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ORIGINAL PAPER

Optimal and prototype dimensioning of 48V P0+P4 hybrid drivetrains

Matthias Werra[1] · Axel Sturm[1] · Ferit Küçükay[1]

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Abstract

This paper presents a virtual toolchain for the optimal concept and prototype dimensioning of 48 V hybrid drivetrains. First, this toolchain is used to dimension the drivetrain components for a 48 V P0+P4 hybrid which combines an electric machine in the belt drive of the internal combustion engine and a second electric machine at the rear axle. On an optimal concept level, the power and gear ratios of the electric components in the 48 V system are defined for the best fuel consumption and performance. In the second step, the optimal P0+P4 drivetrain is simulated with a prototype model using a realistic rule-based operating strategy to determine realistic behavior in legal cycles and customer operation. The optimal variant shows a fuel consumption reduction in the Worldwide harmonized Light Duty Test Cycle of 13.6 % compared to a conventional vehicle whereas the prototype simulation shows a relatively higher savings potential of 14.8 %. In the prototype simulation for customer operation, the 48 V hybrid drivetrain reduces the fuel consumption by up to 24.6 % in urban areas due to a high amount of launching and braking events. Extra-urban and highway areas show fuel reductions up to 11.6 % and 4.2 %, respectively due to higher vehicle speed and power requirements. The presented virtual toolchain can be used to combine optimal concept dimensioning with close to reality behaviour simulations to maximise realistic statements and minimize time effort.

Keywords Virtual development methods · Hybrid drivetrains · 48 V · Fuel consumption · Mobility

1 Introduction

Fuel consumption reduction is one of the key objectives of today's vehicle development. In this regard, the drivetrain electrification is one of the most important development. A broad field emerges between low voltage concepts with 48 V and high voltage battery electric vehicles with voltages up to 800 V. At this point, 48 V electrification offers an optimal compromise between low costs in combination with very high savings potential [1]. Low costs are achieved by avoiding great technical effort for high voltage safety precautions [2, 3]. Compared to the 12 V power supply, the 48 V level allows higher electric power of the electric machine (EM) and battery.

State-of-the-art 48 V hybrid systems are capable of all hybrid functionalities, e.g. recuperation, electric driving and coasting while engine is off. This ultimately increases the overall system efficiency and accordingly reduces fuel consumption. With the 48 V power supply a reasonable electric power of up to 30 kW can be realised [4–6]. Even higher power is possible but it has to be evaluated if the benefit of the additional power can justify the effort. For higher electric power the current is high what leads to bigger wire cross-sections and has an impact on the costs, weight and installation space. Furthermore, it affects the size and the thermal management of the EM as well as the 48 V battery which has to provide the electric power.

There are various hybrid topologies with respect to the position of the EM in the drivetrain. Each topology enables different hybrid functionalities and has distinct advantages and disadvantages [1]. The EM position ranges from belt driven starter generators (P0) up to topologies with a decoupler between the internal combustion engine, short ICE, and the gearbox (P2) as well as with an additional electric rear axle (P4) [5]. Furthermore, there are diverse combinations of different topologies in order to link their specific advantages and

Matthias Werra
m.werra@tu-braunschweig.de

Axel Sturm
a.sturm@tu-braunschweig.de

Ferit Küçükay
iffzg@tu-braunschweig.de

[1] Institute of Automotive Engineering, TU Braunschweig, Hans-Sommer-Str. 4, 38106 Brunswick, Germany

additionally avoid disadvantages when choosing a single EM topology. Topologies with more than one EM provide a higher sum of electric system performance to enhance the capabilities regarding the various hybrid functionalities. Currently, vehicle manufacturers are starting to broadly expand the 48 V technology regarding their high volume vehicles [7]. Additionally, vehicle suppliers are developing add-on solutions for 48 V systems and demonstration vehicles to show the possible fuel reduction potential of various 48 V hybrid systems [8, 9].

The best topology is a result of the combination of different boundary conditions, such as the type of the electric machine and its efficiency map, selected gear ratios and the vehicle use cases. A P0+P4 topology shows a significant fuel consumption reduction potential. Compared to other hybrid topologies it shows a high performance improvement due to the additional electric rear axle [5, 10]. In this paper, a profound dimensioning and detailed analysis of a 48 V hybrid P0+P4 topology is performed. The usage of a virtual toolchain allows for dimensioning and evaluating a high number of electrified drivetrains. The result is an optimal—in terms of fuel consumption and performance—48 V hybrid P0+P4 drivetrain with an optimal electric power of the P0 and P4 electric machines, 48V battery capacity and gear ratio for the electric machines in the Worldwide harmonized Light Duty Test Cycle (WLTC). In a second step, the optimal drivetrain is simulated using a detailed simulation model for prototype dimensioning to allow close to reality statements regarding the fuel savings potential. Based on this, the optimal drivetrain configuration is analysed in the WLTC and in customer cycles. Furthermore, the optimal and prototype dimensioning models are compared with each other. As a result, statements in terms of model differences between the two simulation models are possible.

2 Virtual toolchain for optimal concept and prototype dimensioning

To achieve the highest fuel savings potential and best performance it is crucial to determine and dimension the drivetrain in an optimal way. In the vehicle's concept phase simulation models are used to evaluate a concept under certain requirements. Regarding the simulation methodology various boundary conditions can be taken into account. At first, a conventional vehicle and its use cases are defined as a basis. The use cases serve to compare and evaluate different system setups. Then, requirements such as minimum fuel consumption reduction potentials or minimum performance requirements are specified. Afterwards, various 48 V hybrid topologies defining the position of the EM in the drivetrain are determined. In this paper, the P0+P4 configuration is chosen. In addition to that, the dimensioning of the electric components has an essential impact on the fuel consumption and the hybrid functionalities. By using simulation models, the fuel and energy consumption are calculated to evaluate the defined system setups. This allows the analysis of fuel consumption reduction potentials in the investigated parameter space. Eventually, statements on optimal system configurations can be made in an early development stage while fulfilling the predefined requirements.

At the Institute of Automotive Engineering at the TU Braunschweig different tools for various powertrain development stages exist. The concept dimensioning process is used for big parameter variations and powertrain concept comparisons. It allows a fast calculation and concept optimization. The prototype dimensioning is a detailed simulation model with component based modeling for a detailed powertrain evaluation, optimization and dimensioning. Therefore, a virtual toolchain for the optimal concept and prototype dimensioning has been developed which is shown in Fig. 1. Within the first step the optimal concept is determined by a

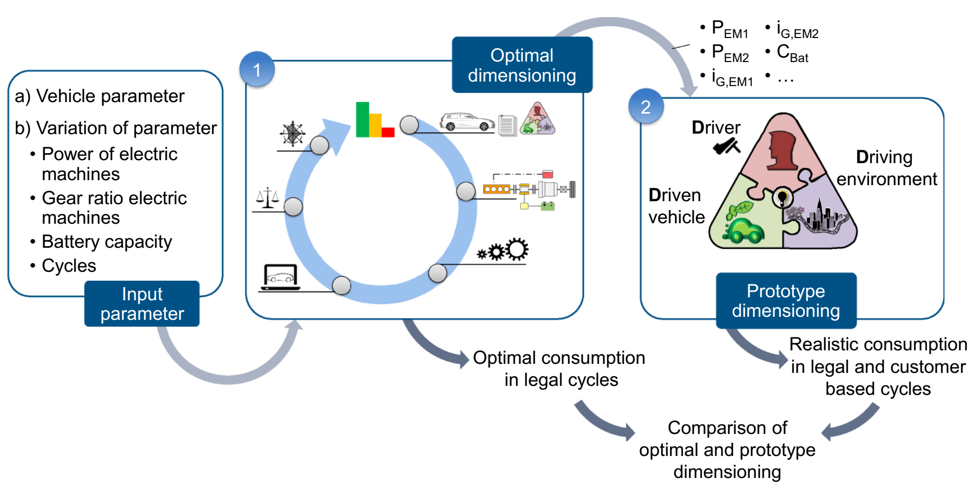


Fig. 1 Virtual toolchain for optimal concept and prototype dimensioning of 48 V hybrid drivetrains

parameter variation of particular powertrain key parameters such as the power of the EM and the EM gear ratios as well as the 48 V battery capacity. The result of the optimal concept dimensioning is an optimal hybrid drivetrain regarding the fuel consumption combined with optimal performance. This hybrid drivetrain configuration is put into the second prototype simulation model in order to generate realistic consumption statements in the WLTC and customer based cycles. This is done by the distinction of driving environment and driver behavior. The two simulation models are explained in detail in the following.

