PARAMETRIC MODELLING OF WEIGHT AND VOLUME EFFECTS IN BATTERY ELECTRIC VEHICLES, WITH FOCUS ON THE GEARBOX

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ABSTRACT

The modeling of battery electric vehicles (BEVs) still represents a challenge for vehicle manufacturers. The installation of the new types of components needed for BEVs gives rise to uncertainties in the quantification of parameters like the vehicle's weight. Indeed, vehicle weight plays a key role, since it has a drastic effect on the vehicle's range, which is an important selling point for BEVs. Uncertainties in weight estimation create weight fluctuations during the early development phase and the need to resize components like the electric machine or battery. This in turn affects the components' volume and weight. However, such resizing can also lead to component collision and unfeasibility of the vehicle architecture. To solve this problem and to support concept engineers during the early development phase, an iterative approach is required that is capable of estimating weight and volume fluctuations in the relevant components. The approach should also consider the geometrical interdependencies of the components, to ensure that no collisions occur between them. Taking the gearbox as an example application, this paper presents a novel approach that satisfies these requirements.

Keywords: Battery electric vehicles, Parametric modeling, Design engineering, Product architecture, Optimisation

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1 INTRODUCTION

The number of battery electric vehicles (BEVs) on the road has risen in recent years as a result of the European restrictions on permitted CO2 emissions from passenger cars (The European parliament and the council of the European Union, 2019). Although BEVs allow manufacturers to comply with the European Union standards, there are currently no established BEV architectures and few predecessor vehicles that can serve as a starting point for the early development phase.

To develop a new BEV architecture, concept engineers need to assess several design parameters (Nicoletti *et al.*, 2020a). These design parameters include vehicle dimensions, required range, acceleration time and form the vehicle portfolio that is the basis for further architecture development. A poorly chosen vehicle portfolio can lead to an architecture unfeasibility. However, this unfeasibility may only be detected after weeks or even months of development and may require a restart of the architecture development from scratch.

Two parameters are particularly critical during the portfolio definition: electric range and vehicle curb weight (VCW). As it is an important selling point for BEVs, an electric range is selected that matches the targeted vehicle segment and customer group. Once the electric range requirements have been set and the VCW is known, it is possible to derive the vehicle consumption and the required battery capacity. However, at this point in the development, it is only possible to make a rough estimate of the VCW. Such an estimate is naturally error-prone, and further inaccuracies will occur in the quantification of vehicle consumption and battery capacity. Therefore, the vehicle architecture development necessitates an iterative process, in which the battery capacity is adjusted with each successive iteration loop.

Compared to diesel or gasoline, Li-Io batteries have a low gravimetric and volumetric energy density (Frieske *et al.*, 2018). This means that their weight and volume fluctuate with the capacity adjustments made in each iteration loop. Not only do battery weight fluctuations lead to further variations in the VCW in the form of secondary weight changes (SWCs) (Nicoletti *et al.*, 2020c), but capacity adjustments also trigger variations in battery volume, i.e. secondary volume changes (SVCs).

The battery is not the only component affected by SVCs and SWCs. The weight and volume of other components (such as the electric machine, gearbox, and wheels) are also dependent on the VCW. These components can also cause SVCs and SWCs in each iteration loop. Furthermore, because of geometrical interdependencies between the components, SVCs can lead to a component collision, rendering the vehicle architecture unfeasible. This raises the complexity of the yet-to-be-developed vehicle architecture, in which multiple components influence each other's weights and volumes.

A method capable of modeling SWCs and SVCs would support concept engineers while defining the vehicle portfolio and thus avoid the occurrence of an architecture unfeasibility in the further steps of the development. This paper presents an approach that satisfies this need.

In the following section, the state of the art is discussed and the research gap identified. Based on the research gap, an approach is presented, that models the SWCs and SVCs (Section 3). To exemplify the approach, we apply our model to the gearbox because it is, by virtue of its position, geometrically interdependent with the battery, and therefore impacts the permissible battery dimensions and maximum installable capacity. Subsequently, the results obtained with the developed approach are presented in Section 4. Finally, Section 5 provides the conclusion and outlook.

2 STATE OF THE ART

VCW and SWC modeling have previously been researched by (Fuchs and Lienkamp, 2013), who proposed an approach for estimating SWCs in BEVs. However, the data used for modeling the vehicle components is outdated and refers to a limited number of BEVs. Furthermore, Fuchs is unable either to derive a vehicle architecture or to test feasibility, since he does not model the volume of the components.

