure is often applied to prevent belt slip during low-speed transient conditions, further increasing energy losses [32].

Improving the efficiency of the CVT ratio-control system is therefore essential for increasing the overall transmission efficiency and improving its suitability for BEVs applications. In this context, [13] proposes a simplified CVT configuration as presented in figure 8 e, that were validated for electric-vehicle applications through simulations and bench testing. As presented in figure 8 e, these systems employ two variable diameter pulleys connected by a chain or metallic belt, with efficiency strongly influenced by the input torque and transmission ratio. The hydraulic systems required to generate and maintain the clamping force introduce additional losses and may limit the response at varying rotational speeds. For BEVs, however, the torque converter can be eliminated and the hydraulic pump replaced by an electromechanical actuator, potentially improving both efficiency and dynamic response. As shown in figure 14, CVT losses increase with input torque, nevertheless, appropriate control and design optimization can bring the efficiency of simplified CVTs closer to that of DCTs, says [13].

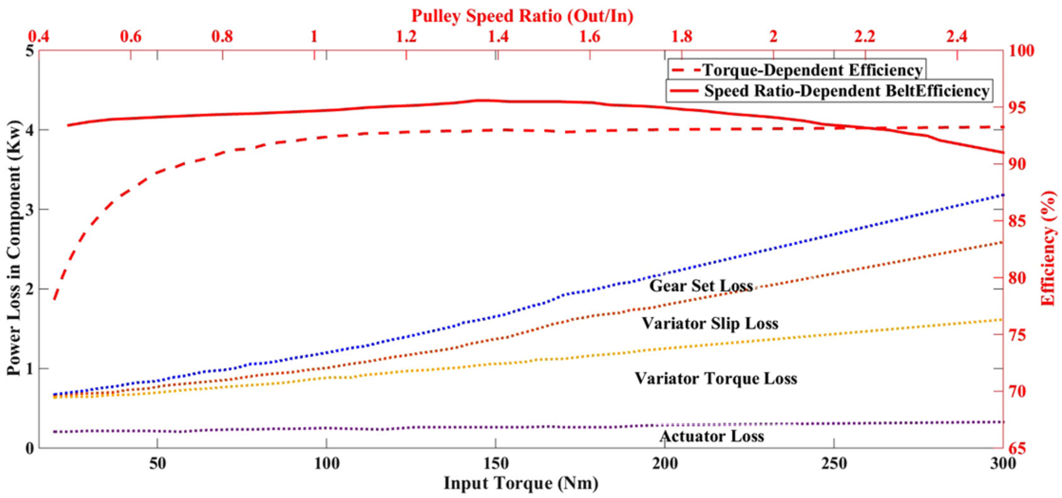


Figure 14. Power losses and efficiencies of components of CVTs [13]

According to [31], the dominant sources of losses in a CVT are associated with ratio variation, the oil pump and the torque converter, which together account for more than 80% of the total losses across the investigated driving cycles. Their relative contribution, however, varies considerably with the operating conditions. For the 10–15 cycle, oil-pump losses represent the largest share (47.2%), followed by the torque converter (26.1%) and ratio-variation losses (18.7%). In the US06 cycle, ratio-variation losses become dominant (48.8%), followed by the oil pump (24.7%) and torque converter (10.9%). A similar trend is observed for the NEDC cycle, where the oil pump accounts for 39.5% of the losses, followed by ratio variation (33.0%) and the torque converter (16.4%) [31].

Furthermore, Table 1 summarizes the average efficiencies reported for different transmission architectures and numbers of gear ratios, together with their main power-loss contributors. Based on these characteristics, potential improvement measures are identified to reduce transmission losses and optimize each transmission architecture for integration into BEV powertrains.

Table 1. Transmissions efficiencies and the ways of improvement in context of BEVs

GB type	Number of gears	Average efficiencies		Power loss contributors	Improvement direction
		Efficiency values from literature or commercial transmissions ***(Max. input torque / Max. input RPM.)***	**Reference intervals for efficiency**	**Dominant loss contributors**	**Ways of improving efficiency of transmission in BEVs powertrains**
SST	1	η > 0.97* *(300 Nm / 14000 RPM)* [37]	η ≈ 0.98 [19]	Gear mesh, bearings, seals, oil churning and windage	Improving the lubrication system by introducing a reservoir-type of oil gutter in the upper part of the housings, thereby minimizing the dynamic oil level and reducing churning losses, particularly at medium and high rotational speeds where lubrication losses become significant.
MST	2	η ≈ 97% GKN-BMW i8 *(250 Nm / -)* [38]	-	Gear mesh, bearings, seals, oil churning and windage	Integration into a powertrain equipped with two traction electric motors. Such an architecture can eliminate the need for a clutch/torque converter and a synchronization mechanism during gear shifting by synchronizing the two electric motors [12, 40].
	3	η ≈ 98% MSYS *(- / -)**** [39]			
MT	2	-	η ≈ 092 – 0.97 [30]	Gear mesh, bearings, sealings, synchronizers	Reduced lubrication/churning losses and optimization of gear geometry and bearing systems; however, the need for manual shifting and torque interruption limits its suitability for BEVs.
	3	-			
	4	-			