2.1 Optimal concept dimensioning environment

The concept dimensioning is a development environment consisting of various methods, tools and processes following consecutively. They portray the necessary development steps for identification, dimensioning and optimization of new electrified powertrains. The process chain is illustrated in Fig. 2 and is based on following inputs: the vehicle with its basic parameters in combination with the vehicle requirements (1) and the hybrid function requirements with the topology (2). The inputs can be given for one vehicle or for a fleet with various vehicles [11]. Based on the inputs the vehicle and the powertrain components have to be dimensioned (3) for further simulation (4), evaluation (5) and optimization (6). The third step, the dimensioning of the key components shown on the bottom right of Fig. 2, is an integrated tool including further detailed component information and model based component masses [12]. It displays the link between the requirements, the vehicle and the powertrain. At this step, the energy converters ICE and EM, the energy storages tank and battery and the remaining drivetrain with all the components are defined by the user. Either one particular drivetrain or a range for the different key components, such as power range for the EM, can be set up for simulation.

By this, a various number of different hybrid drivetrain variants are defined which can be simulated in the fourth step. The simulation is performed with an adaptive modular simulation tool. This model, developed at the Institute of Automotive Engineering, is a backwards-simulation model and is built modularly in order to include all powertrain variants and degrees of electrification [13, 14]. Based upon the simulation data the powertrain configurations are evaluated in order to find the most promising concept in terms of fuel consumption and performance. Fig. 3 displays the layout and the different modules of the simulation model which are used in the fourth step illustrated in Fig. 2. The first module is the vehicle including all necessary vehicle parameters and the simulation cycle. Based on the cycle profile the driving resistances and corresponding torque and speed at the wheel,

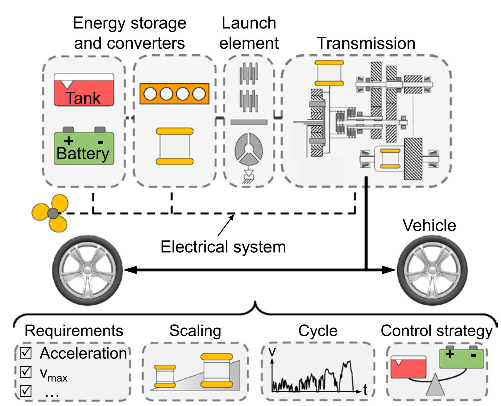
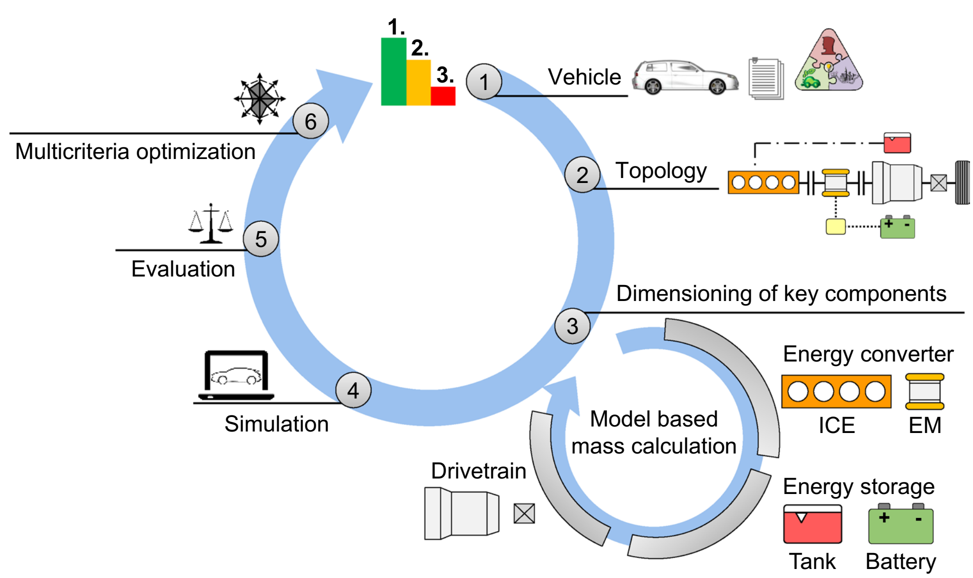


Fig. 3 Modular backwards-simulation model used for the optimal dimensioning [13, 14]

Fig. 2 Process chain for the optimal concept dimensioning and evaluation of drivetrains [11]



which are the entry of the next module (transmission), are calculated backwards and thus represent the output of the transmission. The transmission module consists of mathematical description of the transmission with information about the mechanical connections, shifting elements as well as the transmission modes. The transmission efficiency is calculated by utilizing either a simplified loss approach or generated transmission loss maps [15].

The transmission module is capable of holding up to nine EMs enabling the simulation of complex Dedicated Hybrid Transmission concepts as well as almost every future transmission concept. Within the transmission module all operation points—torques and speeds—are calculated for the defined EMs and the transmission input. The input torque and speed of the transmission are output parameters for the launch element module. The launch element can be either a direct connection, dog clutch, slipping clutch or torque converter. The input torque and speed are the output of the propulsion module, thus representing the entry of the propulsion model. In the propulsion module conventional and parallel hybrid variants (from P0 up to P2) as well as pure electric or fuel cell concepts can be simulated. The efficiencies of all energy converters (ICE, EM) are calculated using efficiency maps. The scaling of the power of the energy converters is achieved by scaling the torque of the energy converter. The corresponding efficiency map is scaled as well in order to fit the new maximum torque. Furthermore, an operating strategy is used for the simulation for the optimal coordination of the drivetrain components. Within this paper an adapted Equivalent Consumption Minimization Strategy (ECMS) is used. The ECMS is similar to the control strategy described in [16–18]. The ECMS approach is the comparison of petrochemical energy from the tank E_{Tank} with the electrochemical energy from the battery E_{Battery} with the aim to minimize an equivalent fuel consumption according to:

$$E_{\mathrm{Equivalent}} = E_{\mathrm{Tank}} + (k_1 + k_2 \cdot \Delta\,\mathrm{SOC}) \cdot E_{\mathrm{Battery}}. \tag{1}$$

Two equivalence factors k_1 (general equivalence factor) and k_2 (State-Of-Charge (SOC) weighting for achieving a reliable SOC level) are used for the comparison between the two energy forms. For a charge sustaining simulation, the vehicle has to be SOC neutral within the cycle (SOC at cycle start identical with SOC at cycle end). To achieve SOC neutrality and to achieve the best and most accurate consumption result within the efficiency simulation the cycle simulation is iterated several times to adapt the equivalence factor k_1. The limits for iteration are commonly set to 0.01 l/100 km. The equivalence factor k_1 can vary between different concepts and system configurations to prevent an unequal evaluation or a preference of certain concepts within the concept comparison process. The factor k_2 weights the electrical energy according to the current SOC. Due to this, the operating strategy approach is enhanced to achieve a better SOC stability.