(Angerer, 2020) extends the model developed by Fuchs by implementing a detailed longitudinal and transversal dynamic simulation for BEVs. He focuses on the vehicle's dynamics and how this is impacted by various parameters, including the VCW. Since the main focus of Angerer's research is not on deriving a vehicle package, volume modeling of vehicle components is only of marginal importance and is not treated sufficiently for deriving a vehicle architecture.

(Felgenhauer, 2019) creates a model that enables automatic derivation of a vehicle architecture. He considers several vehicle components and models their geometrical interdependencies. However, his

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approach does not consider VCW fluctuations. Furthermore, the focus is on the front-end of the vehicle, which excludes the battery from the modeling.
(Fuchs, 2014), models the entire vehicle (including the battery) and generates a complete vehicle architecture. The VCW is modeled by adding the weight of the powertrain components (which are modeled singularly) to a basis weight. This approach is only able to estimate the SWCs of powertrain components since the modeling does not include the weights of the other vehicle components.
To our knowledge, no author has ever developed an approach that considers both SVCs and SWCs. Therefore, the scope of this paper is to introduce a novel approach, which can be applied to every vehicle component, capable of estimating these effects.

3 METHOD

Figure 1 illustrates our approach. It consists of six steps: input initialization, longitudinal dynamic simulation, volumetric modeling, identification of geometrical interdependencies, weight modeling, and output of the feasible vehicle architecture.

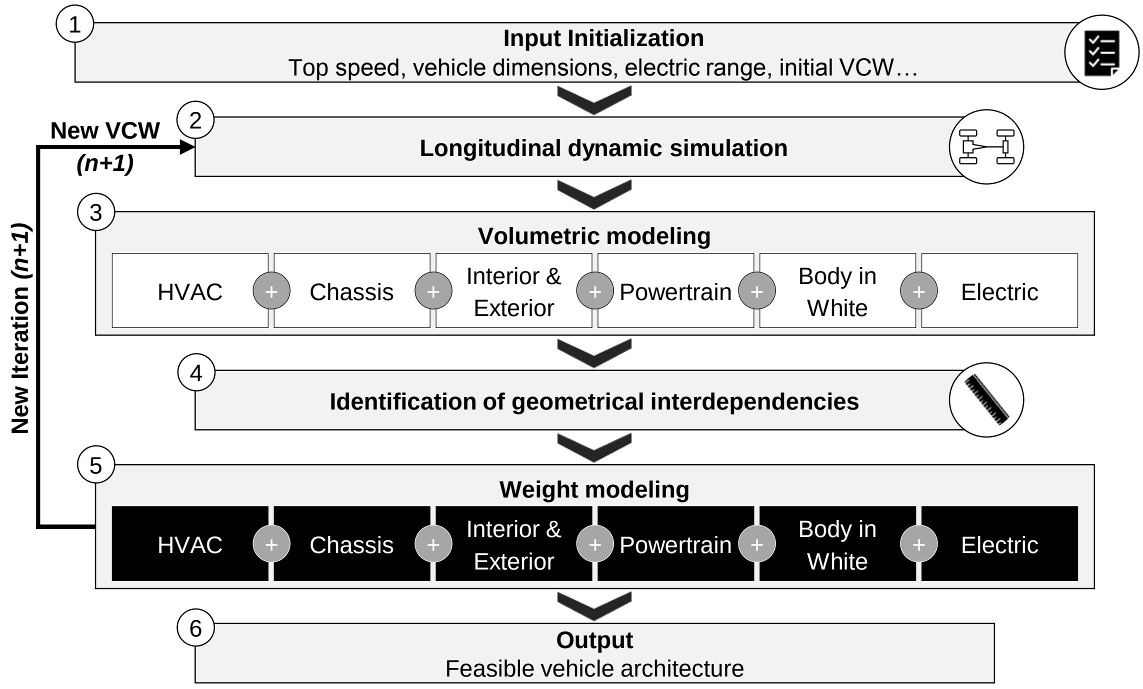

Figure 1. Overview of the proposed approach

The first stage in the process of deriving the vehicle architecture is to initialize the input parameters (Step 1, Figure 1). We conducted extensive literature research to identify the parameters that are available in the early development phase (and that can therefore be used as an input) (Nicoletti *et al.*, 2020a). The identified parameters are implemented as model inputs. Examples are general powertrain requirements (such as the vehicle's top speed), vehicle dimensions, required electric range, and initial VCW. The latter is needed to calculate the vehicle consumption for the first iteration loop and is estimated empirically (Felgenhauer *et al.*, 2019).
Based on the input parameters, a longitudinal dynamic simulation (Step 2, Figure 1) calculates the powertrain specifications, such as required power, vehicle consumption, and required battery capacity. The implemented longitudinal simulation is described in (König *et al.*, 2020).
The outputs of Step 2 are combined with the inputs of Step 1 and employed in the dimensioning of the vehicle components (Step 3, Figure 1). For this purpose, the vehicle is divided into six modules, the volume of each module being derived using empirical and semi-physical models.
Once the module volumes have been defined, the components are positioned in the vehicle (Step 4, Figure 1). This necessitates identifying the geometrical interdependencies between the components. These interdependencies can be further expressed as dimensional chains (Felgenhauer *et al.*, 2018; Nicoletti *et al.*, 2020b). In this step, the feasibility of the vehicle architecture is tested to ensure that no collisions occur between the vehicle components.