	5	-			
	6	-			
AMT	2	η ≈ 97% GKN-BMW i8 *(250 Nm / -)* [38]	η ≈ 092 – 0.97 [30]	Gear mesh and shifting/clutch losses Bearings, sealings, synchronizers	[19] The possibility of clutchless operation, enabled by precise control of the electric motor speed and torque. Research has focused on reducing gear-shifting time through advanced control strategies, including two-layer neural networks [41] and multi-body dynamic models combined with closed-loop motor control [42]. Integration into a powertrain equipped with two traction electric motors. Such an architecture can eliminate the need for a clutch/torque converter and a synchronization mechanism during gear shifting by synchronizing the two electric motors [12, 40].
	3	-			
	4	-			
DCT	2	η ≈ 0.86** *(250 Nm / 9000 RPM)* [13] η ≈ 0.95* [33]	η ≈ 090 – 0.95 [30, 33]	Wet-clutch viscous losses, churning, windage Gear mesh, bearings, sealings, synchronizers	Elimination of synchronizers, as gear changes are performed exclusively by switching between two clutches corresponding to fixed transmission ratios. Use of dry clutches and improved lubrication, achieved by reducing the dynamic oil level and, consequently, minimizing churning losses. Integration into a powertrain equipped with two traction electric motors. Such an architecture can eliminate the need for a clutch/torque converter and a synchronization mechanism during gear shifting by synchronizing the two electric motors [12, 40].
	3	η ≈ 0.83** *(250 Nm / 9000 RPM)* [13]			Use of dry clutches and improved lubrication, achieved by reducing the dynamic oil level and, consequently, minimizing churning losses. Integration into a powertrain equipped with two traction electric motors. Such an architecture can eliminate the need for a clutch/torque converter and a synchronization mechanism during gear shifting by synchronizing the two electric motors [12, 40].
	4	η ≈ 0.80** *(250 Nm / 9000 RPM)* [13]			
CVT	n	η ≈ 0.89** *(250 Nm / 9000 RPM)* [13]	η ≈ 0.87 – 0.93 [33]	Hydraulic pump, clamping/ratio variation, torque converter, reverse mechanism	Elimination of the torque converter, replacement of hydraulic actuation with electromechanical actuation, optimization of clamping force to reduce belt/chain slip and over-clamping losses, and reduction of hydraulic/pump losses.

*efficiency results of tests

**efficiency estimated by calculation based on power losses calculations or approximations

***MT with "DSD Overlap Shift" technology uses powershift elements and multi-cone clutches to separate the friction and locking functions of the synchronizer, ensuring continuous torque transfer without energy consumption to maintain the engaged gear [39]

5. SIMULATION BASED EVALUATION METHODOLOGY OF PERFORMANCE (DYNAMIC & ENERGETIC) AND PERFORMANCE INDICATORS (KPI)

Simulation-based approaches are widely used to evaluate the dynamic and energetic performance of BEV powertrains under standardized driving cycles. Two main modelling approaches are commonly employed: *forward* and *backward* simulation, which differ primarily in the direction of the energy and information flow through the powertrain [43, 44].

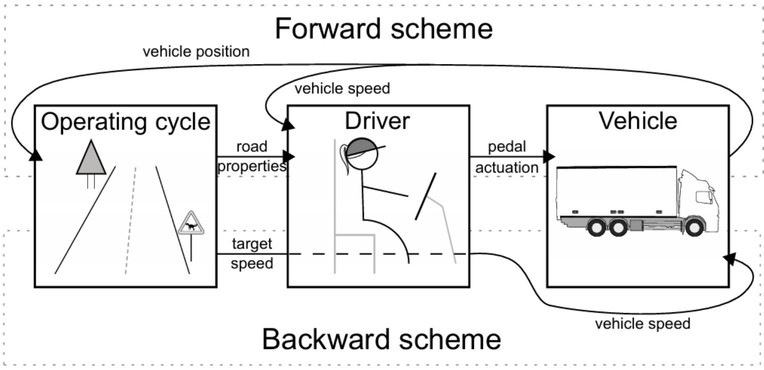


Figure 15. Topology of a conceptual simulation model, showing the forward method at the top and the backward method at the bottom [44]

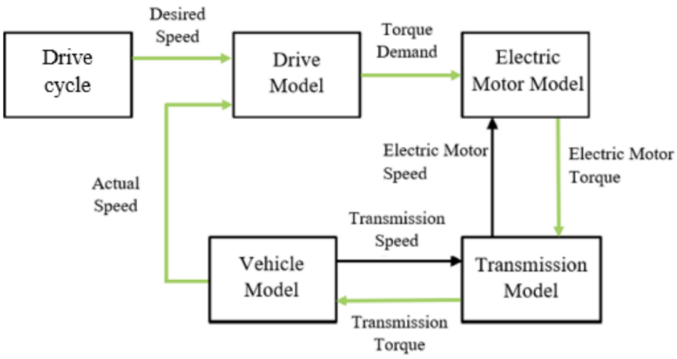


Figure 16. Schematics of simulations that uses forward method [43]

In *forward simulation*, as showed in figures 15 and 16, the driving cycle provides the target vehicle speed, which is tracked by a driver/controller model that generates the required acceleration or braking commands. The resulting torque and power are then propagated through the electric machine, transmission and vehicle model toward the wheels. The vehicle response is calculated from the longitudinal dynamics, and the resulting speed is fed back to the driver model, forming a closed-loop simulation. Consequently, forward simulation reproduces the physical cause–effect relationships of the vehicle and is particularly suitable for evaluating dynamic behaviour, including acceleration, gear shifting and drivability.

In contrast, *backward simulation* starts from the prescribed vehicle speed profile and determines the force, torque and power required at the wheels. These requirements are propagated upstream through the transmission and electric machine to determine the corresponding energy demand from the battery. This approach is computationally simpler and is therefore particularly suitable for energy-consumption and efficiency analyses over standardized driving cycles.

The vehicle longitudinal model is based on the force balance between the tractive force generated at the wheels and the main resistances opposing vehicle motion, including rolling resistance, aerodynamic drag, road-gradient resistance and acceleration resistance. These relationships provide the basis for determining the torque and power requirements of the electric powertrain in both simulation approaches.

For the present review, the distinction between the two methods is particularly relevant because the objective is to assess the benefits of single-speed versus multi-speed transmissions. Forward models are more appropriate when the analysis focuses on dynamic performance, shift behavior and drivability, whereas backward models provide an efficient framework for comparing energy consumption, drivetrain efficiency and transmission operating points over standardized driving cycles. Consequently, the choice of simulation methodology should be linked to the specific performance indicator and optimization objective investigated in each study.