For the optimal concept environment, the evaluation is either done upon single evaluation criteria such as the fuel consumption in a given cycle or a multi-criteria evaluation system [19]. Within the presented 48 V analysis the powertrains are not optimized but evaluated based upon their cycle efficiency and performance.

The described optimal concept simulation model with the ECMS operating strategy determines the local energetic optimum for each simulation time step to evaluate and optimise powertrain concepts. Due to this local optimisation in each time step, it represents an optimum for the operation of a particular vehicle for a known driving cycle. In real vehicle operation this local optimisation cannot be applied as the actual vehicle's driving cycle is not known a priori. The ECMS is adapted to execute a more realistic driveability and to prevent driveability errors, such as a lot of mode changes every simulation time step. This is realised by implementing a minimum transmission mode runtime and a minimum ICE runtime. As a result of this method, the optimal concept dimensioning model represents the theoretical local optimum of a particular drivetrain configuration in a specific driving cycle. Hence, no verification of the optimal simulation model by comparing it to measurements is useful and necessary [13].

2.2 Prototype dimensioning environment

The prototype simulation model is a development environment for the simulation of vehicle drivetrains to generate realistic statements regarding the fuel consumption and the behavior of the drivetrain components. The following section describes the main components of the prototype simulation model. It is based on the 3D simulation model which is part of the 3D-method introduced and established by Küçükay [20]. In general, the 3D-method with its three main influences on the load of various vehicle components or component groups and their interaction—the Driving environment, the Driver and the Driven vehicle with the components itself (short: 3D)—is a modular variant based development platform to identify representative data for specific automotive problems and system evaluation within this 3D customer parameter space. The corresponding classification and impact of the parameter space on the vehicle components is described later in this section. At the Institute of Automotive Engineering, the 3D simulation model is used to simulate a variety of vehicle drivetrain configurations within the parameter space. In the last years, the 3D simulation model has been further developed in several scientific works [21, 22]. It has been used in various applications whereby the simulation tool with its main functionalities has been verified. The simulation model for the prototype dimensioning environment in this paper is based on this verified 3D simulation model. It is a further development regarding the modularity to model various 48 V hybrids using only one simulation model.

The first part describes the general structure of the simulation model. Afterwards, statements are made in terms of reliability of simulated results regarding the fuel consumption. Fig. 4 presents the general structure of this modular simulation model. This variable model is capable of representing any 48 V hybrid vehicle with the EM positioned parallel to the ICE. The three parts—Driving environment, Driver and Driven vehicle—include all necessary key components. The Driving environment describes the cycle the vehicle is driven in and can represent various speed profiles. It is possible to simulate legal cycles such as WLTC as well as driving environments based on customer data. The customer simulation consists of statistics extracted out of vehicle measurements. Based on these statistics, speed profiles are generated and represent target vehicle speeds. The target vehicle speed profile is classified into three different environment classes: urban, extra-urban and highway. The driver meets the requirements of legal cycles and has the function of different representative real driver behavior. To meet legal cycles, the driver is implemented as a combination of a controller for the accelerator as well as for the brake pedal. Furthermore, gear shifting points are calculated automatically and comply with legal requirements. For the 3D-method the driver behavior is generated and implemented statistically. The driver has to interact with the pedals and selector lever in accordance with the target vehicle speed profile. Statistical calculations for the accelerator and brake pedal gradients and end positions as well as engine shifting speeds for vehicle with manual transmission (MT) are representative for the driver interaction with the vehicle. This driver behavior depends on the driving style which can be mild, average or sporty [20, 21, 23]. The third part is the driven vehicle with all of its main components between the energy storages (tank and battery) and the wheel. This forward-simulation model calculates the driving resistances to operate the vehicle as supposed to in the driving environment. A traction force is produced to overcome these driving resistances. The simulation model calculates the torques and speeds of all drivetrain components as well as voltages and currents of the electric components. All drivetrain components are simulated based on their physical behavior combined with maps characterizing specific behavior. For example, the ICE speed is calculated based on the produced torque of the ICE combined with the clutch torque and all inertias. The torque is modelled based on the load due to pressing the accelerator pedal by the driver and 12 V battery load. This ultimately results in a produced torque by the ICE. This torque combined with delaying elements due to delays between pushing the accelerator pedal and actually produced torque in combination with inertias result in the torque powering the drivetrain. Maps are used to calculate fuel consumption.

Additionally, the electric module shown in Fig. 4 consists of the 48 V battery, the power electronics PE and the EM. The electric power is generated at different EM positions in the drivetrain. The variable simulation model calculates the EM torque, speed and inertia impact at the corresponding positions shown in Fig. 4 by the dashed line depending on the user input. Thus, in a P4 hybrid drivetrain the EM is positioned at the rear axle with its own final drive. It is possible to combine up to two EMs in the drivetrain. In order to operate the 48 V hybrid vehicle, an operating strategy decides on division of power between the ICE and the EM. By doing so, the hybrid and conventional functionalities are coordinated based on rules. The functionalities are: conventional drive and start-stop-operation, load point shifting, electric drive and recuperation, torque fill, boosting and charging while standing. The torque allocation for the P0 and P4 EM, e.g. while boosting, load point shifting and recuperating, is calculated based on their performance capabilities and efficiencies of these two EMs. Depending on the driving requirement, either one or both EMs are actuated. For high driving requirements the complete potential of both the P0 and P4 EM is exploited, e.g. while boosting or to exploit the complete recuperation potential in a particular

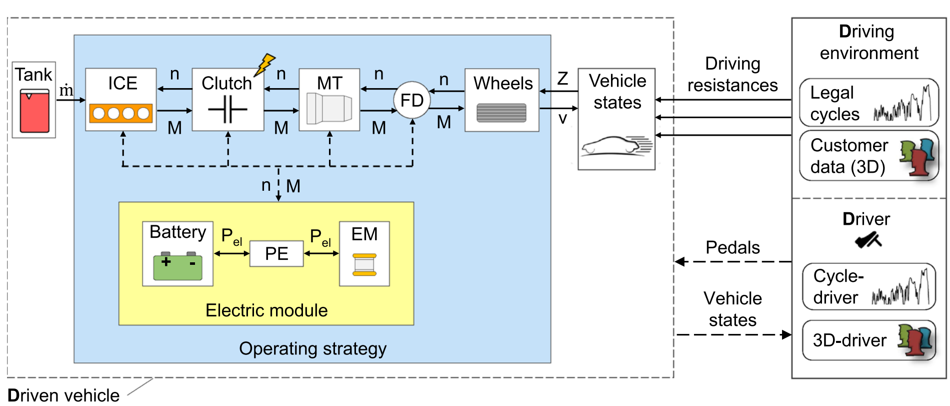


Fig. 4 Modular forward-simulation model used for the prototype simulation

driving situation. The main aim of the operating strategy is to reduce energy consumption as well as to realise close to reality behavior of all drivetrain components in stationary and especially in dynamic situations. For this, another key aspect is the thermal behavior of the electric components. It is modelled based on the thermal physics of these components [24]. This allows the operating strategy to take the derating of the electric components into account. This essentially influences the driving performance and thus the overall system efficiency.

To operate a 48 V hybrid vehicle with a MT, the clutch has to be electrified to avoid the need of a clutch pedal in the vehicle. With this, the operating strategy controls the electrified clutch when the driver is changing the gear by the gear lever manually. The operating strategy controls the opening and closing speeds based on representative courses. These representative courses—including clutch pedal start and end positions and gradients—have been extracted out of measurement data from representative customer data of the 3D-method [20, 21, 23].