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The previously calculated component volumes are then used, together with the input parameters and the outputs of the longitudinal simulation, to estimate the corresponding component weights using empirical modeling (Step 5, Figure 1). The component weights are grouped in the same modules as used in Step 3 to derive the corresponding module weights. A new VCW is then obtained by totaling the individual module weights. This will not necessarily be equal to the initial VCW used in Step 1, so the vehicle consumption is recalculated by inserting the new VCW into the longitudinal simulation. As the VCW has changed, the simulation now outputs different power and battery capacity requirements, which influences the volume and weight of the components. A new iteration loop is then conducted and the SVCs and SWCs are estimated. Finally, the loop outputs a new VCW estimate.
The sequence of steps iterates until the difference between the VCW of the iteration *n* and that of the iteration *n-1* falls within a given tolerance level. Once this condition is fulfilled, the method outputs the feasible vehicle architecture (Step 6, Figure 1).
To further explain our approach and show how individual components are modeled, we use the example of the gearbox. For this purpose, we create a database of existing gearboxes, using data taken from the benchmarking site a2mac1 (A2MAC1 EUL). With the help of this database, it is possible to identify features and parameters that are of relevance to the dimensioning process.
The method shown in Figure 1 is now applied to the gearbox:

- Input initialization and longitudinal dynamic simulation (Step 1 and Step 2).
- Volumetric modeling (Step 3)
- Identification of geometrical interdependencies (Step 4)
- Weight modeling (Step 5)

These steps are discussed in the following sections. Although this paper focuses on the gearbox, the same method can be applied to any vehicle component. After validating the model (Section 3.5), we apply it to quantify the geometrical interdependencies between the gearbox and the battery (Section 4).

3.1 Input initialization and longitudinal dynamic simulation

Gearbox dimensioning requires a certain set of input parameters. These are the maximum rotational speed of the electric machine, its maximum and nominal torque, and the gearbox transmission ratio (Nicoletti *et al.*, 2020a). These variables are either required as inputs (Step 1, Figure 1) or are calculated from the longitudinal simulation (Step 2, Figure 1).

3.2 Volumetric modeling

A BEV gearbox has two possible layouts: coaxial or with parallel axles. These two categories can be further subdivided into four layouts (Figure 2).
The first layout (Layout A, Figure 2) consists of a two-stage gear unit with three shafts. The input (black line connecting the electric machine and Wheel 1) and the output shaft (black dotted line) are positioned in a parallel configuration. The gearbox pinion (Wheel 1) is connected directly to the electric machine and is in contact with Wheel 2. This is the first stage of the gearbox. The resulting transmission ratio is calculated from the rotational speed of the two shafts (n_1, n_2), the number of teeth in their wheels (z_1, z_2) or the wheel diameters (d_{w1}, d_{w2}) as shown in Eq. (1) (Niemann and Winter, 2003, p. 35).

$$i_{12} = \frac{n_1}{n_2} = \frac{z_2}{z_1} = \frac{d_{w1}}{d_{w2}} \quad (1)$$

In the second stage (Wheels 3 and 4), the total transmission ratio is increased further. Finally, Wheel 4 is connected to the differential housing (D) which splits the torque between the output shafts.
The second layout (Layout B, Figure 2) consists of a planetary gear connected to a differential gear in a parallel axle layout. The planetary gear consists of four elements: sun, planets, anulus, and planet carrier. The electric machine transfers its torque to the sun gear (S) which is connected to the planets (P), which in turn are connected to the anulus (A). In Figure 2, the anulus is fixed to the gearbox housing. The planets are also connected by the planet carrier (C). The planetary gear is the first stage of the gearbox. The second stage is created by fixing a wheel onto the planet carrier and connecting it with the outer wheel of the differential (Wheel 4).
The third layout (Layout C, Figure 2) has the same components as Layout A. However, the electric machine is positioned coaxially to the axle, i.e. the input and output shafts are coaxial. The input shaft of the electric machine is hollow, and the output shaft is fed through it. This layout has smaller dimensions than Layout A. The major constraint is that the axle distances (a_{12}, a_{34}) between the two

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stages have to be equal, as shown in Eq. (2). Due to the constraint of Eq. (2), the number of teeth and the diameters of the wheels in the two stages are interdependent.

$$a_{12} = (d_{w1} + d_{w2}) \cdot 0.5 = (d_{w3} + d_{w4}) \cdot 0.5 = a_{34} \quad (2)$$

The fourth layout (Layout D, Figure 2) consists of a planetary gear with two planetary stages. The first stage is built from the connection between the sun (S) and the first planetary stage (P1). The planets P1 are also connected to the second planetary stage (P2), which is held together by a planet carrier and connected to the anulus (A). Finally, the planet carrier is connected to the differential stage (D).