This distinction is summarized in Table 2, which compares the two approaches in terms of input/output variables, data flow, advantages, limitations and their applicability to SST–MST comparison and transmission optimization.

Table 2. Comparison between Forward and Backward simulation methods

		Forward Method Direct – natural causality Cause → Effect	Backward Method Indirect– non-natural causality Effect → Cause
Input		Acceleration / braking commands (DRIVER INPUT)	Driving cycles (e.g. WLTP, NEDC, HWFET, FTP-75, UDDS, etc…)
Output		Vehicle speed, position and travelled distance, energy consumption, etc…	Required torque, energy consumption, losses, etc…
Simulation data flow		Control system → Vehicle → Motor → Transmission → Drive Wheels	Drive Wheels → Transmission → Energy Source (Motor , Inverter + Baterry)
Advantages		• Reproduces vehicle dynamic behaviour with low error; • Suitable for transient and dynamic analyses (e.g., acceleration, gear shifting, drivability);	• Suitable for energy comparisons; • Does not require detailed and complex modelling of control systems; • Much faster; • Well suited to optimization simulations using standardized driving cycles;
Disadvantages		• Requires complex modelling with an elaborate inter-system control strategy; • Longer simulation times;	• Cannot reproduce vehicle dynamic behaviour with high accuracy;
Analysis objective **(SST vs MST comparisons and MST optimization)**	**Comparison of energy performance (range, energy consumption)**		✗
	Powertrain efficiency		✗
	Dynamic performance analysis (acceleration, gear shifting, response, etc.)	✗	
	Transmission optimization (control and gear shifting)	✗	
	Parameter sensitivity study		✗
	"Driveability" validation with different transmission topologies	✗	

As illustrated in Figure 17 and reported in [12, 45], BEV performance can be assessed through dynamic and energetic indicators (so called Key Performance Indicators - KPIs).

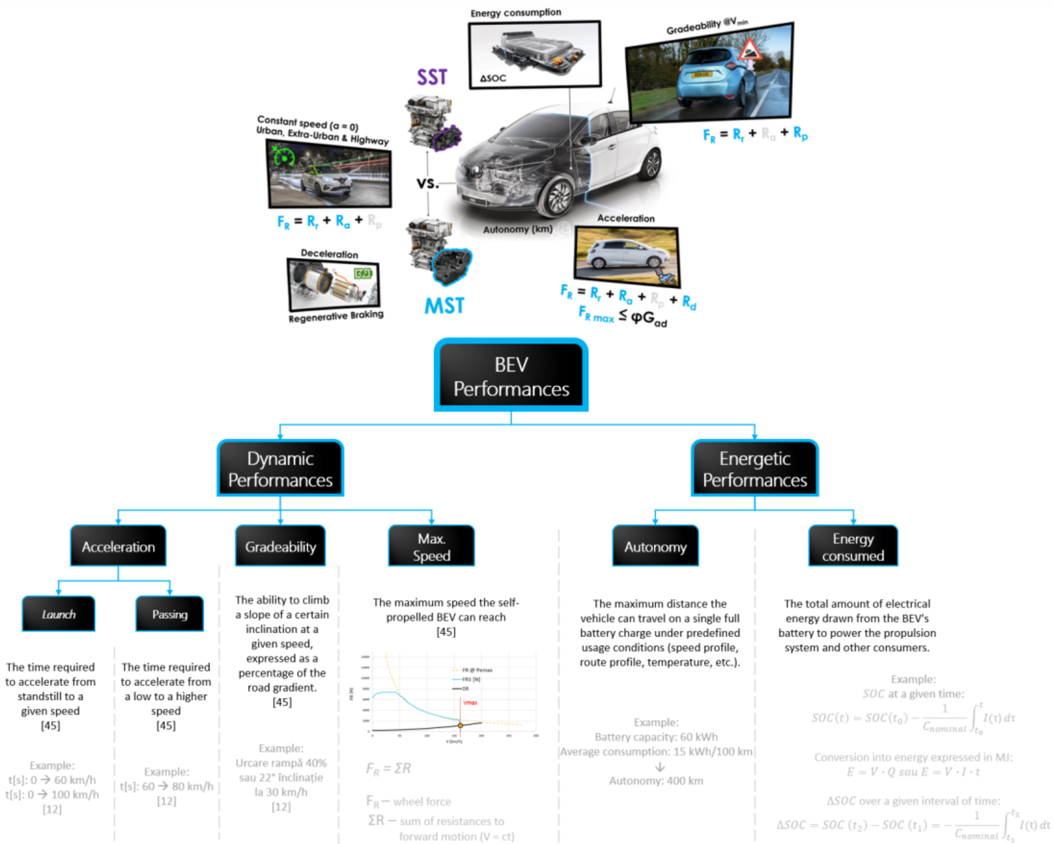


Figure 17. Key performance indicators which evaluate performances of BEV

Dynamic performance is characterized primarily by acceleration time, passing acceleration, gradeability, and maximum speed. Acceleration performance may be evaluated using 0–60 or 0–100 km/h times, while passing capability can be expressed through the time required to increase vehicle speed over a defined interval. Gradeability represents the maximum road inclination that the vehicle can negotiate at a specified speed.

Energetic performance is mainly evaluated through driving range and energy consumption. Range represents the maximum distance achievable with a fully charged battery under specified operating conditions, whereas energy consumption quantifies the electrical energy required to propel the vehicle, typically expressed per unit distance or through the corresponding battery SOC variation.

Figure 17 summarizes these KPIs considered for evaluating the impact of SST and MST configurations on BEV performance.

6. DYNAMIC-ENERGETIC PERFORMANCES TRADE-OFF

MST in BEVs powertrain architecture does not inherently guarantee simultaneous improvement of dynamic and energetic performances. The additional transmission ratios provide greater freedom to operate the EM within the favourable operating regions, but the associated transmission losses and control requirements may offset part of the potential benefit.