As mentioned in the beginning of this section, the 3D simulation model—which is the basis for the presented prototype simulation model in this paper—has been verified in various applications in the last years. For example, in [23] the 3D simulation model is applied for a conventional vehicle (SUV) with an automatic gearbox with 8 gears for a full load acceleration. The comparison between simulation results and test bench measurement is shown in Fig. 5 for various parameters, such as engine speed and longitudinal acceleration. The solid line represents the simulation results and the dashed line shows the measurement data. This comparison shows a high accordance of the simulation results

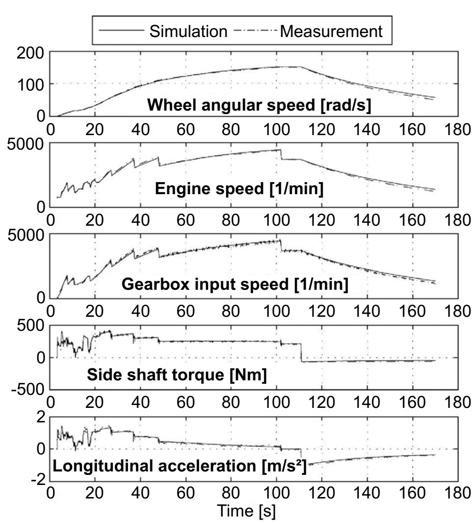


Fig. 5 Verification of the 3D simulation model by a comparison of simulation results of a full load acceleration of a conventional vehicle with test bench measurement performed in [23]

to the test bench measurement [23]. In Ref. [25], the model is applied for a P2 high voltage hybrid vehicle and proves high accordance to measurement test bench data. In other scientific works, e.g. in [19], the model is used to simulate an electric vehicle and is verified in this application area by vehicle measurement data. The further developed modular prototype simulation model presented in this section has been applied as well in a simulation of a conventional vehicle with manual transmission combined with an electrified clutch. The simulation has shown an almost identical fuel consumption in the WLTC compared to measurements of the vehicle on a test bench [26, 27]. All in all, the fuel consumptions and thus the fuel savings simulated with this model are assumed as sufficiently reliable and robust.

2.3 Simulation model differences

Both used simulation models are fundamentally different. The concept dimensioning simulation tool is a backwards-simulation based upon the cycle profile (speed and time). The prototype dimensioning simulation tool is a forward-simulation model which has a driver model. In addition to the simulation of legal cycles, this allows the simulation of customer cycles based upon the 3D-method.

A main model difference is the depth of aggregate modeling and their loss approaches. In general, the aggregate modeling in the prototype simulation model is more detailed than in the optimal simulation model. For example, the optimal simulation calculates the ICE torque and speed based on the backwards-calculation without any inertia impact. The resulting ICE torque and speed in the prototype simulation is the product of sub models which simulate the load requirement of the driver, delaying elements and inertia calculation in combination with characteristic maps, e.g. combustion torque maps depending on the engine load and speed. The differences in level of detail between the two simulation models apply for other aggregates such as the EM.

Regarding the loss approaches, the concept dimensioning utilizes a simplified transmission loss approach which calculates the losses depending on the transmission output power. The characteristic curves are derived and approximated from a relation between the transmission input and output power. The approach does not factor in shift processes. Basis for the simplified loss approach can be vehicle measurements, simulation results with lossmaps or simulated lossmaps. The approach allows an overall good approximation for different transmissions [13]. In contrast to this, the prototype simulation uses a map based approach. The overall transmission losses are dependent on torque, speed and the current gear. Basically, the lossmaps are generated by a detailed model whereby the transmission kinematics are defined mathematically so that the losses of all gearbox components can be calculated, e.g. the gears, bearings and seals [15]. In the optimal dimensioning, the battery model is a simplified open circuit model with one resistance without thermal behavior simulation. The battery efficiency is calculated by utilizing a simplified RC model with SOC dependent characteristic curves for resistance and voltage. In the prototype simulation, the approach of the optimal simulation model for battery efficiency calculation is combined with a model regarding thermal behavior. This has an essential impact on the battery charging and discharging performance. The thermal behavior is represented by a 0-D thermal model. The temperature impact is considered based on dependent characteristic curves for the maximum possible charge and discharge power.

The concept simulation tool calculates with static lossmaps for every cycle time step with a frequency of 1 Hz. With the prototype simulation the aggregates are more detailed and also include dynamic processes with a higher frequency of 1000 Hz. The advantage of the concept dimensioning tool is a low simulation time, high flexibility for the parameterization of different powertrains and the possibility of parallelizing the simulation to the maximum of cores provided by the computer hardware. This allows high amounts of simulations per day whereas the prototype simulation allows only one simulation per simulation program.

Further differences refer to the operating strategy and operation limitations. The concept dimensioning uses an optimal consumption approach while the prototype simulation uses a realistic rule-based operating strategy. Detailed thermal behavior modeling is implemented in the prototype model resulting in realistic derating behavior. In comparison to the prototype simulation, the concept dimensioning has no detailed brake system modeling regarding the brake torque distribution between the axles. The impact of the different simulation models and their respective modeling is discussed in the next chapters. The most important differences are summarised in Table 1.

3 Simulation model comparison

For the 48 V hybrid topology comparison a conventional mid-sized vehicle with front-wheel drive and a gasoline engine is chosen as a basis. The corresponding vehicle parameters are displayed in Table 2. The conventional vehicle is equipped with a state-of-the-art 12 V alternator which can recuperate energy in order to supply auxiliaries and realises a stop/start system for the conventional vehicle. The manual transmission of the base vehicle has six gears with the following gear ratios from gear 1 to gear 6: 3.8, 2.2, 1.3, 1.0, 0.8 and 0.6. The gear ratios are rounded to the first decimal place.

To evaluate the impact of the model differences discussed in Sect. 2, the base conventional vehicle is simulated with both models. The fuel consumption (input energy) and its corresponding composition of energy usage or losses respectively (output energy) in the WLTC are presented in Fig. 6. The conventional vehicle in the optimal dimensioning simulation has a WLTC fuel consumption of 5.30 l/100 km while the prototype dimensioning simulation has a fuel consumption of 5.53 l/100 km. This corresponds to a difference of 4.3 % between the two models. The energy input—chemical energy in the form of petrol—in the energy balance shown in Fig. 6 is presented for one WLTC with the bar on the left plus the delta SOC of the 12 V on-board power supply for both the optimal and prototype simulation model. The corresponding balanced energy usage (output) for overcoming the driving resistances, the auxiliary supply (Aux) and the losses in the energy conversion of the ICE, in the gearbox and the mechanical brakes are shown on the corresponding right bar. Energy losses due to ICE and EM dragging, within the clutch, tire slip, EM and battery are summarised within one bar on the top. Reason for this is the low energy loss amount compared to the other losses. The main differences are the energy losses of the ICE, the mechanical brakes and the gearbox which lead to different fuel consumptions between the two simulation models. These losses are analysed in detail in the following.

For the ICE losses Fig. 7 is shown for further discussion. The points in Fig. 7 illustrate the energetically weighted parts of the ICE operating points in the engine map with the optimal concept and prototype simulation. The backwards-simulation results stem from defined operating points within the WLTC. In contrast to this, the prototype simulation results depend on clutch engaging and the driver load.

