Optimization-based Component Sizing Method for Electrified Heavy-Duty Powertrain Concepts

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Abstract. The demand for battery-powered, exhaust emission-free mobile machinery is increasing. However, the high energy requirements of mobile machines result in large, heavy and expensive battery storage systems. Driven by continuously decreasing but still fairly high battery costs, the efficiency of the drive system is becoming the focus of development activities, even in an industry where robustness, service lifetime, productivity and power density have been the driving factors up to now. Interactions between topology, component sizing, and control strategy require a system-level approach for the development of drive systems. In this paper, a novel optimization-based sizing method for electrified heavy-duty powertrain concepts is presented and applied to subsystems of a wheeled excavator. Scalable component cost and efficiency models are used in a system simulation to calculate objective function values for a multi-objective optimization. The result of the optimization, a trade-off between energy consumption and costs provides the OEM with a decision support in the development process. The modular structure of the approach allows a subsequent extension of the design parameter space as well as the detailing of the models used.

1. Introduction

In reaction to the ongoing climate change and the existential threat for Europe and the world, the European Union has agreed on the Green Deal. Part of the Green Deal is the goal of becoming the first climate-neutral continent and releasing no net greenhouse gas emissions by 2050 [1]. In addition, ongoing urbanization contributes to increasing exceedance of current emission limits in inner-city areas. In order to comply with current and future emission limits, local and regional authorities are planning and already implementing action plans that include the establishment of low- and zero-emission zones in inner-city areas. In this context, the Committee for European Construction Equipment expects increasing restrictions on the use of diesel-powered mobile machinery [2]. In the future, the demand for alternative drive concepts will increase not only in the automotive sector, but also in the field of mobile machinery. One possibility for operating construction machinery in zero-emission zones is the use of battery-powered electrical machines. The use of passenger cars in urban areas is characterized by short distances, short periods of use, low power requirements and numerous opportunities for charging a battery. In comparison, the use of construction machinery requires high power over a long period of time (e.g. an eight-hour workday) without the possibility of recharging in between. As a result, comparatively large amounts of energy have to be stored, which leads in combination with the low energy density of electric accumulators to exceedingly large, heavy and expensive batteries. Therefore,
in the development of battery-electric mobile machines, the energy efficiency of the drive system is becoming much more important than it was the case in the development of conventional Diesel driven machines. Mobile machines are characterized by the interaction of several consumers (e.g. travel drive, cylinder drive and swing drive in hydraulic excavators). The system efficiency and thus the energy requirement of the machine therefore depends not only on the sizing of the components, but also on the choice of a suitable topology and an efficient control strategy. In order to be able to take into account the mutual influence of topology, component sizing and control strategy on the energy requirement of the machine in early phases of the development, it is necessary to develop and evaluate on a system level [3].

For the development of hybrid and battery electric vehicles, optimization-based approaches that focus on minimizing system characteristics, such as fuel consumption or manufacturing costs, have proven beneficial in the automotive sector [4]. Multiple approaches for finding an optimal vehicle topology [5–9], an ideal component sizing [9–16] and an appropriate control strategy [11–14, 17] using several different kinds of optimization strategies, such as Genetic Algorithms, Particle Swarm Optimization, Nested Optimization or extended coordination methods can be found in the literature [4, 18].

Optimization algorithms are also used for development tasks in the field of mobile machinery. For example, approaches for optimizing the component sizing and control strategy of hybrid energy storage systems consisting of batteries and fuel cells or capacitors can be found in the literature [19–21]. Furthermore, there are existing approaches that address topology selection [22] and component specification of electrical components as well as internal combustion engines for hybrid mobile machinery [23–28].

Compared to an automotive application, in which the travel drive is the main consumer, the energy consumption of mobile machinery is composed of the parallel operation of numerous travel and working drives. Therefore, methods from the automotive sector focusing on one single consumer cannot be used directly for the development mobile machine drive systems. However, individual component models used in the automotive sector, such as efficiency and cost models for electric motors and power electronics, can be used for the development of mobile machines. Still, further models are required for the efficiency and cost calculation of hydraulic components and heavy-duty gearboxes, which are not used in the automotive sector. Most studies from the field of mobile machinery have only focused on specific domains, such as the optimization of energy storage systems. Furthermore, there has been no detailed investigation of the optimal hydraulic component or heavy-duty gearbox sizing. To the best of our knowledge, there are no optimization-based development approaches for battery-electric drive concepts (travel and work drive) of mobile machinery. To answer the question of how an efficiency- and cost-appropriate drive concept consisting of topology, component sizing and control strategy can be found for battery-electric mobile machines, a novel method is required that uses a system model capable for the parallel operation of multiple consumers as well as performant models for heavy-duty components, which can be used by well-established optimization algorithms.