The results discussed in the previous sections indicate that the magnitude and direction of these effects depend strongly on the transmission architecture, gear-ratio selection, shift strategy, electric-machine characteristics and driving conditions. Therefore, considering dynamic and energetic performance independently may not fully capture the overall benefit of an MST. To investigate this interaction, the studies identified in the literature review for which both dynamic and energetic performance data were available are summarized in table 3.

Table 3. Dynamic and energetic performance improvements reported for BEVs with MST relative to their original SSTs

Author / Configuration	Dynamic Improvement	Energetic Improvement	Drive Cycle
Gao et al. / 2-speed I-AMT [12]	0–100 km/h: **+19.5% faster**; top speed: **+31%;**	**7.3–11.3% lower energy consumption**	**NEDC UDDS 1015**
Zhou et al. / 2-speed [12]	0–60 km/h: 5.8 → 4.9 s (**+18.4% faster**); top speed: 102 → 198 km/h (**+94.1%**); gradeability: 23.45 → 43.67% (**+86.2%**)	**+2–4% driving range**	**NEDC**
Walker et al. / 2-speed [17]	0–60 km/h: 6.48 → 5.22 s (**+19.4% faster**); top speed: 151 → 222 km/h (**+47.0%**); gradeability: 25 → 45% (**+80%**)	**Lower driving range** than SST in the reported case	**HWFET UDDS**
Tian et al. / 2-speed E-bus [12]	top speed: 99 → 102 km/h (**+3.03%**); 0-60 km/h: 18.9 → 18.8 s (**+0.53% faster);** gradeability 21.1 → 21.3% (**+1%**)	**10.36% energy saving**	**CUDC**
Tian et al. / 4-speed E-bus [12]	top speed: 99 → 104 km/h (**+5.05%**); 0-60 km/h: 18.9 → 18.5 s (**+2.12% faster**); gradeability 21.1 → 21.9% (**+3.79%**)	**29.41% energy saving**	**CUDC**
Yin et al. / optimized 2-speed [12]	Dynamic targets maintained	**+4.29% range (AC ON); +5.28% (AC OFF)**	-
Bertucci et al. / Tesla S60, 2-speed DCT [16]	0–100 km/h: 6.54 → 6.31 s (**+3.5% faster**)	**5.75% lower energy consumption**	**WLTC**

Bertucci et al. / Tesla S60, CVT **[16]**	0–100 km/h: 6.54 → 5.50 s (**+15.9% faster**)	**7.40% lower energy consumption**	**WLTC**
Bertucci et al. / Nissan Leaf, 2-speed DCT **[16]**	0–100 km/h: 12.01 → 11.81 s (**+1.7% faster**)	**3.75% lower energy consumption**	**WLTC**
Bertucci et al. / Nissan Leaf **CVT** **[16]**	0–100 km/h: 12.01 → 10.91 s (**+9.2% faster**)	**7.00% lower energy consumption**	**WLTC**
Ruan et al. / CVT **B-segment** **[47] / [13]**	0–60 km/h: 7.3 → 6.3 s (**+13.7% faster**); 0–100 km/h: 14.4 → 13.4 s (**+6.9% faster**); top speed: 112 → 181 km/h (**+61.6%**); gradeability: 48 → 60% (**+25%**) [47]	**17.4% lower energy consumption [13]**	**FTP-75 + HWFET (combined)**
Ruan et al. / 2-speed DCT **B-class** **[47] / [13]**	0–60 km/h: **0%**; 0–100 km/h: **0%**; top speed: **+57.1%**; gradeability: **0%** [47]	**16.4% lower energy consumption [13]**	**FTP-75 + HWFET (combined)**
He et al. / 2-speed DCT **[46]**	0–50 km/h: 5.48 → 3.72 s (**+32.1% faster**); 0–100 km/h: 11.19 → 9.45 s (**+15.5% faster**); top speed: 169 → 228 km/h (**+34.9%**); gradeability: 25 → 36% (**+44%**)	**Up to +18.25% regenerative-energy recovery**	**WLTC**

The reported results cannot be compared directly in absolute terms, since the investigated vehicles, transmission configurations, performance indicators and driving cycles differ between studies. Nevertheless, they provide useful evidence for assessing the relationship between dynamic enhancement and energetic improvement and for identifying cases in which improvements in one performance category are accompanied by limited or even negative effects in the other.

A particularly illustrative case is the study of Walker et al. [17], with the results extracted in table 4, which demonstrates that the introduction of an MST does not necessarily translate into an energetic improvement, even when substantial dynamic-performance gains are achieved. For the optimized 2-speed configuration, the 0–60 km/h acceleration time was reduced from 6.48 s for the optimized SST to 5.22 s, corresponding to a 19.4% improvement, while the maximum speed increased from 151 to 222 km/h (+47%) and gradeability from 25% to 45% (+80%). In contrast, the 0–100 km/h acceleration remained practically unchanged, with 13.48 s for the SST and 13.49 s for the MST. These results indicate that the principal benefit of the two-speed transmission was the extension of the available wheel-torque and vehicle-speed operating range rather than a systematic improvement of all acceleration metrics.

The energetic results reveal an important trade-off. Despite the optimized two-speed transmission increasing the mean motor efficiency from 86.0% to 87.2% on HWFET and from 79.0% to 80.2% on UDDS, the reported driving range remained lower than that of the optimized SST. The range decreased from 157.6 to 141.8 km on HWFET (−10.0%) and from 142.1 to 130.4 km on UDDS (−8.2%). The difference is mainly associated with the lower assumed overall powertrain efficiency of the two-speed configuration, which was taken as 80%, compared with 90% for the SST.

The study is particularly relevant because gear-ratio optimization was also performed using a genetic algorithm. The optimized ratios were 6.82 for the SST and 11.47/4.64 for the two-speed transmission. The optimization simultaneously targeted mean motor efficiency and driving range while constraining the required dynamic performance. Thus, the results demonstrate that gear-ratio optimization can substantially improve the dynamic capability of an MST, but that this optimization alone does not guarantee an energetic advantage when the transmission itself introduces higher losses.