Table 1 Simulation model differences between the concept and the prototype dimensioning simulation tool

Parameter	Optimal Concept	Prototype
Aggregate modeling	Map based	Detailed + Map based
Transmission losses	Simplified	Map based
Simulation frequency	1 Hz	1000 Hz
Simulation time	Low (>1000/day)	Medium (100/day)

Table 2 Vehicle parameters of the basis conventional front-wheel drive D-Segment vehicle

Parameter	Unit	Value
m	kg	1508
$c_d \cdot A$	m^2	0.64
f_r	–	0.008
F_{Fric}	N	40
r_{dyn}	m	0.308
P_{ICE}	kW	110
$P_{\text{Alternator}}$	kW	2
$t_{0-100\,km/h}$	s	9.1

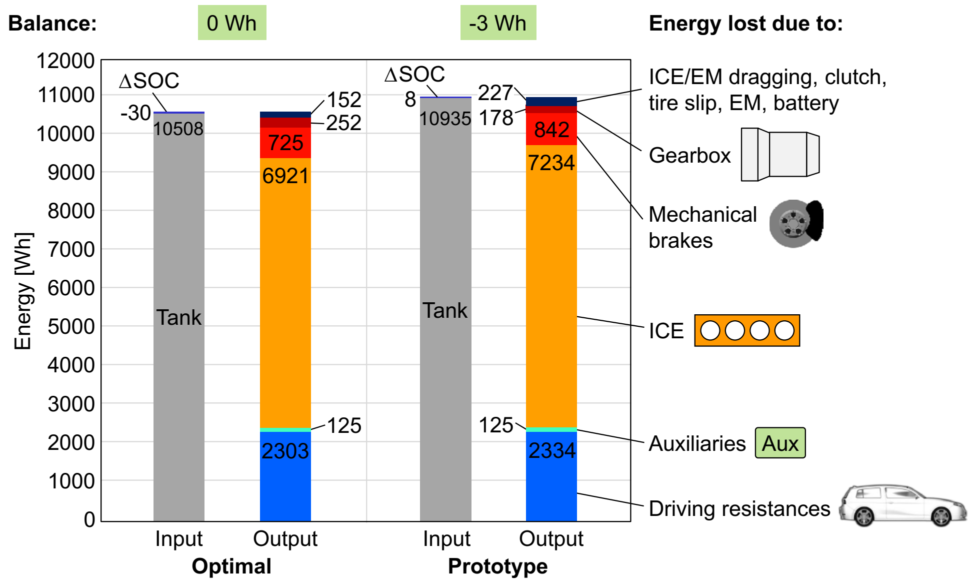


Fig. 6 Energy balance for the conventional vehicle with the optimal concept and prototype dimensioning model for one WLTC simulation

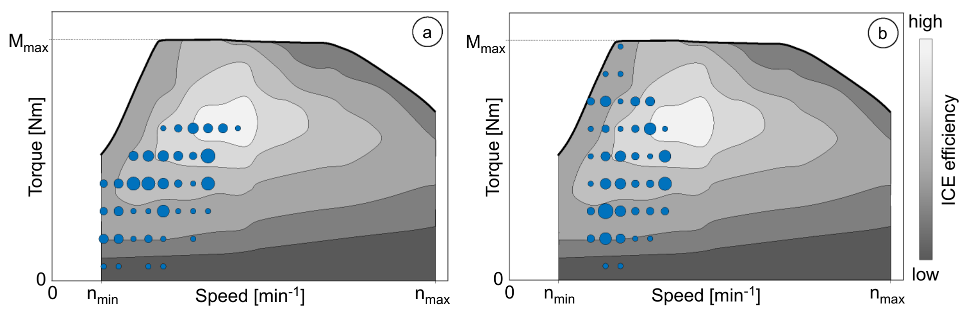


Fig. 7 Energetically weighted parts of the ICE operating points of the conventional vehicle in the optimal concept (**a**) and prototype simulation (**b**) for one WLTC simulation

In order to follow the WLTC, the driver reacts by pressing the accelerator pedal and thus setting the load of the ICE. Depending on the current vehicle speed differences—especially in acceleration phases and launching events—the ICE is operated close to its nominal power. This results in higher mechanical output power compared to the optimal concept simulation. In combination with the efficiency map shown in Fig. 7, this ultimately results in higher overall losses and lead to a higher fuel consumption.

Further loss differences result in the gearbox losses. Reason for this is the simulation model approach. While the optimal concept dimensioning uses a linear loss approach, the gearbox losses in the prototype simulation model are based on lossmaps.

The losses due to conversion of mechanical into thermal energy caused by mechanical braking are different in the two simulation models. While operating points and braking phases are defined in the backwards-simulation, the forward-simulation calculates the braking phases based on the driver input. In deceleration phases, the driver pushes the braking pedal depending on the difference between vehicle and target speed in the WLTC. This results in higher braking power in the first seconds of braking phases due to the drift to the target speed than in the optimal simulation.

Due to these differences between the two simulation models the fuel consumption results differ by 0.23 l/100 km. Additionally, Fig. 6 shows no calculation or simulation errors which could lead to wrong input or output energy calculations. For the optimal simulation the difference between input and output energy is 0 Wh. This means the calculated input energy in form of tank energy (petrol) is exactly the sum of all output energy values. For the prototype simulation, the difference is – 3 Wh. This minor difference is due to negligible rounding and interpolation within the simulation model.

4 Optimal 48 V hybrid drivetrain

The conventional vehicle used in Sect. 3 is the basis for hybridization with the 48 V P0+P4 hybrid drivetrain. The P0 is a belt starter generator directly connected to the crank shaft of the ICE and the second EM is at the rear axle as a central drive. Fig. 8 shows the basic layout of the powertrain.

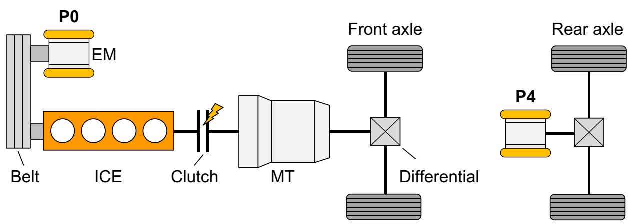


Fig. 8 48 V P0+P4 hybrid drivetrain topology

The one gear transmission at the rear axle has no disconnect clutch.

The aim of the hybridization is to improve fuel efficiency and performance compared to the basis conventional vehicle. In terms of performance, the requirement for this concept study is to achieve the same or better performance than the model variant with higher ICE power. For this, various catalogue values of typical D-Segment vehicles can be taken as basis. For example, the Audi A4 35 TFSI with an ICE power of 110 kW accelerates from 0 to 100 km/h in 8.9 s compared to simulated 9.1 s of the basis conventional vehicle described in Sect. 3 [28]. The sportier variant Audi A4 40 TFSI with an ICE power of 140 kW accelerates from 0 to 100 km/h in 7.3 s [29]. Thus, the minimum requirement for the P0+P4 hybrid drivetrain is to have the same or better acceleration than the 140 kW variant with an acceleration time from 0 to 100 km/h of at least 7.3 s. In addition, to comply with highway cycles combined with no need of a disconnect clutch for the EM at the rear axle due to speed limits of the EM, the maximum vehicle speed is defined to be at least 180 km/h.

For the parameter variation the hybrid components are varied in order to find the best concept. The different parameters are displayed in Table 3. With a step size of 5 kW the electric peak power of the P0 EM varies from 5 to 15 kW. The smallest option of 5 kW is considered to realise necessary charging only while standing. The highest value of 15 kW is chosen to enhance the P0 functionalities like load point shifting and additional recuperation in thrust phases [5, 9]. The peak power of the P4 EM varies from 10 to 30 kW. The maximum value for the electrical power of 30 kW is due to the extra effort with higher electrical power (see Sect. 1). The impact of changing EM power regarding the weight is considered by model based mass calculations [12]. The 48 V battery capacity varies from 0.5 to 2 kWh with four variants.