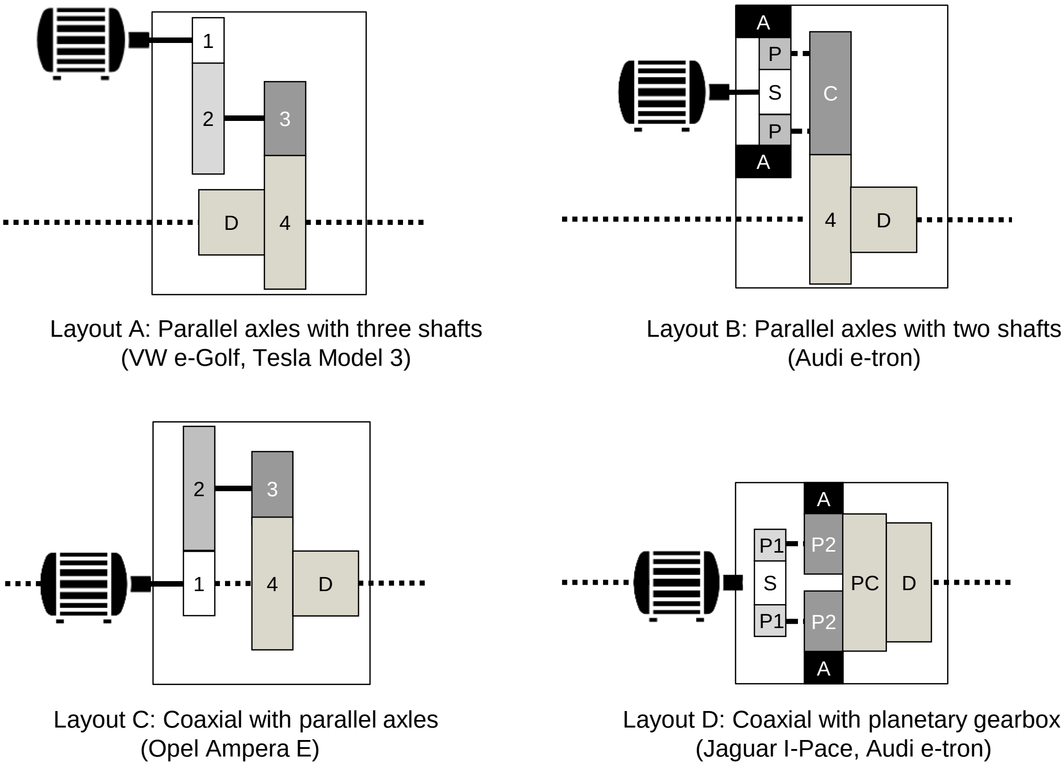

Figure 2. Overview of possible gearbox and electric machine layouts

We do not consider two-speed gearboxes, such as in the BMW i8 or the Porsche Taycan, since their usage is currently limited to vehicles in the premium price class.
Based on the chosen layout, volumetric modeling is employed to estimate the SVCs of the gearbox (Figure 3). The model structure follows the approach presented by (Angerer *et al.*, 2018 - 2018).

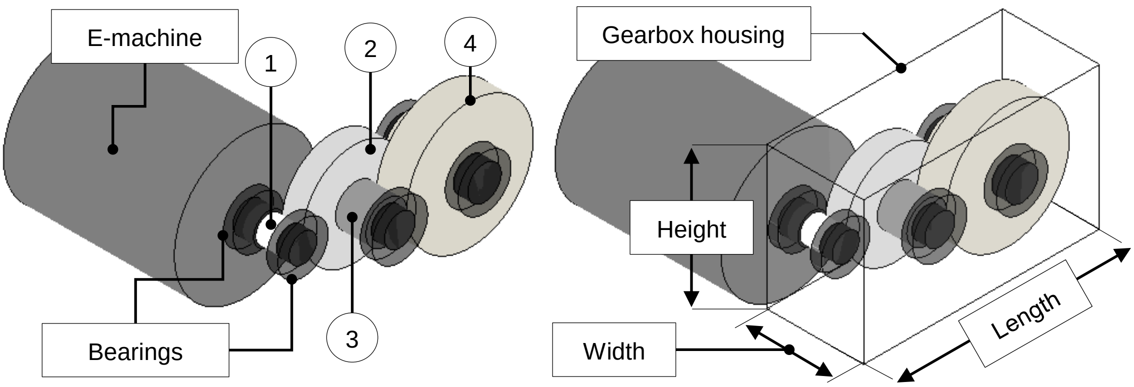

Figure 3. Calculated shaft, bearing, and wheels (left-hand side) and resulting housing dimensions (right-hand side) for a simulated Layout A gearbox