This case therefore represents a strong dynamic-improvement / energetic-penalty region in the proposed dynamic–energetic trade-off, and highlights the importance of considering transmission efficiency together with gear-ratio selection when evaluating MST benefits in BEVs.

Table 4. Dynamic and energetic performance of the optimized SST and 2-speed transmission reported by Walker et al. [17]

Parameter	Optimized SST	Optimized 2-speed	Change
Gear ratio(s)	6.82	11.47 / 4.64	-
HWFET range	157.6 km	141.8 km	**−10.0%**
UDDS range	142.1 km	130.4 km	**−8.2%**
Mean motor efficiency, HWFET	86.0%	87.2%	**+1.2%**
Mean motor efficiency, UDDS	79.0%	80.2%	**+1.2%**
0–100 km/h	13.48 s	13.49 s	**≈ 0%**
0–60 km/h	6.48 s	5.22 s	**19.4% faster**
Top speed	151 km/h	222 km/h	**+47.0%**
Gradeability	25%	45%	**+80%**

A particularly favourable case is reported for the 4-speed E-bus investigated by Tian et al. with the data from [12], where the maximum speed increased from 99 to 104 km/h, the acceleration time decreased from 18.9 to 18.5 s, and gradeability increased from 21.1% to 21.9%. These dynamic improvements were accompanied by a 29.41% reduction in energy consumption over the CUDC cycle. These results represent a strong example of simultaneous dynamic and energetic improvement enabled by the additional transmission ratios.

The B-segment vehicle investigated by Ruan et al. [13, 47] provides a particularly strong example of simultaneous dynamic and energetic improvement. The CVT configuration reduced the 0–60 km/h and 0–100 km/h acceleration times from 7.3 to 6.3 s and from 14.4 to 13.4 s, corresponding to improvements of 13.7% and 6.9%, respectively. The maximum vehicle speed increased from 112 to 181 km/h (+61.6%), while gradeability increased from 48% to 60% (+25%). These improvements demonstrate the ability of the continuously variable ratio to decouple the acceleration and maximum-speed requirements from the electric-machine operating range. The CVT ratio was continuously varied between 0.5 and 2.5, with a final-drive ratio of 4, compared with a 2.15 single-speed ratio and a 2.15/1.46 two-speed DCT configuration, respectively.

The energetic results are equally significant. For the B-segment vehicle, electricity consumption decreased from 10.3 to 7.02 kWh/100 km in the city cycle, from 11.6 to 8.07 kWh/100 km on the highway cycle, and from 11.0 to 7.6 kWh/100 km in the combined cycle. These values correspond to calculated reductions of approximately 31.8%, 30.4% and 30.9%, respectively. In addition, the authors report a 17.4% reduction in energy consumption for the CVT configuration relative to the reference SST under their defined evaluation metric [13]. The battery capacity required to achieve a 200-km range was also reduced from 28 to 18 kWh [13].

The improvement is associated with the ability of the CVT to continuously adapt the transmission ratio according to vehicle speed and accelerator demand, maintaining the electric machine closer to its high-efficiency operating region. The motor operating-point analysis confirms this behaviour: in both HWFET and FTP-75, the CVT configuration (green points on motor map in figure 18) keeps the motor operating points closer to the high-efficiency regions than the single-speed and stepped-transmission configurations. In contrast, the two-, three- and four-speed DCTs show progressively more favourable operating regions but with increasingly limited additional benefit [13, 47]. Importantly, the energetic improvement does not increase monotonically with the number of discrete transmission ratios. The combined-cycle energy consumption decreases from 11.0 kWh/100 km for the SST to 9.2 kWh/100 km for the 2-speed DCT and 8.9 kWh/100 km for the 3-speed DCT, but slightly increases to 9.0 kWh/100 km for the 4-speed DCT. This indicates that the additional reduction in electric-machine losses obtained by introducing further ratios can be offset by the additional transmission losses and associated operating conditions. The CVT achieves the lowest reported consumption, 7.6 kWh/100 km, by continuously adapting the transmission ratio and therefore providing greater freedom to maintain the electric machine within favourable efficiency regions.

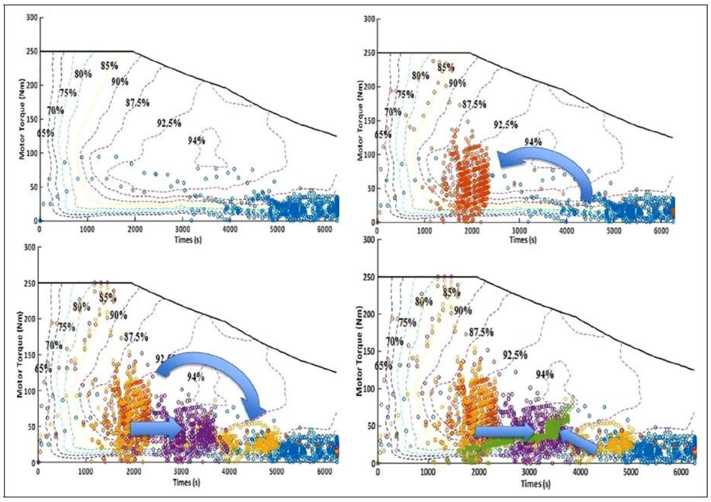


Figure 18. Electric motor operating points after the HWFET cycle with a B-segment BEV: top-left – SST; top-right – 2-speed DCT; bottom-left – 3-speed DCT; bottom-right – 4-speed DCT; CVT – green points [13]

More moderate but consistent improvements are reported by Bertucci et al. [16] for both the Tesla S60 and Nissan Leaf equipped with a 2-speed DCT. For the Tesla S60, the 0–100 km/h acceleration time was reduced by 3.5%, accompanied by a 5.75% reduction in energy consumption. For the Nissan Leaf, the corresponding improvements were 1.7% and 3.75%, respectively. These results suggest that even relatively modest dynamic improvements can be accompanied by measurable energetic benefits when the additional transmission ratio is effectively exploited.