Table 3 Parameter variation for the 48 V hybrid drivetrain analysis

Parameter	Unit	Lower Value	Step	Upper Value
$P_{P0,\max}$	kW	5	5	15
$P_{P4,\max}$	kW	10	5	30
E_{Battery}	kWh	0.5	0.5	2.0
Ratio e-axle	–	8	1	16

At this point, the weight impact of various battery capacity is considered by model based mass calculations as well [12]. To realise different battery capacities, a reference battery with 0.5 kWh and a peak discharge power of 20 kW and charge power of 21 kW is assumed. By upscaling the battery capacity, the peak power is scaled according to the scaling factor of the battery capacity as well. Thus, the combination of a P0 and a P4 EM with high electric power and low battery capacity with low performance capability leads to the fact that the battery power is not sufficient to operate both EMs at the same time. In contrast to this, the battery power with capacities at or above 1.5 kWh are sufficient to operate both EMs with all regarded performances. The belt drive ratio for the P0 EM is not varied to not increase simulation time extremely and is set to 3.1. The ratio variation for the e-axle is applied for the one gear transmission. For the two speed transmission only the second gear ratio is varied. The first gear is always dimensioned according to the traction limit of the rear axle and a maximum spread requirement to ensure no traction force interruption during a full load acceleration. The maximum spread $\varphi_{\max}$ is defined as the maximum speed $n_{\max}$ divided by the nominal speed n_N

$$\varphi_{\max} = n_{\max}/n_N. \tag{2}$$

The additional weight of the hybrid components and the additional mechanical components of the electrified rear axle ranges for the defined parameter space between 60 and 85 kg. Fig. 9 shows the WLTC consumption and the hybrid acceleration time from 0 to 100 km/h for the variants of the parameter variation with the complementary pareto fronts. In general, the P0+P4 reduces the fuel consumption due to the hybrid functionalities and improves the performance because of the e-axle and the higher system power. The change from one gear to two gear e-axle transmission

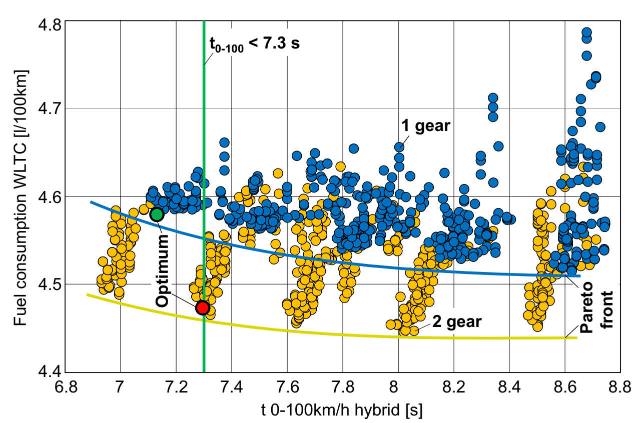


Fig. 9 WLTC fuel consumption and hybrid acceleration time from 0 to 100 km/h for different variants of 48 V P0+P4 with one gear and two gear transmission with the optimal concept dimensioning

improves the performance and the cycle efficiency. Due to the addition of a second gear which enables to fulfil the maximum speed requirement, the first gear ratio can be higher and allows for higher torques. The efficiency improvement mainly results from better operation points of the EM at the electric axle. The variants which fulfil the requirement of an acceleration time in at least 7.3 s and provide the lowest fuel consumption are the optimum variants for each topology and listed in Table 4. Both variants fulfil the maximum vehicle speed requirement, the two gear transmission allows even higher maximum speed due to the second gear and the dimensioning of the gear ratios. Costs play an important role at this point. The 48 V electrification is oftentimes integrated as a low cost electrification [1]. However, compared to other hybrid topologies, e.g. a P2 hybrid system, the P0+P4 hybrid has a higher cost impact per fuel saving [5, 30]. The high performance improvement due to the additional electric rear axle of the P0+P4 hybrid compared to other 48 V hybrid topologies is one of the main reasons it was chosen for the analysis in this paper. For a front wheel driven vehicle, the performance increase of a P4 system is higher than the performance increase with a P2 hybrid system, especially in low speed areas since the maximum powertrain torque is not limited due to the traction limit of the front axle. Thus, the chosen P0+P4 hybrid system represents a good compromise of higher costs in combination with higher performance improvement compared to other 48 V hybrid systems. Based on the fuel consumption of the investigated vehicle the fuel reduction with the one gear transmission is 13.6 % whereas the two gear transmission achieves a reduction of 15.6 %. At this point it is essential to take the costs factor and technical effort for the fuel saving and performance increase into account. The two gear transmission costs are about twice as high as the one gear transmission costs [31]. Compared to the one gear transmission, the two gear transmission is disadvantageous regarding integration. The integration at the rear axle is a higher technical challenge due to limited installation space. Thus, there is a higher chance the two gear transmission does not fit at the rear axle. Additional challenges for the two gear variant are higher development efforts and increased efforts for the operation and shifting strategy since comfort criteria have to be taken into account

Table 4 Important result parameters of the optimal concept dimensioning simulation

Parameter	Unit	One gear	Two gear
$t_{0-100\,\mathrm{km/h,hybrid}}$	s	7.13	7.30
$v_{\max}$	km/h	180	220
E_{WLTC}	l/100 km	4.58	4.47
$\eta_{\mathrm{EM,average,WLTC}}$	%	84.49	85.96
Relative e-axle costs	%	100	200

as well, e.g. torque decline during shifting processes. All in all, the higher fuel savings potential of the two gear variant do not justify the comparatively higher costs as well as technical development and application effort combined with additional installation space. Thus, the two gear transmission is not further investigated and the one gear solution with a fuel consumption of 4.58 l/100 km is chosen for the following evaluation. This variant consists of a P0 EM with an electric power of 5 kW, the P4 EM with 30 kW, a 48 V battery capacity of 1.5 kWh and the overall one gear transmission ratio of 12. The additional weight of the complete hybrid system with the additional mechanical components at the rear axle is +70 kg compared to the conventional vehicle.

5 Prototype 48 V P0+P4 analysis

The results for the prototype simulation for the 48 V P0+P4 hybrid drivetrain are shown and evaluated in the following. First, the WLTC results are compared to the optimal concept dimensioning results shown in Fig. 10. As stated in Sect. 4, the WLTC fuel consumption of the P0+P4 hybrid drivetrain is 4.58 l/100 km for the optimal concept simulation with a fuel saving regarding the conventional vehicle of 13.6 %. The prototype simulation results in a fuel consumption of 4.71 l/100 km which corresponds to a saving of 14.8 %. As Fig. 10 illustrates, the P0+P4 fuel consumption differs about 2.8 % between the two simulation models. All in all, the fuel consumption saving is higher with the prototype simulation than with the optimal simulation. Mainly, this is a result of the significantly higher fuel consumption of the conventional vehicle simulated with the prototype simulation model.