Given the model inputs (Section 3.1), it is possible to roughly estimate the wheel dimensions (Naunheimer et al., 2019, p. 228). Based on this, the model calculates the shaft, wheels, and bearing loads. Shafts and wheels are dimensioned based on these loads and suitable bearings chosen from a

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bearing catalog. Finally, the calculated components are positioned according to the chosen layout (Figure 2) and the housing dimensions are derived from the size of the wheels (Figure 3).

3.3 Identification of geometrical interdependencies

This section shows how the components affect each other, using the interdependency between the gearbox and the battery as an example. The four layouts (Figure 2) can be combined with three different positionings: in front of, behind, or coaxial to the axle (Figure 4).

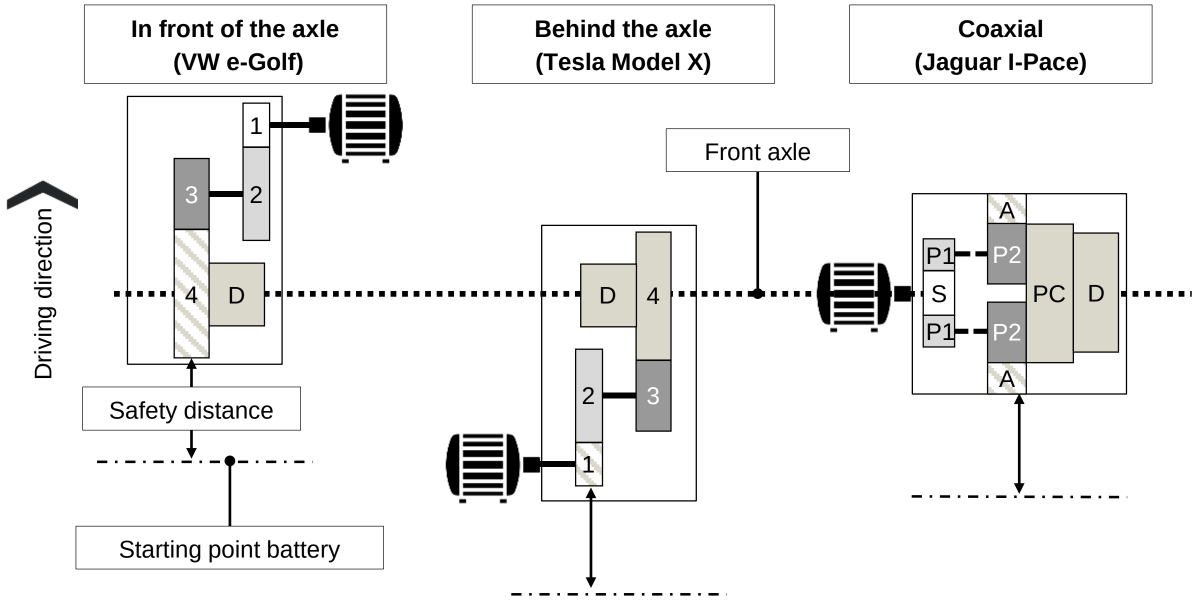

Figure 4. Influence of gearbox layout and position on battery positioning

The position of the gearbox influences that of the battery. To comply with crash safety requirements, a minimum distance has to be maintained between the battery and the drive unit (safety distance, see Figure 4). This is to ensure that in the event of a frontal collision, the drive unit will not damage the battery. While the gearbox position can vary, a safety distance always has to be ensured. This means that the battery can be placed closer to or further away from the front axle, depending on the chosen gearbox position and layout.

If the gearbox position and layout are known, the critical gearbox component for the battery can be identified. Figure 4 shows the dependency between the critical gearbox component (indicated by the grey hatched pattern) and the battery position.

In the first case (gearbox in front of the axle, Figure 4), the critical component is the differential (Wheel 4). In the second configuration (gearbox behind the axle, Figure 4), the critical elements are the electric machine and the gearbox pinion (Wheel 1). Finally, in the third case (coaxial gearbox position, Figure 4) the critical element is the anulus (A) of the gearbox. Since the critical component depends on the gearbox layout and position, to model the impact of the gearbox on the battery, the volumes and dimensions of all the gears are estimated (see Section 3.2).

