Battery-Electric Powertrain Design Analysis for an Efficient Passenger Vehicle

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Abstract—In this paper, different battery-electric powertrain designs have been investigated for a passenger hatchback vehicle. Specifically, the energetic effect of technological and topological choices in the powertrain design is analyzed. Thereto, a feasible set of battery-electric powertrain configurations, varying in topology, machine technology, and transmission architecture is considered. For each configuration, the electric machine size(s), gear ratio value(s), and controls are jointly optimized in an integrated (bi-level) fashion using the particle swarm optimization (PSO) algorithm. The results showcase that the combined component and topological choices can lead to a maximum reduction of 17.61% in the vehicle’s energy consumption and can significantly influence the total electric machine sizing in the powertrain. Furthermore, it has been shown that the energy consumption of the vehicle is lowered by: (i) using two axles (AWD) instead of one axle (RWD) to drive the vehicle (-12.0%); (ii) choosing a distributed drive system, over a central drive system (-6.6%); followed by (iii) using permanent-magnet-synchronous type machine(s) instead of asynchronous induction machines (-3.0%); and, (iv) using multiple electric machines in single-axled topologies (-0.80%). The smallest energetic impact was observed for (v) using two-speed instead of single-speed gear box(es) (-0.38%); however, it resulted in the largest reduction in the (total) electric machine size(s) (-10.3%).

I. INTRODUCTION

More recently new passenger car models equipped with battery-electric powertrains are being introduced by original equipment manufacturers (OEMs) [1]. Among these battery-electric vehicles (BEVs) announced and already on the market, there exist a large variation in the powertrain design: the drivetrain architecture, i.e. locations of the electric machine(s) (front-wheel (FWD), rear-wheel (RWD) or all-wheel driven (AWD)), type of electric machine(s) (e.g. permanent-magnet synchronous (PSM) or asynchronous induction machine (AIM)) and how they are connected via transmission components to the driven wheels (topology). Given this large variation in powertrain designs, the main question that arises is what is the optimal powertrain configuration for such a BEV is, and more fundamentally, what underlying influence the possible design choices have on the vehicle’s key performance indicators (KPIs), like the vehicle’s energy consumption.

Related literature: In the context of BEVs, the powertrain design problem and the influence of design choices therein has previously been addressed in literature. Common practice herein is to subdivide the design problem into four different layers: (i) drivetrain topology, (ii) component technology, (iii) component sizing, and (iv) powertrain control. This is also referred to as a system-level design (SLD) [2]. Most powertrain design research has only focused on one or two design levels at maximum. Mainly addressing, for a fixed topology, the component specifications (e.g. optimal sizing and technology), investigating the effect of the transmission technology [3]–[8], the type of electric machine technology [9], [10], or both at the same time [11]. Other research investigates the effect of different topology designs (e.g. optimal sizing per layout) [12]–[16], in which the technology of the components are kept fixed most of the time. To the knowledge of the author, there is no present work that investigates the powertrain design problem for a BEV addressing both the topology layout, and transmission and machine technology simultaneously.

Statement of contributions: Against this backdrop, this work investigates both the effect of technological and topological design choices in a battery-electric powertrain for a passenger vehicle. In particular, it focuses on understanding the influence of these design choices with respect to the vehicle’s energy consumption given its performance constraints. The main underlying research question is centered on how to optimally design the electric powertrain with respect to the selection of the topology, machine technology and gearbox type. Thereto, an efficient passenger vehicle, i.e., with the BMW i3 94 Ah characteristics (cf. Table I), is selected to simulate the effect of the design choices on a vehicle currently on the market.

To investigate the different powertrain options, a modular backwards-facing quasi-static simulation model of the BEV powertrain has been developed, schematically represented in a forward fashion by the overview in Fig. 1. The model includes the longitudinal vehicle dynamics, electric machine(s), gearbox/final drive transmission unit(s), a battery pack and a control system (power-split and gear shifting). In addition,
a regenerative braking strategy is developed to model the behavior of a braking controller. Using the developed simulation model, a variety of powertrain configurations have been studied for the selected hatchback vehicle.

The powertrain design considerations that are investigated through this study are related to: (1) the electric machine technology (PSM or AIM), (2) the gearbox type (single-/twin-speed), (3) the drive system type (central or distributed drive), (4) the number of electric machines (1, 2 or 4), and (5) the drivetrain architecture (for RWD and AWD vehicles). The first two design choices refer to the component technology used whereas the other three influence the topology layout of the resulting powertrain design. For each powertrain configuration, the electric machine size(s), gear ratio(s), and controls are jointly optimized to be able to compare the energetic influence of each design choice and to extract the best configuration.

Organization: The remainder of this paper is organized as follows. First, in Section II, the electric powertrain system design problem and its mathematical formulation are presented. Secondly, in Section III, the model of the BEV is derived, including all the component modules of its powertrain system. Next, the optimization results for the different topology configurations are analyzed and discussed in Section IV. The work is concluded by Section V in which its most important findings are summarized.