Table 5. Transmission characteristics and energetic performance of alternative B-class BEV configurations of study by Ruan et al. [13]

B-class configuration	**Transmission ratio(s)**	**Final drive**	**Overall transmission efficiency***	**City [kWh/100 km]**	**Highway [kWh/100 km]**	**Combined [kWh/100 km]**	**Reported energy-consuption reduction**	**Battery capacity for 200-km range [kWh]**
SST	2.0	4	93%	10.30	11.60	11.00	0%	28
2-speed DCT	2.7 / 0.9	4	86%	9.04	9.40	9.20	**+16.4%**	23
3-speed DCT	2.7 / 1.6 / 0.9	4	83%	8.74	9.10	8.90	**+3.3%**	23
4-speed DCT	2.7 / 1.9 / 1.3 / 0.9	4	80%	9.10	8.90	9.00	**−1.1%**	23
CVT	0.72–2.16	5	78–89%	7.02	8.07	7.60	**+17.4%**	18

The 2-speed DCT investigated by He et al. [46] exhibits the largest dynamic improvement among the cases considered, with acceleration improvements of 32.1% for 0–50 km/h and 15.5% for 0–

100 km/h, together with increases of 34.9% in maximum speed and 44% in gradeability. The energetic benefit, however, is reported in terms of regenerative-energy recovery, which increased by up to 18.25%, rather than as a reduction in driving-cycle energy consumption. Therefore, although this case demonstrates the potential of MSTs to simultaneously enhance dynamic capability and regenerative performance, it is not directly comparable with the energy-consumption-based results and should be treated separately in the normalized trade-off analysis.

Table 6. Synthesis of dynamic & energetic performance trade-offs identified in selected BEV MST studies

Case	Dynamic performance	Energetic performance	Main observation
Walker et al. [17]	Strong improvement	Energetic penalty	Demonstrates that substantial dynamic gains do not necessarily result in improved energy performance.
Tian et al. [12] – 4-speed	Moderate improvement	Very large improvement	Demonstrates that simultaneous dynamic and energetic improvements can be achieved with an appropriately selected multi-speed configuration.
Ruan et al. [13,47] – CVT	Strong improvement	Strong improvement	Demonstrates the potential of continuous ratio variation to improve both dynamic and energetic performance.
He et al. [46] – 2-speed DCT	Largest dynamic improvement	Regenerative-energy improvement	Demonstrates a strong dynamic benefit, but uses a different energetic KPI and is therefore not directly comparable with energy-consumption cases.
Bertucci et al. [16]	Moderate improvement	Moderate improvement	Represents a more balanced improvement between dynamic and energetic performance.

In order to select the proper gear ratios for an MST of a BEV, multiple research paper uses different gear ratio optimization criterias that follows to maximize the energy performance of the e-powertrain, while simultaneously maintaining the requirements of dynamic performances (acceleration, top speed, gradeability). In this context:

- First gear ratio is sized in order to ensure good launch performance and gradeability;
- The last gear ratio is selected to enable the desired maximum vehicle speed while maintaining an efficient operating range for the e-machine;
- Intermediate gear ratios (applicable for transmissions with more than two speeds) are introduced to limit large gear ratio steps, which can lead to slow or abrupt shifts and efficiency losses;

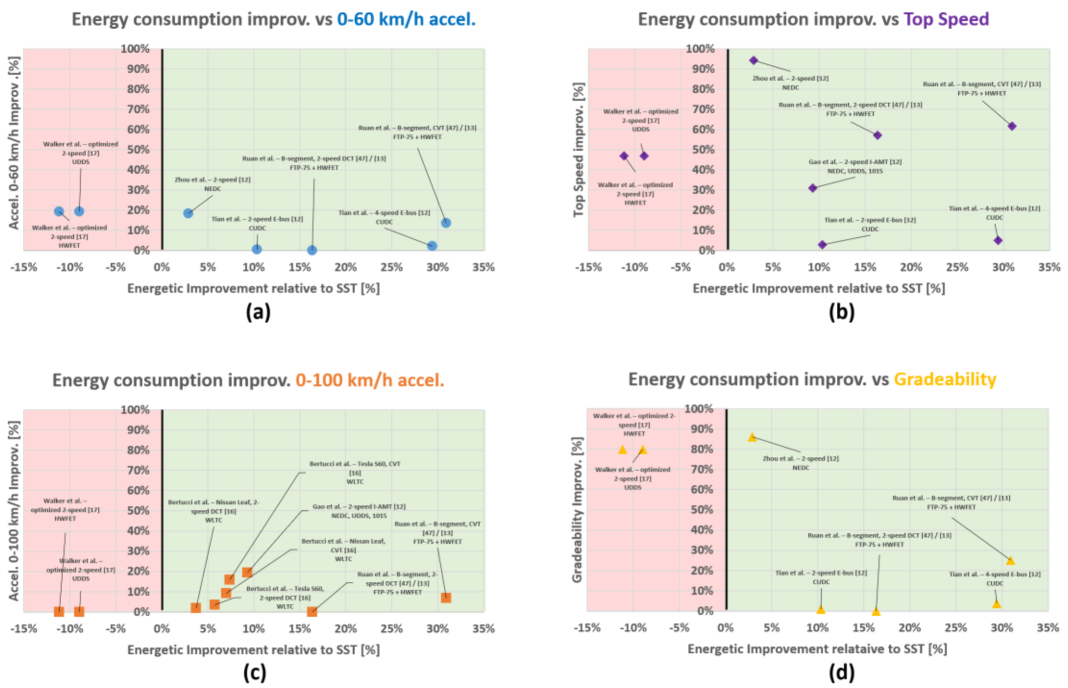


Figure 19. Dynamic-Energetic performance trade-offs of BEVs with MST relative to SST: (a) 0-60 km/h acceleration; (b) top speed; (c) 0-100 km/h acceleration; (d) gradeability

Therefore, the selection of transmission gear ratios involves a multi-objective optimization, which requires a trade-off between energy efficiency, dynamic performances, driving comfort and mechanical complexity with associated costs.