3.4 Weight modeling

Weight modeling makes it possible to estimate the SWCs of the gearbox. To do this, we divide the gearbox into its key components: wheels, shafts, bearings, and housing.

We assume that the material used for the wheels and shafts is 16MnCr5 steel. Since the model presented in Section 3.2 calculates the volumes of both wheels and shafts, the corresponding weights can be derived by multiplying the volume with the steel density.

Since the bearings are taken directly from a catalog, we also obtain the corresponding weight for each chosen bearing from it.

We suppose that the alloy AlSi9Cu3 is used for the gearbox housing. Volumetric modeling estimates the housing volume from the dimensions of the wheels. The weight of the housing is therefore calculated by multiplying the estimated volume with the alloy density.

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3.5 Model validation

For the model validation, we created a validation database of four vehicles, in which the gearbox inputs (Section 3.1) were collected for each vehicle. The inputs were used to simulate the gearboxes of each of the validation vehicles. The resulting gearbox length, width, height (as defined in Figure 3), and weight were then calculated and compared with the real values of the validation vehicles. Table 1 shows the percentual deviation between the simulated and the real-world dimensions.

Table 1. Overview of model performance for different gearboxes with Layout A

Model	Model year	Length	Height	Width	Weight
BMW i3	2019	2.94 %	0.69 %	23.11%	1.92 %
VW e-Golf	2014	7.64 %	-2.89 %	-10.96 %	0.92 %
VW e-Up	2016	12.00 %	-7.84 %	6.64 %	6.06 %
Renault Twizy	2014	-0.21 %	-10.50 %	6.91 %	10.98 %

Leaving the gearbox weight aside, the deviation in gearbox dimensions is always below 23 %. The deviations are mostly caused by the irregular shapes of the gearbox housing, which cannot be modeled in detail during the early development phase. Further detailing of the gearbox dimensions is required but can be achieved only in the following steps of the development.

The deviation between the calculated and the real-world gearbox weight is due to the partial integration of the gearbox housing in the electric machine housing. In this way, it is possible to enclose both electric machine and gearbox in a single housing, thus saving weight. Since the level of integration depends on the manufacturer's strategy, we cannot model this feature.

Furthermore, the model was compared with the software GAP (Höhn *et al.*, 2008) of the Chair of Machine Elements of the Technical University of Munich. Since the employed version of GAP does not model the dimensions and weight of the housing, it cannot be implemented in our approach. Nevertheless, GAP is suitable to verify the dimensioning of the single wheels. For this scope, several combinations of transmission ratio and maximum torque were simulated in GAP and compared with the model results. Figure 5 shows the results of the two tools for the first stage (Wheels 1 and 2, Figure 3) of a lay shaft gearbox. The mean absolute error (MAE) between the model and GAP is 4 mm for the first wheel, and 9 mm for the second wheel.

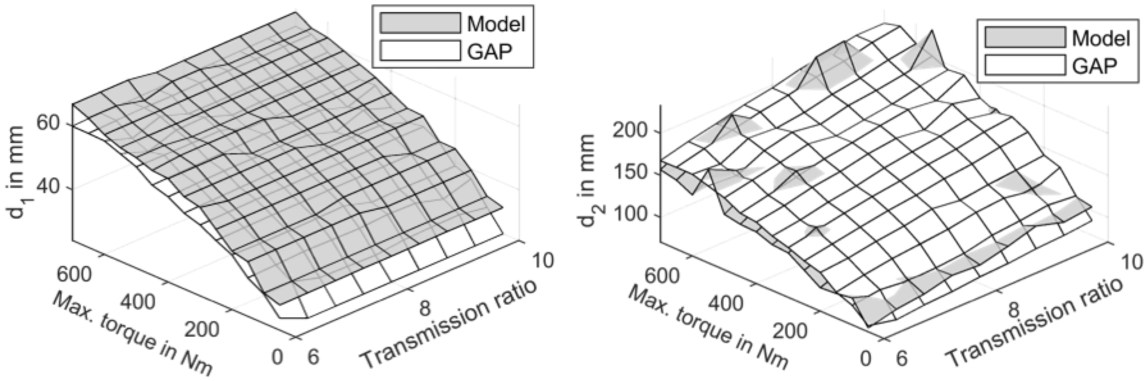

Figure 5. Comparison between the wheel diameters (first stage of a lay shaft gearbox) calculated by our model and GAP

4 RESULTS

In this section, we quantify the SWCs and SVCs in the gearbox by investigating the influence of the maximum input torque (from the electric machine) and the transmission ratio. These parameters impact the gearbox dimensions and are in turn influenced by the vehicle weight. As explained in Section 1, weight fluctuations during the vehicle architecture design can trigger changes in the powertrain components. If a weight fluctuation causes a VCW increase, higher torque is needed to maintain the same powertrain requirements (such as the same acceleration time). This higher torque may, in turn, necessitate an adjustment of the transmission ratio.