II. E-POWERTRAIN SYSTEM DESIGN

In this section, the electric powertrain design problem is described in more detail. Thereto, the different topology designs that are considered here are presented and a mathematical formulation of the battery energy minimization problem, jointly optimizing the electric machine size(s), the transmission units’ gear ratio(s) and control(s) subjected to the component and vehicle-specific performance constraints, is given. The set of powertrain topologies will be discussed first.

A. Topology design

Topology is defined as the layout of the components in a powertrain, how they are connected to each other and to the wheels of the vehicle. Depending on the set of components used, a large number of different topology design are possible. In this work, topologies are considered that are constructed with up to two differently sized electric machines EMₖ, with machine index k, attached to one gearbox GBₗ per driven axle l, and the choice whether (or not) a final drive transmission FD is used to connect to the pair of wheel(s) W. The powertrain system is powered by a single battery pack BAT via integrated power electronics (not modeled separately), being the only source of (electrical) energy supply. The resulting set of different topology designs is given in Fig. 2. Here, the topologies can be categorized column-wise into central (C) and distributed (D) drive systems. Furthermore, the topologies can be classified row-wise into single-axle (RWD) and double-axle driven (AWD) architectures. Powertrain configurations, denoted by Tₖₗᵢⱼ ( ), are considered with either having single-speed or two-speed gearbox(es), which are connected to either PSM or AIM machines. The subscript

\[ i \in \{C1, \ldots, C3, D1, \ldots, D3\} \] indicates the topology index, \( j \in \{1, 2\} \) the number of gears in the (discrete) gearbox(es) and \( m \in \{PSM, AIM\} \) the machine technology used in the topology configuration. Given these topological and technological design variations, a set of 36 different functional powertrain configurations T could be created from the set of 9 possible topology layouts. To be able to compare the optimal design of all 36 configurations, a powertrain optimization problem is mathematically formulated next.

B. Mathematical problem formulation: bi-level optimization

The powertrain design problem focuses on the minimization of the supplied battery energy \( \Delta E_{bat} \) (kWh), obtained by integration of the supplied battery power \( P_b(t) \) (W) over a representative drive cycle (e.g. WLTP). To get an optimal system design for each powertrain configuration \( T_{i,j}^{m} \in T \), the physical components and the control algorithm are (co-)designed in an integrated manner [2]. Therefore, the combined powertrain design problem is formulated as a bi-level optimization, where both the powertrain design parameters \( x_p \) and time-dependent control variables \( x_c(t) \) are optimized. Due the unidirectional coupling between the plant and control variables, the problem is solved using a nested optimization approach [2], with two individual optimization loops. In the outer loop, expressed by the high-level problem \( \mathcal{P}_{h} \), the optimal sizing parameters \( x_p^* \) are found by the given optimal control signals \( x_c^*(t) \), as the result of the lower-level problem \( \mathcal{P}_{l} \) in the inner loop.
Component sizing (high-level problem): The high-level problem, as expressed in Eq. (1), aims to find the optimal sizing parameters \( x_p = \{ P_m, R_x \} \), related to the EM size(s), \( P_m = \{ P_{m1}, P_{m2} \} \), and the set of gear ratio value(s), \( r_g = \{ r_{g1}, r_{g2} \} \) of the gearbox(es) \( R_x = \{ r_{x1}, r_{x2} \} \) present in the specific topology configuration \( T_{i,j}^m \), by solving:

\[
P_h : \min_{x_p} \Delta E_o(x_p | x_r, x_v, \Lambda(t), T_{i,j}^m) \\
\text{s.t. } g_p(x_p) \leq 0, \quad h_p(x_p) = 0,
\]

(1)

given the vehicle parameters \( x_v \), and the drive cycle information \( \Lambda(t) \), subjected to the vehicle performance and component requirements, which are captured by the inequality \( g_p \) and equality constraints \( h_p \), as given below.

- \( g_{p1} \): top speed \( v_{\text{max}} \) (km/h)
- \( g_{p2} \): kinetic top speed \( \omega_{\text{max}} \) (v_{\text{max}}) \( r_\text{x} \) (rpm)
- \( g_{p3} \): acceleration time for \( 0-100 \) (km/h) \( t_{\text{acc}} \) (s)
- \( g_{p4} \): low-speed gradeability \( q_0 \geq 30 \) (%)
- \( g_{p5} \): mutual gear sizing \( r_{\text{g1}}(1) \geq r_{\text{g1}}(2) \) (-)

\( h_{p1} \): battery energy content \( E_b \) (kWh)
\( h_{p2} \): final drive ratio \( \tau_{\text{f1}} = 1 \) (-)

The performance constraints are related to the minimum top speed \( (g_{p1}) \), the maximum achievable top speed without over-speeding the E-machine(s) \( (g_{p2}) \), the acceleration time \( (g_{p3}) \) and the minimum gradeability from stand-still \( (g_{p4}) \). The component constraints are related to the gear step design (always speed reducing during up-shifting), and the battery size \( (h_{p1}) \) and the final drive ratio(s) \( (h_{p2}) \) that are kept at a fixed value. The latter assuming that the combined gearbox and final drive ratio is optimized in case of a central drive topology. In order to solve for the lower-level optimal control problem, as expressed in Eq. (2), the given functional powertrain configuration, feasible machine size(s) and gear ratio value(s), vehicle parameters and drive cycle data are provided as input from the high-level problem.