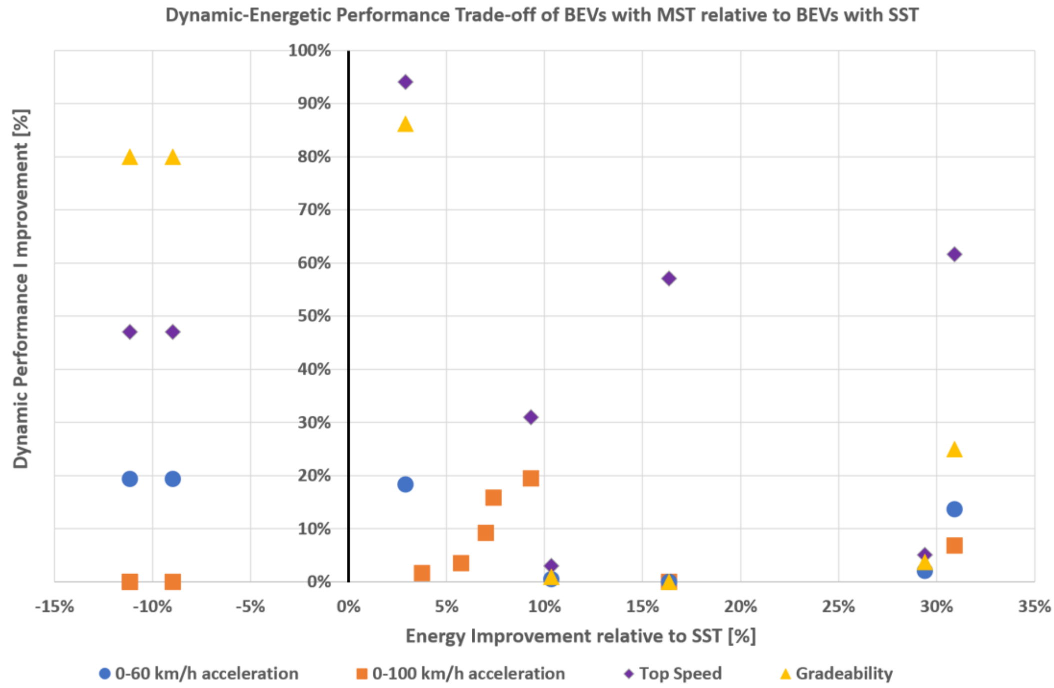


Figure 20. Combined dynamic-energetic performance trade-off of BEVs with MST relative to BEVs with SST

Figure 19 (a-d) illustrate the relationship between the energetic and dynamic performance improvements obtained with MSTs relative to SSTs. The horizontal axis represents the improvement in energy consumption relative to the SST BEVs configuration, while the vertical axes represent the improvements in 0-60 km/h acceleration, 0-100 km/h acceleration, top speed and gradeability, respectively, also, relative to SST BEVs configurations. The green region corresponds to positive energetic improvements i.e., reduced energy consumption with MST on several cycles that are stated in the points description, whereas the red region indicates and energetic penalty despite the dynamic-performances improvements.

The results show that the introduction of an MST does not necessarily lead to simultaneous improvements in energetic and dynamic performances. For example, the optimized two-speed configuration investigated by Walker et al. [17] archives approximately 47% higher top speed, 80% higher gradeability and 19.4% improvement in 0-60 km/h acceleration, but is associated with an energetic penalty of approximately 9-11%, depending on the driving cycle. Conversely, the Tian. et al. four speed E-bus, with extracted data from review paper [12], achieves a 29.4% reduction in energy consumption, while the improvements in 0-60 km/h acceleration, top speed and gradeability are approximately 2.1%, 5.1%, 3.8% respectively. This represents a more balanced energetic benefit, although with comparatively moderate dynamic gains.

The Ruan et. al. CVT [47] / [13] configuration provides one of the strongest simultaneous improvements, with approximately 30.9% lower energy consumption, together with improvements of 13.7% in 0-60 km/h acceleration, 6.9% in 0-100 km/h acceleration, 61.6% in top speed and 25% in gradeability. Similarly, the Zhou et al. two speed configuration, with data extracted from review paper [12] shows a relatively small energetic improvement of approximately 2.9%, but very large improvements in top speed (94.1%) and gradeability (86.2%). These cases demonstrate that the magnitude of energetic and dynamic improvements is not necessarily proportional.

The combined representation in figure 20 confirms that MST configurations can occupy substantially different regions of the dynamic-energetic performance space. Some configurations provide simultaneous energetic and dynamic improvements, whereas others achieve significant dynamic benefits at the expense of energy consumption. Overall, in the papers studied, the results indicate that the performance benefit of an MST depends not only on the number of gears, but also on the transmission architecture, gear-ratio selection, shift strategy, vehicle characteristics, operating conditions or driving cycles followed. Therefore, the results support a multi-objective optimization approach, in which energetic and dynamic requirements are considered simultaneously rather than independently.

7. SUMMARY, MAIN FINDINGS AND FUTURE DIRECTIONS

The review investigated the potential benefits and limitations of integrating multi-speed transmissions (MSTs) into battery electric vehicle (BEV) powertrains, with particular emphasis on dynamic performance, energetic performance, transmission efficiency, gear-ratio optimization and shift strategies. A unified database was developed to collect vehicle characteristics, electric-machine parameters, transmission architectures, gear ratios, efficiency data, optimization methods, driving cycles and reported performance indicators, enabling a quantitative comparison of results reported across the studied literature.

The analysis confirms that the main motivation for introducing an MST is the possibility of decoupling vehicle-level requirements from the operating range of the electric machine. Additional transmission ratios can increase wheel torque during launch and gradeability conditions, reduce motor speed at high vehicle speeds, and allow the electric machine to operate closer to its high efficiency region. However, these potential benefits must be evaluated together with the additional losses, mass and control complexity of introducing the MST.