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The limits of parameter variation are determined from the vehicles in the gearbox database. The maximum torque limits are set between 150 Nm and 600 Nm. For the transmission ratio, we set the lower limit to 6 and the upper one to 10. Figure 6 shows the SVCs and SWCs for a Layout A gearbox.

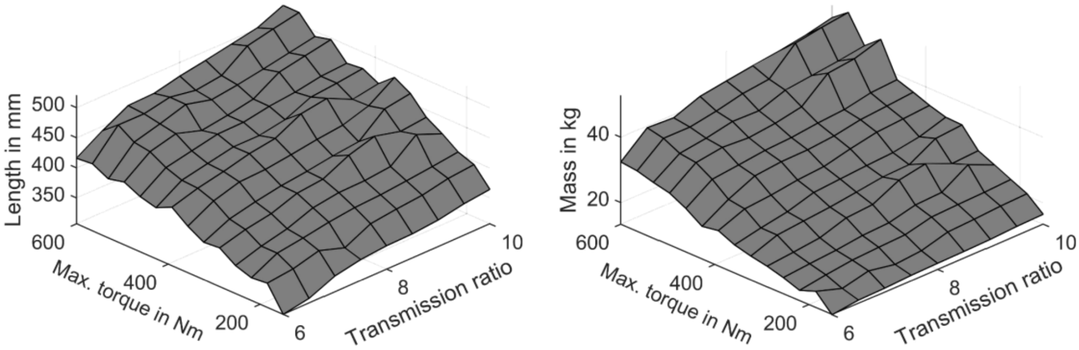

Figure 6. SVC (left side) and SWCs (right side) for a Layout A gearbox

To enable SVC quantification, the left-hand side of Figure 6 shows the variation in the gearbox when torque and transmission ratio increase. An increase in transmission ratio requires a greater number of teeth for the wheels as well as greater wheel diameters, while an increase in torque requires bigger shafts as well as wider teeth. Both effects impact the gearbox length and, in turn, its volume. The effects observed with the SVCs result in bigger wheels and shafts, which explains the larger gearbox mass (right-hand side of Figure 6) with increasing transmission ratios and torques.

To assess the interdependency between the gearbox and the battery, we perform a simulation of a Layout A gearbox positioned in front of the axle. The components that are relevant for estimating the battery position are those placed behind the axle. In this case, it is the differential (Figure 4). We calculate different combinations of transmission ratio and maximum torque and derive the *critical length* of the gearbox for each combination (left-hand side of Figure 7). An increase in critical length corresponds to an equal shift in the battery position away from the axle, i.e. a reduction in battery space.

Figure 6 clearly shows that the SVCs and SWCs in the gearbox are discontinuous, which hinders a direct assessment of their effect on other components. To solve this problem, we plot the resulting critical length as a scatter plot (Figure 7, right, grey rounded markers) rather than a discontinuous surface. Using the least square method, we then derive the continuous surface which optimally fits the calculated points (Figure 7, right, dark grey surface). The formula for the surface (which has an R square value of 90 %) is given in Eq. (3).

$$Critical\ length = 46.22\ mm + (5.99\ mm) \cdot trans.ratio + \left(0.07\frac{mm}{Nm}\right) \cdot \max.torque \qquad (3)$$

Although Eq. (3) represents a simplification of the model, it is a continuous curve and therefore enables an estimation of the SVCs in the battery. The maximum torque is multiplied by the factor 0.07, which means that an increase of 13 Nm in the maximum torque triggers an increase of 1 mm in the critical length. The same effect is obtained by increasing the transmission ratio by approximately 0.17.

The question now is: How does the battery capacity change if the critical length increases? We answer this question using the Audi e-tron battery as an example. With the data of (Doerr *et al.*, 2019) it can be estimated that the e-tron battery has a gross volumetric energy density at pack level of 216 Wh/l. Using the battery dimensions given by (Doerr *et al.*, 2019), it can be further estimated that a space loss in the X direction (the driving direction) causes a capacity loss equal to approximately 48 Wh/mm.

Although this capacity loss is not particularly high, it is based on the assumption that the volumetric energy density of the battery also remains constant if the battery dimensions are changed. This is not always the case, since a drop of 1 mm could also result in an entire row of cells no longer fitting, which would cause an abrupt drop in both the installed capacity and the volumetric energy density.