In the following, the P0+P4 result differences are discussed in order to explain the fuel consumption differences. The corresponding energy balances are shown in Fig. 11. The input energy—chemical energy in the form of petrol—is presented for one WLTC with the bar on the left plus the delta SOC of the 12 V on-board power supply combined with the 48 V battery SOC start and end difference for both simulation models. The energy usage (output) for overcoming the driving resistances, the auxiliary supply (Aux) and the losses of the energy conversion of the ICE and EM, in the gearbox and in the battery are shown on the corresponding right bar. Energy losses due to ICE and EM dragging, within the clutch, tire slip and mechanical brakes are summarised with one bar on the top. Compared to the conventional vehicle the mechanical brake losses due to conversion of mechanical into thermal energy are almost zero. This is because almost all braking events are processed by the EM by recuperation. Slight differences between the summarised losses are due to clutch losses. The balanced driving resistances of the P0+P4 hybrid vehicle are 35–36 Wh higher than of the conventional vehicle in both simulation

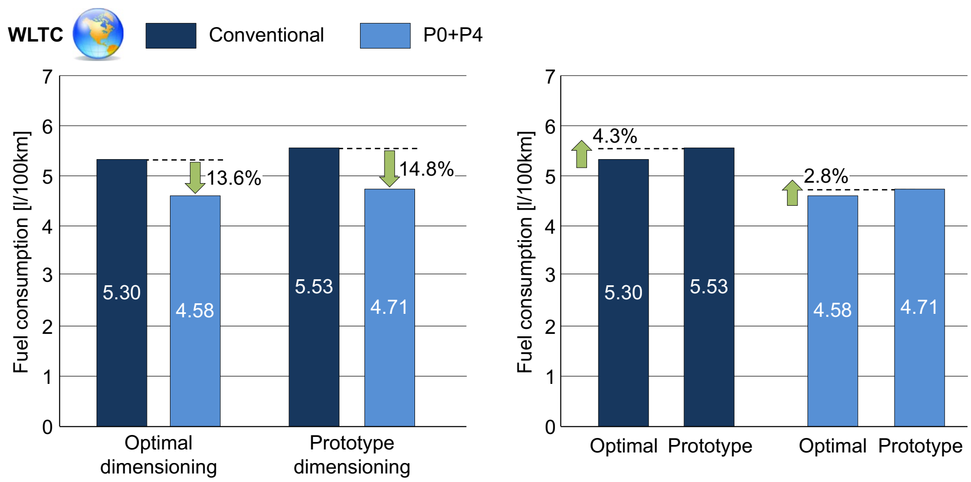


Fig. 10 Comparison of fuel consumption for the conventional and 48 V P0+P4 hybrid drivetrain with the optimal concept and prototype simulation model

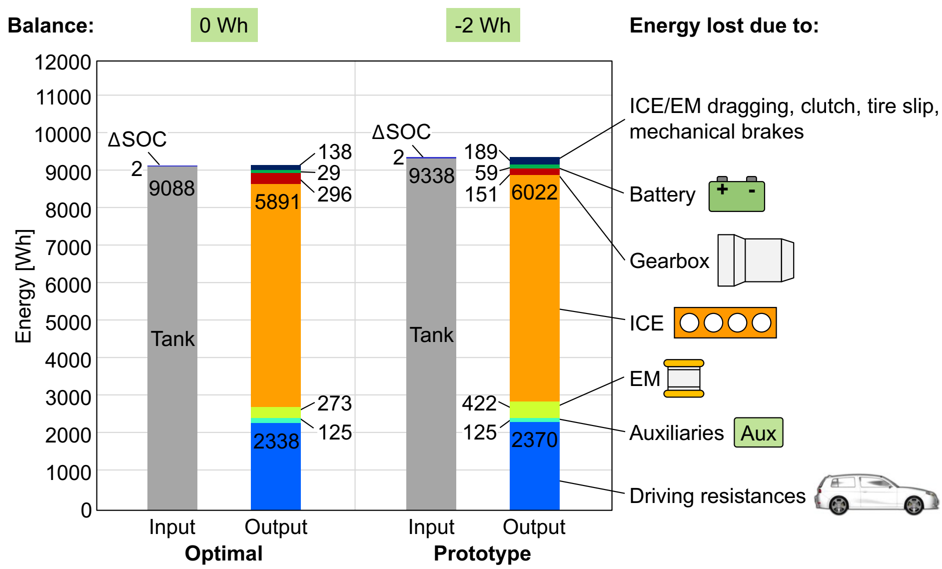


Fig. 11 Energy balance for the 48 V P0+P4 hybrid drivetrain with the optimal concept and prototype dimensioning model for one WLTC simulation

models. This value is due to the weight increase of 70 kg for the hybrid vehicle and represents the increase of resistances due to the rolling resistance. The additional mass also affects the acceleration resistance. Since Fig. 11 shows the balances of the driving resistances, the mass effect on the acceleration resistance does not affect this bar as it is the balance of traction and thrust phases. The three main loss differences which lead to differences in fuel consumption between the two models are due to ICE and EM conversion as well as gearbox losses. The argumentation for higher gearbox losses within the prototype simulation is valid as stated in Sect. 3. To explain and illustrate the differences in usage of the ICE between the two simulation models, the energetically weighted parts of the ICE operating points in the engine map are presented in Fig. 12 for the optimal concept and prototype simulation. For the 48 V P0+P4 hybrid drivetrain, electric energy is used for substitution of ICE operating points by driving in electric mode. Especially ICE operating points with bad efficiency are substituted. The electric energy results from the recuperation of energy in braking phases and from load point shifting. To guarantee SOC neutrality, load point shifting is used so the ICE's load point is shifted to better specific fuel consumption. This improves the ICE efficiency, the battery is loaded and ultimately the overall system efficiency is enhanced. All these aspects lead to a fuel consumption reduction. In contrast to the results discussed in Sect. 3, the energy turnover is shifted for both simulation models from operating points with low efficiency towards higher efficiency and higher load areas. Low efficiency areas are substituted by powering the vehicle in electric driving mode. Furthermore, the energy turnover is mainly focused towards the best efficiency area of the engine map. This leads to an increase in overall engine efficiency compared to the ICE energy turnover in the conventional vehicle. Two main differences lead to higher energy turnover in worse efficiency areas and thus higher fuel consumption

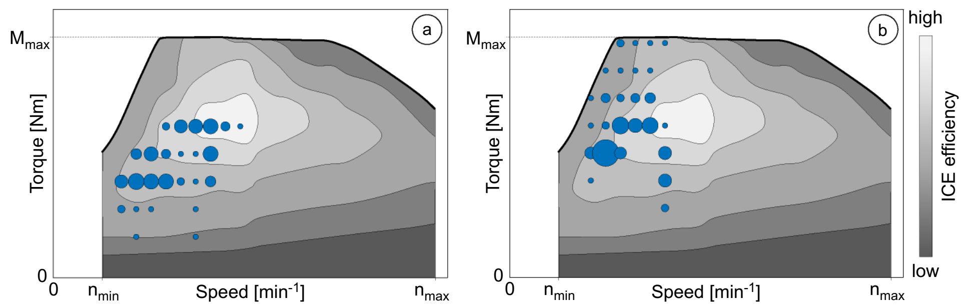


Fig. 12 Energetically weighted parts of the ICE operating points of the 48 V P0+P4 hybrid drivetrain in the optimal concept (**a**) and prototype simulation (**b**) for one WLTC simulation

with the prototype simulation compared to the optimal concept simulation. First, the ICE is operated close to the full-load curve. Explanations for this behavior have already been given in Sect. 3. Secondly, the optimal concept simulation operates the ICE more often in its best efficiency area due to the ECMS operating strategy. The realistic rule-based strategy within the prototype simulation operates the ICE based on the driver input in terms of load requirements. These base on the current difference of the vehicle speed compared to the target value within the WLTC. This ultimately leads to ICE operating points moving away from best efficiency points towards higher load in combination with low engine speed.