Control optimization (low-level problem): The low-level problem aims to find for each feasible combination of values \( x_p \), the optimal control signals \( x_c(t) \) present in the powertrain configuration \( T_{i,j}^m \), by solving:

\[
P_l : \min_{x_c(t)} \Delta E_o(x_c(t) | x_p, x_v, \Lambda(t), T_{i,j}^m) \\
\text{s.t. } g_c(x_c(t) | x_p) \leq 0, \quad \xi(t) \in [\xi, \bar{\xi}],
\]

(2)

where \( \xi(t)(-) \) refers to the system’s battery state-of-charge (SoC) and \( g_c \) is the set of inequality constraints that are included in the simulation model, given by:

- \( g_{c1} \): max. electric machine torque \( \leq \tau_{\text{m1}} \) (Nm)
- \( g_{c2} \): min. electric machine torque \( \leq \tau_{\text{m1}} \) (Nm)
- \( g_{c3} \): max. electric machine speed \( \leq \omega_{\text{m1}} \) (rad/s)
- \( g_{c4} \): max. battery charge current \( \leq I_b \) (A)
- \( g_{c5} \): max. battery discharge current \( \leq I_b \) (A)

related to the physical limitations of the components in the powertrain, such as the maximum \( (g_{c1}) \) and minimum \( (g_{c2}) \) torque, and maximum speed \( (g_{c3}) \) of each electric machine \( E_{\text{M1}} \); and, the maximum charge \( (g_{c4}) \) and discharge \( (g_{c5}) \) current of the battery pack BAT.

The control variables \( x_c(t) = \{ \gamma(t), \phi(t) \} \), in case of multiple E-machines and 2-spd gearbox(es), are the gear position \( \gamma(t) = \{ \gamma(l) \} \) of each gearbox \( l \in \{1,2\} \) and the torque split \( \phi(t) \in [0,1] \) between the electric machines:

\[
\phi(t) = \frac{\tau_{\text{m1}}(t)}{\tau_{\text{m1}}(t) + \tau_{\text{m2}}(t)},
\]

(3)

where \( \tau_{\text{m1}} \) (Nm) is the torque generated by \( E_{\text{M1}} \) and \( \tau_{\text{m2}} \) (Nm) is the torque generated by \( E_{\text{M2}} \) (Nm). The presence of the actual control variables depends on the specific topology.

III. VEHICLE MODELING

The backward-facing vehicle simulation model, as schematically shown in Fig. 1, is constructed in a modular way and incorporates the different topology layouts, component technologies as power-based building blocks. Here, each model block represents a storage, conversion or transmission loss element that is present in the powertrain. Furthermore, the power flows are indicated by arrows from component input to output (load) in a forward fashion, yet the power demands are simulated in a backwards fashion (up-stream). Below the assumptions and each of the (quasi-static) powertrain component models used in the BEV simulation model is discussed in more detail.

A. Vehicle road-load

Given a speed profile \( v(t) \) (m/s) without road inclination and the vehicle parameters in Table I, the traction force, \( F_t(t) \) (N), required to propel the vehicle in longitudinal direction is [22]:

\[
F_t(t) = \frac{1}{2} \rho_a A_t c_d v^2(t) + c_r m_v g + \lambda \frac{dv(t)}{dt},
\]

(4)

where \( \rho_a = 1.225 \) (kg/m³) is the air density, \( A_t \) (m²) is the frontal area, \( c_d(-) \) is the air drag coefficient, \( c_r(-) \) is the tire rolling friction coefficient, \( m_v \) (kg) is the total mass of the vehicle, \( g = 9.81 \) (m/s²) is the gravitational acceleration and \( \lambda = 1.05 (-) \) [23] is the rotational inertia factor that accounts for the mass of the rotating parts. The corresponding angular speed \( \omega_w \) (rad/s) and torque demand \( \tau_w \) (Nm) at the (four) wheels of the vehicle are given by:

\[
\omega_w(t) = \frac{v(t)}{r_w},
\]

(5)

\[
\tau_w(t) = F_t(t) r_w,
\]

(6)

where \( r_w \) (m) is radius of an individual wheel. Dependent on the drivetrain configuration of the powertrain topology, the required (wheel) torque \( \tau_w \) (Nm) is distributed over the wheel pair of the front \( (\tau_{w1}) \) and/or rear axle \( (\tau_{w2}) \) of the vehicle:

\[
\tau_w(t) = \tau_{w1}(t) + \tau_{w2}(t),
\]