This simple example shows how SVCs in components such as the gearbox can trigger SVCs in other components such as the battery and thus influence the maximum installable capacity.

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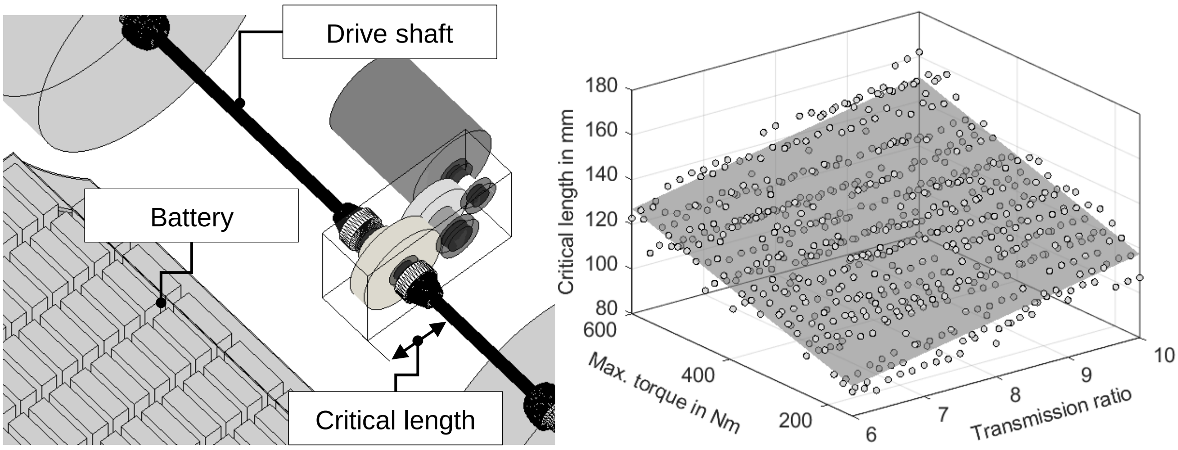

Figure 7. SVCs triggered on the battery by a Layout A gearbox positioned in front of the axle

The SVCs described by Eq. (3) are a simplification and discontinuities may occur by varying transmission ratio and maximum torque. However, this estimation can support concept engineers in the early development phase (where the knowledge of the components is still limited) as a rule of thumb, to estimate the available battery space and the influence of parameters such as transmission ratio and maximum torque. Without the modeling proposed in this paper, it would not be possible to quantify the occurring SVCs in such an early phase of the vehicle architecture design. Such rules of thumbs can be used to correctly define the vehicle portfolio. The single components can be modeled in more detail in the following steps of the development, where more detailed information and models are available.

5 CONCLUSION AND OUTLOOK

BEV architectures require the usage of new components, which leads to component interdependencies that still have to be quantified. To solve this problem, we propose an innovative approach to quantify both SVCs and SWCs (Figure 1). To explain the approach, we exemplary model the gearbox, nevertheless the same procedure can be applied to any vehicle component.

The evaluation of the gearbox model shows a good approximation to the GAP model, which ensures the correctness of the implementation. Furthermore, the model requires a low number of inputs and is therefore suitable for the early development phase. A drawback of the gearbox model is, that it is not possible at this stage of the development to consider factors such as cooling and lubrication requirements or the deformation of the gears once loaded. These effects are particularly important to model the gearbox losses but require finer modeling, which is not possible at this stage of vehicle development. Nevertheless, the presented model is still able to provide a satisfactory approximation of the gearbox dimensions, and can therefore describe its impact on other components such as the battery.

Although the SVCs triggered by the gearbox are not particularly high, there are other components (such as the passenger compartment, the wheels, and the electric machine) that also impact the battery. Quantification of all the relevant interdependencies (with due consideration for the SVCs and SWCs) is only possible by modeling every vehicle component (following the six-step method presented for the gearbox) and integrating them using the approach presented in Figure 1. Further models to size the battery and the electric machine will be presented in future publications.

The approach presented in this paper can support concept engineers during the early development phase to define a correct vehicle portfolio. Since in this phase there is no detailed information about the vehicle, the concept engineers do not have easy ways to ensure that the range requirement or the acceleration time contained in the vehicle portfolio are feasible. The hereby presented tool, uses the input generated by the concept engineers, automatically generates a rough package of the vehicle, and tests the feasibility of the vehicle architecture. In this way, concept engineers can test in a few seconds whether a given set of vehicle requirements is attainable. This real-time knowledge avoids errors in the vehicle portfolio definition, saves time, and reduces development costs.

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