The quantitative comparison demonstrates that MST does not inherently provide simultaneous dynamic and energetic improvements. The results vary substantially between studies and depend on the transmission architecture, gear ratio selection, shift strategy, vehicle characteristics and driving cycles. For example, Walker et al. [17] reported improvements in 0-60 km/h acceleration, top speed and gradeability, but with an energy consumption penalty. In contrast, Tian et al. [12] achieved a strong reduction in energy together with moderate improvements in 0-60 km/h acceleration, top speed and gradeability.

The strongest simultaneous improvement identified in the reviewed cases was reported for the B-segment vehicle equipped with a CVT by Ruan et al. [47] / [13], with a strong 30.9% energy consumption on FTP-75 + HWFET combined cycles, while 0-60 km/h and 0-100 km/h acceleration improved by 13.7% and 6.9% respectively. The same configuration increased top speed by 61.6% and gradeability by 25%. These results illustrate the potential of CVT to simultaneously address dynamic and energetic requirements.

The analysis of transmission losses shows that the energetic benefit of an MST cannot be addressed solely from the electric machine efficiency improvement. Gear meshing friction, bearing and sealing losses, lubricant churning, windage, clutch losses and auxiliary-system consumption can offset the improvement obtained by moving the electric machine toward a more favourable operating region. In particular, the literature indicates that transmission losses become increasingly important at high rotational speeds and low load operating conditions. DCT clutch losses can also represent a significant contribution to total transmission losses of this topology of gearbox. CVT belt/chain frictional losses and clamping forces of shifting and maintaining the selected gear ratio also represent a significant contribution to total transmission losses of CVT gearbox topology.

Gear ratio selection and shift strategy therefore emerge as two of the most important design variables. The reviewed studies show that the first ratio is generally associated with launch and gradeability requirements, while the highest ratio is primarily related to maximum speed and high speed motor operation. Intermediate ratios are used to reduce excessive gear ratios cannot be considered independently from the shift strategy, since each ratio set requires an appropriate shifting schedule.

The dynamic-energetic trade-off analysis presented in figures 19 and 20 provides a consolidated view of the reviewed results. The figures demonstrate that MST configurations can occupy very different regions of the performance space: from strong dynamic improvement accompanied by an energetic penalty, as in Walker et al. [17], to simultaneous strong dynamic and energetic improvement, as demonstrated by Ruan et al. [47] / [13], and to configurations prioritizing energetic improvement while providing more moderate dynamic gains, such as Tian et al. 4-speed E-bus [12]. Consequently, the number of transmission ratios alone is not a sufficient indicator of the expected benefit of an MST on a BEV.

Despite the increasing number of studies on BEV MST, several research gaps remain:

- first, a standardized methodology for comparing SST and MST configurations is still lacking. The review studies use different vehicle characteristics, battery assumptions, electric machines, driving cycles and performance definitions, which makes direct comparison difficult. A common evaluation framework considering identical vehicle, battery and electric-machine conditions would significantly improve the reliability of quantitative comparisons.
- second, transmission losses require greater attention in the optimization process. Several studies primarily optimize electric machine operation or driving cycle energy consumption, while detailed transmission losses are not always represented. Future optimization should therefore consider efficiency maps of the complete transmission, including gears, bearings, clutches, lubrication and auxiliary systems, rather than applying a constant mean transmission efficiency.
- third, gear ratio optimization and shift strategy optimization should be treated as a coupled problem. The literature indicates that an optimized set of ratios does not automatically result in optimum vehicle performance if the corresponding shift schedule is not properly designed. Future research should investigate integrated optimization of gear ratios, shift points and shift quality under multiple driving conditions.
- fourth, experimental validation remains insufficient compared with simulation-based investigations. Many reported improvements originate from numerical simulations, while experimental or real vehicle validation is less frequently available. Future studies should therefore place greater emphasis on hardware in the loop, transmission bench and vehicle level validation under representative driving cycles.

Finally, future MST design should increasingly adopt a multi-objective optimization framework that simultaneously considers energy consumption, electric machine efficiency, acceleration, maximum speed, gradeability, regenerative braking, shift quality, gear ratio selection and transmission losses. The results of this review indicate that optimizing a single KPI can lead to solutions that are highly favourable for one performance category but detrimental to another. The dynamic-energetic trade-off identified in this review therefore provides a useful basis for future MST selection and optimization for BEV applications.

Abbreviations

AC	Air Conditioning
AI	Artificial Intelligence
AMT	Automated Manual Transmission
BEV	Battery Electric Vehicle
CO_2	Carbon Dioxide
CUDC	China Urban Driving Cycle
CVT	Continuously Variable Transmission
DCT	Dual-Clutch Transmission
DSD	DSD Overlap Shift
EEA	European Environment Agency
EM	Electric Machine
EU	European Union
EV	Electric Vehicle
FTP-75	Federal Test Procedure-75
GB	Gearbox
GHG	Greenhouse Gas

HWFET Highway Fuel Economy Test
ICE Internal Combustion Engine
ICEV Internal Combustion Engine Vehicle
I-AMT Inverse Automated Manual Transmission
IVT Infinitely Variable Transmission
KPI Key Performance Indicator
MGT Magnetic Gear Transmission
MT Manual Transmission
MST Multi-Speed Transmission
NEDC New European Driving Cycle
PC Passenger Car
PAT Planetary Automatic Transmission
PMSM Permanent Magnet Synchronous Motor
PWT Powertrain
RPM Revolutions Per Minute
SAE Society of Automotive Engineers
SOC State of Charge
SST Single-Speed Transmission
TCU Transmission Control Unit
UDDS Urban Dynamometer Driving Schedule
US06 US06 Driving Cycle
WAM With Additional Measures
WEM With Existing Measures
WLTC Worldwide Harmonized Light-Duty Vehicles Test Cycle

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