(7)
TABLE I

VEHICLE TECHNICAL SPECIFICATIONS[17]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>VEHICLE BODY</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Curb weight (EU)</td>
<td>$m_c$</td>
<td>1320</td>
<td>(kg)</td>
</tr>
<tr>
<td>Frontal area</td>
<td>$A_t$</td>
<td>2.38</td>
<td>(m²)</td>
</tr>
<tr>
<td>Air drag coefficient</td>
<td>$c_d$</td>
<td>0.29</td>
<td>(-)</td>
</tr>
<tr>
<td>Wheel base</td>
<td>$l$</td>
<td>2.57</td>
<td>(m)</td>
</tr>
<tr>
<td>Center of gravity height</td>
<td>$h_g$</td>
<td>0.47</td>
<td>(m)</td>
</tr>
<tr>
<td>Static weight distribution (FR)</td>
<td>-</td>
<td>47/53</td>
<td>(%)</td>
</tr>
<tr>
<td>POWERTRAIN</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Drivetrain configuration</td>
<td>-</td>
<td>RWD</td>
<td>(-)</td>
</tr>
<tr>
<td>Electric machine technology</td>
<td>-</td>
<td>PSM</td>
<td>(-)</td>
</tr>
<tr>
<td>Peak motor power</td>
<td>$P_m$</td>
<td>125</td>
<td>(kW)</td>
</tr>
<tr>
<td>Rated motor speed</td>
<td>$\omega_{\text{in,r}}$</td>
<td>4800</td>
<td>(rpm)</td>
</tr>
<tr>
<td>Maximum motor torque</td>
<td>$\tau_m$</td>
<td>250</td>
<td>(Nm)</td>
</tr>
<tr>
<td>Maximum motor speed</td>
<td>$\omega_m$</td>
<td>11400</td>
<td>(rpm)</td>
</tr>
<tr>
<td>TRANSMISSION</td>
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</tr>
<tr>
<td>Tyre model</td>
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<td>Ecopia EP600</td>
<td>(-)</td>
</tr>
<tr>
<td>Tyre size</td>
<td>-</td>
<td>155/70-R19</td>
<td>(-)</td>
</tr>
<tr>
<td>Rolling resistance coefficient[19]</td>
<td>$c_r$</td>
<td>6.4</td>
<td>(kg/mn)</td>
</tr>
<tr>
<td>Wheel radius</td>
<td>$r_w$</td>
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<td>(m)</td>
</tr>
<tr>
<td>Gear ratio</td>
<td>$r_g$</td>
<td>9.665</td>
<td>(-)</td>
</tr>
<tr>
<td>BATTERY</td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Battery pack configuration</td>
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<td>96 l</td>
<td>(-)</td>
</tr>
<tr>
<td>Nominal battery pack voltage</td>
<td>$U_{boc}$</td>
<td>353</td>
<td>(V)</td>
</tr>
<tr>
<td>Nominal battery pack capacity</td>
<td>$Q_0$</td>
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<td>(V)</td>
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<tr>
<td>Battery pack energy content</td>
<td>$E_{boc}$</td>
<td>33.2/27.2</td>
<td>(KWh)</td>
</tr>
<tr>
<td>PERFORMANCE</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Top speed</td>
<td>$v_{\text{max}}$</td>
<td>150</td>
<td>(km/h)</td>
</tr>
<tr>
<td>Acceleration (0-100 km/h)</td>
<td>$t_{\text{acc}}$</td>
<td>7.3</td>
<td>(-)</td>
</tr>
<tr>
<td>Energy consumption (NEDC)</td>
<td>$E_b$</td>
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<td>(KWh/100km)</td>
</tr>
<tr>
<td>Electric driving range (NEDC)</td>
<td>$D_0$</td>
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<td>(km)</td>
</tr>
<tr>
<td>COMPONENT MASSES</td>
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<td></td>
</tr>
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<td>Electric machine mass[17]</td>
<td>$m_{m,0}$</td>
<td>42</td>
<td>(kg)</td>
</tr>
<tr>
<td>Motor inverter mass[20]</td>
<td>$m_{i,0}$</td>
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<td>(kg)</td>
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<tr>
<td>Gearbox/final drive mass</td>
<td>$m_{F,d}$</td>
<td>21</td>
<td>(kg)</td>
</tr>
<tr>
<td>Battery pack mass[21]</td>
<td>$m_{b,0}$</td>
<td>256</td>
<td>(kg)</td>
</tr>
</tbody>
</table>

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B. Braking strategy

The (maximum) amount of regenerative braking that the electric motor(s) can apply without destabilizing the vehicle, is proportional to the normal load(s) acting on the vehicle and is limited by the tire-road adhesion coefficient $\mu = 0.8$ (-) [23]. Dependent on the drivetrain configuration of the powertrain topology, the regenerative braking force $F_{\text{br},r}(N)$ becomes:

$$F_{\text{br},r}(t) = \begin{cases} (1 - \beta(t)) F_{\text{br}}(t), & \text{if RWD,} \\ F_{\text{br}}(t), & \text{if AWD,} \end{cases}$$

where $F_{\text{br}}(t) = (F(t) < 0)$ (N) is the overall braking force. In case of AWD, maximum regenerative braking is possible within the limitation of the tyre-road adhesion. For RWD layouts, the actual regenerative braking that can be applied is determined by the brake fraction $0 \leq \beta \leq 1$ (-), being the ratio of the (maximum) braking force on the front axles, $F_{\text{br},r}(N)$, to the total braking force $F_{\text{br}} = \mu F_{\text{a}}$ of the vehicle [23]:

$$\beta(t) = \frac{F_{\text{br},r}(t)}{F_{\text{br}}} = \frac{F_{\text{br}}(t)}{F_{\text{a}}},$$

where $F_{\text{a}} = m_{CG}g$ represents the (total) vehicle weight and $F_{\text{a},l}(t)$ accounts for the load transfer from the rear axle to the front axle while braking. The latter is dependent on the vehicle deceleration rate, $a_v(t) < 0$ (m/s²) and the center of mass (CoM) of the vehicle body. The total required (net) resultant force at the wheels, $F_{\text{r}}(N)$, becomes then:

$$F_{\text{r}}(t) = F_{\text{l}}(t) + F_{\text{br},r}(t),$$

in which $F_{\text{br},r}(N)$ accounts for the additional force needed from the mechanical brakes to realize the required deceleration rate during braking phase(s).