Losses in the EM are explained with the energetically weighted parts of the P4 EM operating points shown in Fig. 13 comparing the optimal concept and prototype dimensioning simulation. At this point, the fixed gear ratio between the wheels and the EM means that the EM speed is proportional to the vehicle speed. Hence, at higher vehicle speed the EM can only be operated in lower efficiency areas. The EM characteristic curve and efficiency map is based on the EM presented in [26] and [27] with an EM control unit limiting the motor torque at a certain EM speed to zero. This is realised to avoid high losses and thus a high thermal impact on the EM in combination with a high discharging power of the 48 V battery. Negative torque is without any limitation as the losses at higher EM speed are tolerated to exploit the maximum of the recuperation potential. The optimal concept simulation with the ECMS operating strategy calculates the optimal cost efficient operating points of the ICE and EM together. This leads to energetically weighted parts of the EM operating points mostly in areas of highest efficiency. Higher electric power of the EM is avoided to not discharge the battery too fast as it is cost inefficient in terms of the ECMS. In contrast to this, the realistic rule-based operating strategy in the prototype simulation uses the electric power of the EM depending on the load requirement and the SOC. If the SOC is high enough and if the EM power can meet the load requirement for a particular driving situation, the operating strategy decides to accelerate or decelerate the vehicle by the EM and not by the ICE or the mechanical brakes. This leads to energy turnovers in areas of higher load up to the full-load curve of the EM compared to the optimal simulation. Ultimately, the higher output power in areas of comparatively lower efficiency lead to higher EM losses within the prototype simulation.

With using the prototype simulation model the 3D customer based cycles are simulated with the conventional and the P0+P4 vehicle. The simulation is performed for urban, extra-urban and highway areas for the three different driving behaviors mild, average and sporty. Depending on the driving environment the target vehicle speed profile differs. For example, the target vehicle speed is higher in the highway area than it is in the urban area. The driver has to comply with the target vehicle speed by interacting with the vehicle by pedals and selector lever. The three driving behaviors impact the accelerator and brake pedal gradients and end positions as well as engine shifting speeds. For example, the accelerator gradients and end positions of the sporty driver are significantly higher compared to the mild driver. This essentially impacts the loads of the powertrain aggregates. The results regarding the fuel consumption and savings

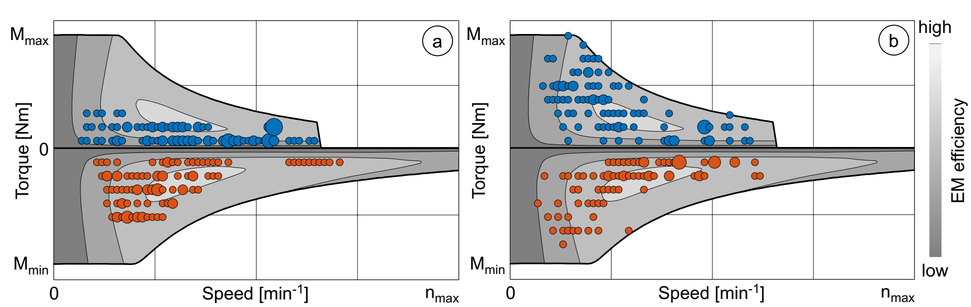


Fig. 13 Energetically weighted parts of the EM operating points of the 48 V P0+P4 hybrid drivetrain in the optimal concept (**a**) and prototype simulation (**b**) for one WLTC simulation

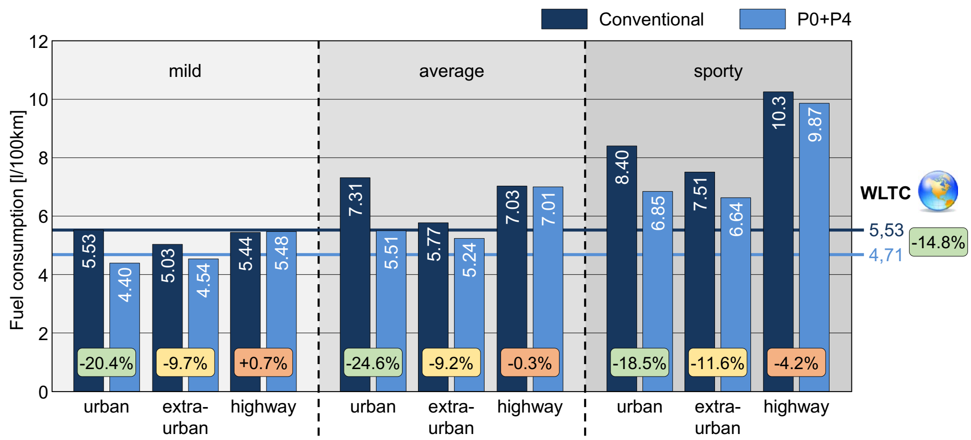


Fig. 14 Fuel consumption of the 48 V P0+P4 hybrid drivetrain compared to the basis conventional vehicle in customer based cycles with the prototype simulation model

potential by the P0+P4 are shown in Fig. 14. For comparison with the WLTC, its consumptions are illustrated with horizontal lines. The highest fuel savings potentials of up to 24.6 % can be achieved in urban areas due to high amounts of launching and braking events which can be realised by the EM. At this point, the driving requirements are crucial for the fuel savings potential. Low and high driving requirements lead to lower savings potential than average driving requirements. Low ones lead to EM operating points with low efficiency whereas high requirements cannot be met by the electric power of the EM being dimensioned too low. The second best savings potentials can be achieved in extra-urban areas with savings of up to 11.6 %. Generally, the driving requirements are higher than in urban areas due to higher vehicle speed. Taking speed limits of the EM into account—upper threshold whereby the EM cannot be operated above—in combination with higher driving requirements so that the EM cannot operate the vehicle in electric driving mode anymore lead to lower fuel savings potential. In highway areas the P0+P4 vehicle only reduces the fuel consumption by few percentages. This is due to higher vehicle speed and driving requirements. The argumentation stated for the extra-urban driving environment is valid for highway areas as well. With mild driving style the 48 V hybrid drivetrain increases the fuel consumption in highway areas by 0.7 % due to low recuperation potential, higher driving requirements and additional system weight.

6 Summary and outlook

In this paper, a virtual toolchain for the optimal concept and prototype dimensioning of 48 V hybrid drivetrains has been presented. In particular, a 48 V P0+P4 hybrid drivetrain has been chosen to dimension the optimal concept of this drivetrain and to make realistic statements regarding fuel consumption in the WLTC and customer based cycles. For this, a virtual toolchain was presented at the beginning. Subsequently, the simulation models for the optimal concept dimensioning and the prototype dimensioning were presented. After that, the simulation models were analysed and compared with each other for the basis conventional vehicle. It was shown that the difference of simulated fuel consumption between the two models is 4.3 %. Then, the optimal concept dimensioning process was carried out which ultimately led to one optimal 48 V P0+P4 hybrid drivetrain. This optimal variant reduces the fuel consumption by 13.6 % compared to the conventional vehicle. The P0+P4 was simulated with the prototype simulation model afterwards showing a fuel savings potential of 14.8 % which differs about 2.8 % compared to the optimal concept result. An energy turnover for the analysis of losses in the ICE and EM has shown that the prototype simulation leads to a more dynamic operation of these components. This results out of the usage of a broader area in the characteristic maps. The prototype simulation results for the customer based cycles have shown a fuel savings potential of up to 24.6 % in urban areas due to high amounts of launching and braking events. Extra-urban and highway areas show a fuel consumption reduction of up to 11.6 % and 4.2 %, respectively due to higher vehicle speed and power requirements. The presented virtual toolchain can be used to combine the optimal concept dimensioning with realistic behavior simulations to maximise realistic statements and minimize time effort.

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Compliance with ethical standards

Conflict of interest The authors declare that they have no conflict of interest.

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