C. Final drive transmission

The presence of a final drive (FD) in the powertrain topology is modeled as a bevel gear differential with a fixed efficiency of $\eta_{\text{fd}} = 0.93$ (-) [24]. In that case, the angular speed $\omega_{\text{fd}}$ (rad/s) and torque $\tau_{\text{fd}}$ (Nm) at the input of the final drive unit at axle $l$ follows from:

$$\begin{cases} \omega_{\text{fd}}(t) = \omega_{\text{FD}}(t) \tau_{\text{fd}}(t), \\ \tau_{\text{fd}}(t) = \tau_{\text{sg}}(t) \eta_{\text{fd}} \text{sg}(\tau_{\text{sg}}(t)) \end{cases}$$

If no final drive is present, the wheels are directly attached to the gearbox and the axle wheel torque is divided without differential losses over the two individual driven wheels at each side (left/right) of the driven axle(s). Hence, the torque and angular speed relations of Eq. (11) become:

$$\begin{cases} \omega_{\text{FD}}(t) = \omega_{\text{FD}}(t), \\ \tau_{\text{fd}}(t) = \tau_{\text{sg}}(t) \eta_{\text{fd}} \text{sg}(\tau_{\text{sg}}(t)) \end{cases}$$

D. Gearbox transmission

The gearbox transmission (GB), being either a single-speed (1-spd) or a twin-speed (2-spd) discrete gearbox, is modeled using a fixed gear-dependent efficiency $\eta_{\text{gp}}(\gamma(t))$. The specific gear efficiency is determined by the number of gear-stages that are necessary to realize the gear ratio [25]. It is assumed that one gear-stage, i.e., one pair of gears, has a constant efficiency of $\eta_{\text{gp}} = 0.985$ (-) and can realize a maximum gear ratio of four [25]. The efficiency per (selected) gear is then given by:

$$\eta_{\text{gp}}(\gamma(t)) = \begin{cases} \eta_{\text{gp}}^0 & \text{if } \tau_{\text{gp}}(\gamma(t)) \leq 4, \\ \eta_{\text{gp}}^0 \left( \frac{\tau_{\text{gp}}(\gamma(t))}{\tau_{\text{gp}}} \right)^{\gamma_{\text{gp}}(\tau_{\text{gp}}(\gamma(t)))} & \text{if } 4 < \frac{\tau_{\text{gp}}(\gamma(t))}{\tau_{\text{gp}}} \leq 16, \end{cases}$$

where $\gamma_{\text{gp}}(\gamma(t))$ represents the gear of the gearbox unit at axle $l$ for which the efficiency is calculated. To account for the presence of a final drive, the value of $\tau_{\text{sg}}$ equals one for a distributed topology and seven for a central drive topology. The angular speed $\omega_{\text{sg}}$ (rad/s) and torque $\tau_{\text{sg}}$ (Nm) at the output of gearbox unit $l$ are calculated as:

$$\begin{cases} \omega_{\text{sg}}(t) = \omega_{\text{sg}}(t) \tau_{\text{sg}}(\gamma(t)), \\ \tau_{\text{sg}}(t) = \frac{\tau_{\text{sg}}(\gamma(t))}{\tau_{\text{sg}}(\gamma(t))} \eta_{\text{sg}}(\gamma(t)) \text{sg}(\tau_{\text{sg}}(\gamma(t))) \end{cases}$$

where the corresponding gear ratio and efficiency are dependent on the gear position $\gamma(t)$ at any given time $t$. 
Fig. 3. MOTOR-CAD® generated efficiency map (M_m) of the permanent-magnet synchronous machine (PSM) (a) and asynchronous induction machine (AIM) (b) in motoring/generator mode, designed for the same peak performance targets: maximum torque \( \tau_{m} = 350 \, \text{Nm} \), peak power \( P_{m} = 150 \, \text{kW} \) and maximum speed \( \omega_{m} = 12000 \, \text{rpm} \) [26]. These base maps are used to linearly scale the electric machine efficiency (\( \eta_{m} \)) in the torque direction with the original (peak) power (\( P_{m,0} \)) of the machines.

E. Electric machine

The electric machine (EM) that is connected to the gearbox is modeled with a lookup map of the motor/generator efficiency \( M_{m} \), as depicted in Fig. 3 for both the PSM and AIM technology. Dependent on the machines’ torque \( \tau_{m} \) (Nm) and speed \( \omega_{m} \) (rad/s) operating point, a value for the efficiency \( \eta_{m} \) (-) is obtained by interpolating this map:

\[
\eta_{m} = M_{m} \left( \omega_{m}, \frac{\tau_{m}}{s_{m}} \right). \tag{16}
\]

To account for the performance of the differently sized EMs, the parameter \( s_{m} \) (-) is used to scale the original efficiency map(s) linearly along the torque axis [22], according to:

\[
s_{m} = \frac{P_{m}}{P_{m,0}}, \tag{17}
\]

where \( P_{m} \) (W) is the peak mechanical power of the resized E-machine and \( P_{m,0} \) (W) is the peak mechanical power of the original E-machine. Using the obtained machine efficiency, the EM electrical input power \( P_{el,m} \) (W) is calculated:

\[
P_{el,m}(t) = \omega_{m}(t) \tau_{m}(t) \eta_{m}. \tag{18}
\]

Similarly, the maximum torque \( \tau_{m} \) (and minimum torque \( \tau_{m} \)) of the (re-)sized EM is scaled accordingly:

\[
\tau_{m} = s_{m} \tau_{m}. \tag{19}
\]

To account for the motor inverter losses, an additional (fixed) efficiency of \( \eta_{f} = 0.95 \) (-) is assumed.

F. Battery

The battery pack of the vehicle is modeled with a basic equivalent circuit model [22]. Data from a Samsung SDI 94 Ah [27] Li-on cell is used to simulate the battery pack that is originally present in the BMW i3 94 Ah vehicle. In the battery model considered, the open-circuit voltage \( U_{oc} \) (V) and internal resistance \( R_{b} \) (\( \Omega \)) are determined by the configuration of the battery, i.e., the number of cells in series \( n_{se} = 96 \) (-), and the number of parallel branches, \( n_{pa} = 1 \) (-), in the pack (cf. Table I):

\[
U_{oc}(t) = n_{se} U_{oc,cl}(\xi(t)), \tag{20}
\]

\[
R_{b}(t) = \frac{n_{se}}{n_{pa}} R_{cl}(\xi(t)), \tag{21}
\]

where \( U_{oc,cl}(\xi(t)) \) (V) and \( R_{cl}(\xi(t)) \) (\( \Omega \)) are the open-circuit and the internal resistance of a single cell, respectively, modeled as function of the state-of-charge \( \xi(t) \). The intermediate SoC of the battery is calculated using Coulomb counting:

\[
\xi(t) = \xi(t_{0}) - \int_{t_{0}}^{t} \frac{I_{b}(t)}{Q_{b}} dt, \tag{22}
\]

where \( Q_{b} \) (Ah) is the nominal battery (pack) capacity, following from the nominal cell capacity \( Q_{cl} \) (Ah), as:

\[
Q_{b} = n_{pa} Q_{cl}. \tag{23}
\]

The internal battery current \( I_{b} \) (A) is determined by applying Kirchhoff’s voltage law to the equivalent battery circuit:

\[
I_{b}(t) = \eta_{co} \frac{U_{oc}(t) - \sqrt{U_{oc}^{2}(t) - 4R_{b}P_{b}(t)}}{2R_{b}}, \tag{24}
\]

where \( \eta_{co} = 0.98 \) (-) is the Coulombic efficiency that is incorporated when the battery is charging \( (I_{b}(t) < 0) \). The power at the battery terminals \( P_{b} \) (W) is computed with:

\[
P_{b} = n_{m} \sum_{k=1}^{n_{m}} P_{mk} + P_{aux}, \tag{25}
\]

where \( n_{m} \) (-) is the total number of electrical machines in the specific topology. A fixed average auxiliary power \( P_{aux} = 800 \) (W) is added, that covers the devices that need to be activated during the homologation test [28]. Subsequently, the internal power of the battery \( P_{b} \) (W) is:

\[
P_{b}(t) = I_{b}(t) U_{oc}(t). \tag{26}
\]
G. Mass

The change in the total mass of the vehicle, $m_v$ (kg), with its scaled powertrain components is calculated by:

$$ m_v = m_0 + m_{pt}, \quad (27) $$

where $m_0$ (kg) is the base mass of the vehicle, i.e. without its original powertrain components (cf. Table I). The (new) powertrain mass, $m_{pt}$ (kg), is then obtained by:

$$ m_{pt} = m_{b,0} + \sum_{k=1}^{n_m} (m_{m,k} + m_{i,k}) + \sum_{l=1}^{n_{gb}} m_{gb,l}, \quad (28) $$

where $n_{gb}$ is the number of gearboxes present in the powertrain topology. The mass models used for the different components are described in Table II, with $m_{m,k} \in \{m_{PSM}, m_{AIM}\}$.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motor inverter</td>
<td>$m_{i,k}$</td>
<td>0.1 ($\frac{P_m}{1000}$)</td>
<td>(kg)</td>
</tr>
<tr>
<td>Gearbox transmission</td>
<td>$m_{gb}$</td>
<td>1.723 ($\frac{P_m}{1000}$)</td>
<td>(kg)</td>
</tr>
<tr>
<td>PSM machine</td>
<td>$m_{PSM}$</td>
<td>0.22 ($\frac{P_m}{1000}$)</td>
<td>(kg)</td>
</tr>
<tr>
<td>AIM machine</td>
<td>$m_{AIM}$</td>
<td>0.31 ($\frac{P_m}{1000}$)</td>
<td>(kg)</td>
</tr>
</tbody>
</table>

H. Model validation

The simulation model is validated for the BMW i3 hatchback vehicle using the parameters given in Table I. Therefore, the energy usage of the original (base) powertrain configuration ($T_{P,PSM}$) with the original component sizes, PSM machine and single-speed transmission was computed over the New European Driving Cycle (NEDC) and compared to the available value from the OEM: 13.1 (kWh/100km). The simulated energy consumption was 13.2 (kWh/100km). The difference between the simulated and published NEDC values of $\epsilon = 1.5$ (%) is found to be acceptable and, therefore, the vehicle model is assumed to be representative and sufficient accurate for the design optimization study as discussed next.

IV. OPTIMIZATION RESULTS

This section describes the optimization results for the hatchback vehicle with the different optimized powertrain configurations, varying in topology choice, machine technology and transmission architecture. The problem is solved subjected to the vehicle performance constraints originated from the OEM specifications (eq. Table I) to analyze the effect of the considered powertrain design choices within the design space of the given vehicle type. As input for the optimization, the representative Worldwide Harmonised Light Vehicle Test Procedure (WLTP) is taken. The energy consumption for the base vehicle with its original powertrain over the WLTP cycle is found to be $E_{v,0} = 15.05$ (kWh/100km), and is used as a reference for comparison in the results next.

A. Numerical implementation and solver

The optimization problems $P_b$ and $P_t$, together with the BEV model are numerically implemented in MATLAB®. A solution to the higher-level problem is found using particle swarm optimization (PSO) [29]. The lower-level control problem has been included in the simulation model using a local minimization method. Here, the control vectors $\{\gamma(t), \phi(t)\}$ are gridded (vectorized) and for each time step $t$ the values for the control variables are selected that minimize the objective function change for that specific time step. For all powertrain configurations, the same PSO algorithm settings were used.

B. Optimal design results

Table III shows the optimization results of the 36 powertrain configurations considered. For each optimized configuration, it states the optimal powertrain mass $m_{pt}^*$ (kg), overall transmission efficiency $\eta_{gb}^*$ (-), overall machine efficiency $\eta_{m}^*$ (-), the total (peak) motor power $P_m^*$ (kW) of the EMs present in the topology, and the optimal vehicle energy consumption $E_v^*$ (kWh/100km). It also includes the relative energy savings $\delta E_v^*$ (%) when changing from a 1-spd to a 2-spd gearbox architecture, and the relative difference in energy consumption $\delta E_{v,0}$ (%) with respect to the previous calculated reference. The data is sorted row-wise by the different topology layouts and gearbox type, where the first column under each parameter contains the values for the PSM technology and the second column the values for the AIM technology.

The topology with the lowest energy consumption is $T_{D,3}$. The optimal configuration for $T_{D,3}$ is equipped with PSM machines, connected to a 2-spd gearbox (i.e. $T_{D,PSM}^{2,2}$), and resulted in an overall lowest energy consumption of $E_v^* = 12.40$ (kWh/100km), being $\delta E_{v,0} = -17.61$ (%) less than the original powertrain configuration. Moreover, the total machine size for this optimal topology configuration, is reduced by $-19.9$ (%). The topology $T_{C,1}$, as originally used in the base vehicle, with a 1-spd gearbox and AIM machine (i.e. $T_{AIM}^{1,1}$) resulted in having the highest energy consumption overall: $E_v^* = 15.44$ (kWh/100km). The relative difference between the optimized powertrain topologies with the highest ($T_{C,1}$) and lowest energy consumption ($T_{D,3}$) led to a variation of $-17.2$ (%) (PSM) and $-16.8$ (%) (AIM) in the absolute energy consumption of the vehicle. This variation is only slightly increased by respectively $-0.3$ (%) (PSM) and $-0.2$ (%) (AIM) when including the gearbox choice.

Focusing on the local effect of each of the powertrain design choices (1) – (5) as presented in Section I, the following observations can be done for the BMW i3 hatchback vehicle without compromising its performance constraints.

1) Machine technology: Using PSM machines gives for all topologies on average a decrease in the vehicle energy consumption of $\delta E_v = -3.0$ (%) (similar for both the 1-spd and 2-spd) compared to the AIM technology. In line with the (overall) efficiency of the PSM machine, which is higher in the same order of magnitude compared to the AIM: $\delta \eta_m = +3.4$ (%). The powertrain mass decreases for all topologies on average with $\delta n_{mb} = -2.8$ (%) in case PSMs with a higher power density (kW/kg) are used.
2) Gearbox architecture: Re-allocating the machine’s operating points towards higher efficiency regions using 2-spd instead of 1-spd gearbox architecture(s) gives, for both the PSM and AIM, a relative average decrease in energy consumption of −0.38% and −0.16%, respectively. The total machine efficiency for both machine technologies increases with 0.62% on average. However, this gain in efficiency is counteracted by the increased powertrain mass caused by the heavier twin-speed gearbox. The relative mass increase is on average +3.4% for the PSM and +3.8% for the AIM, explaining the lower energy consumption gain for AIMS. The gears in a two-speed gearbox can be sized such that a larger (lower) gear is realized. With this larger gear, less torque is required from the electric machines, leading to an average decrease in the total machine size of −10.3% and −8.36% for the PSM and AIM machine(s), respectively. The largest benefits were obtained for topology configurations having one differently sized EM and a 1-spd transmission. Furthermore, for distributed topologies, using a two-speeds was more beneficial than for central topologies. The effect of the two-speed gearbox was less in case multiple EMs were present; for both single and double-axle distributed topologies.

3) Drive system type: Going from a central drive (C) to a distributed drive system (D) gives an average energy saving (for both the PSM and AIM technology) of −6.6%. This relative difference is maximum (−8.5%) for the double-axle (AWD) topology (T_{D1} → T_{D3}). The powertrain mass for distributed driven topologies was on average +6.6% higher than their central counterparts because of the double gearboxes that are required. Nevertheless, this extra mass is counteracted by the absence of the final drive differential losses, increasing the total transmission efficiency in the powertrain. The (total) electric machine size(s) was lower for the distributed drive powertrain systems, compared to the central drive ones.

4) Number of electric machines: Adding an additional, differently sized EM in the topology lowers the absolute energy consumption for all topology variations. The extra EM enables a torque split over the electric machines in such a way that they operate in more efficient operating regions for a particular torque demand, hence increasing the total machine efficiency in the topology. The relative change in the absolute energy when having multiple EMs, is similar for both single-axle central and distributed topologies, with average values equal to δE^\text{m}_t = −0.80% (PSM) and δE^\text{m}_t = −0.69% (AIM) when equipped with 1-spd gearbox(es). The effect is less for configurations having 2-spd gearbox(es), that also influence the location of the working points in the EM's efficiency map. For all dual-motor configurations, one EM was larger sized than the second EM present.

5) Drivetrain configuration: Double-axle driven topologies have the possibility of recuperating electric energy on both the front and rear wheels during regenerative braking, which substantially decreases the vehicle's energy consumption compared to single-axled driven variants. The additional components necessary increase the total powertrain mass, but cause the total machine efficiency to be higher. The machine efficiency that increases with the same order of magnitude for both EM types (+1.5%), together with more regenerative braking capability, results in a average decrease in the energy consumption of δE^\text{m}_t = −9.3% (for central driven topologies and δE^\text{m}_t = −12.0% (for distributed driven topologies, compared to T_{C1} and T_{D1}, respectively. For both drive system types, the total electric machine size decreases with respect to their single-axled equivalents when equipped with 2-spd gearboxes. This trend is not visible in case 1-spd are used.

C. Discussion

A few comments are in order. First of all, the set of optimal component parameters x^*_p and control variables x^*_c(t) for each powertrain configuration was subject to a pre-defined grid (e.g. control steps, component bounds) and the parameters chosen for the optimization algorithm. Additionally, the settings of the optimization algorithm (e.g. stopping criteria) were the same for all powertrain configurations, which could have an influence the optimal solution, especially for configurations with a higher number of design parameters. Although the PSO algorithm converged to an optimum, it did not guarantee finding the exact global solution. Hence, the optimality of the computed results, and the likely-hood of finding a global optimal solution with the PSO solver should be verified in extended research (e.g. by an exhaustive search routine). Based on that, the PSO parameter settings might need to be re-tuned.
V. CONCLUSION

This work investigates the effect of technological and topological design choices in the battery-electric powertrain of an efficient passenger vehicle. Thereto, a powertrain design optimization study, centered on the energy consumption of the vehicle over the WLTP driving cycle, is performed for a group of various powertrain configurations. In total, 36 different powertrain configurations are investigated, constructed from 9 topology layouts, for which two different machine technologies (PSM/AIM) and gearbox architectures (1-spd/2-spd) are considered. Results showed that, for the BMW i3 hatchback vehicle, a maximum decrease in the energy consumption of \(-17.37\%\) can be achieved when changing to a distributed, double-axled powertrain topology, while keeping the original PSM machine technology and 1-spd gearbox type. Among the topology configurations studied, also a variation of the total machine size of \(-19.9\%\) could be observed. From the separate study on the design consideration, the largest effect on the energy consumption on average was achieved by: (i) using two axles (AWD) instead of one axle (RWD) to drive the vehicle, for distributed drive systems \(-9.3\%\) and central drive systems \(-12.0\%\); (ii) choosing a distributed drive system, over a central drive system \(-6.6\%\) followed by (iii) using PSM type machine(s) instead of AIM \(-3.0\%\); and, (iv) using multiple electric machines instead of a single one in single-axled topologies, in case of the PSM \(-0.80\%\) and the AIM \(-0.69\%\). The smallest energetic impact was observed when (v) the powertrain was equipped with a two-speed instead of a single speed connected to the PSM \(-0.38\%\) or the AIM \(-0.16\%\), however, resulted in a decrease of the EM size by respectively \(-10.3\%\) (PSM) and \(-8.36\%\) (AIM).

REFERENCES