In this paper, a novel method for optimization-based component sizing is presented and exemplarily applied to a wheeled excavator using a preselected topology. For this topology, a multi-objective optimization using scalable component models is performed to identify Pareto-optimal component design parameters. The questions of topology synthesis and an optimal control strategy for the specified components will be covered in a future work. The presented method is characterized by a combination of a computationally intensive and therefore for optimization methods unsuitable transient dynamic simulation, which makes it possible to consider the interaction of parallel operated consumers in the component sizing, and a performant quasi-static simulation, which is necessary to apply known optimization methods for the component sizing. The quasi-static simulation uses interchangeable component models that can be extended in the level of detail. The model level of detail can be adapted according to the data availability, so that the method can be used by OEMs in early phases of the product development for decision making.
2. Use Case and Scope

2.1 Considered System

This paper investigates an electrified drive concept for a wheeled excavator. It is based on a conventional drive system (Figure 1, left) for a wheeled excavator with an operating weight of 12.5 t to 17.5 t. A 105 kW diesel engine drives a variable displacement pump that supplies a state-of-the-art load-sensing (LS) system. The LS system supplies five consumers: one hydraulic motor each for the travel drive of the undercarriage and the slewing drive of the superstructure (SS), as well as hydraulic cylinders for the bucket, stick and a boom. The motion of the superstructure is transmitted by a swing reducer (SR). At each consumer valve, the load pressure $p_L$ is signaled to the pump. In the model the two cylinders which actuate the boom are combined into one consumer assuming symmetrical operation. The travel drive is designed as a central drive with a variable displacement hydraulic motor. The working drive comprises the cylinders and the slewing drive. In practice, the travel and the working drive are usually not operated simultaneously. The diesel engine speed remains largely constant during the execution of a work task. Its level is selected by the operator depending on the required power.

The electrified drive topology (Figure 1, right) was conceptualized in a study using discursive methods such as the morphological analysis. Among the defined evaluation criteria, it represents a preferred solution, therefore it was selected for the subsequent optimization which is presented in this paper. The hydraulic travel drive as well as the slewing drive are replaced by electric motors (EM). The travel drive is implemented as four single-wheel drives with planetary hub gearboxes (G). Both drives can thus utilize the high recuperation potential when decelerating the vehicle or the superstructure. In addition to the two rotary consumers, the boom drive has a high recuperation potential. This means that the potential energy can be converted into electrical energy when the boom is lowered. Accordingly, the boom has its own energy supply that enables recuperation. The recuperation potential was derived from measured duty cycles. For reasons of confidentiality, no detailed description of the technical implementation can be given. The stick and bucket drive have a low recuperation potential, so that the drive of these consumers is kept as a conventional LS drive. In addition to recuperation, the new drive concept offers the advantage of dividing the LS system up. In conventional LS systems, consumers operated in parallel with different loads cause throttling losses at the low-load consumers. In the conventional system, the slewing drive and the boom drive are typically the highest-load consumers. In the electrified concept, these throttling losses are thus avoided.

For the sake of brevity, this paper focuses on the component sizing for the travel drive as well the stick and bucket drive of the electrified drive concept (Figure 1, right, highlighted).

![Figure 1. Conventional drive system (left) and electrified drive concept (right) of a mobile excavator.](image)

The subsystems dimensioned in this paper are highlighted.

2.2 Duty Cycle

To ensure consistent performance of the electrified drive concept compared to the conventional drive system, representative operating scenarios of the conventional drive system are used for component dimensioning. These were obtained from measurements for the travel and working drive. For the travel
drive, a combined cycle with driving and grading sections is adopted in this paper. Figure 2, left depicts examples of the excavator speed \( v \) in km \( \cdot \) h\(^{-1} \) and track gradient in ° of a driving section. For the working drive, a 90° full load working cycle is used as a representative cycle. In Figure 2, right the exemplary stick cylinder stroke \( s_s \) in mm and chamber pressures \( p_s \) in bar are shown. In this cycle, the excavator picks up soil at one point and, after a 90° rotation of the superstructure, dumps it onto a truck, for example. The dumping point can be higher or lower than the pickup point.

\[
\text{Figure 2. Excavator velocity and track gradient of a driving section (left) and stroke and pressure of the stick cylinder in the 90° working cycle (right)}
\]

2.3 Design Parameters and Objectives

The approach of this paper is applied to the optimal sizing of the outlined subsystems in Figure 1 while respecting the defined load and motion cycles of the consumers. This concerns the optimal selection of the supply design parameters. Here, the design parameters of the travel drive are the installed EM and power electronics (PE) power as well as the gear ratio of the planetary hub gearboxes. The design parameters of the working drive are the installed EM and PE power and the maximum displacement of the pump. The constant speed operation of the pump is adopted from the conventional system, however, the speed is defined as a design parameter. System costs and energy consumption are considered as the optimization objectives.

3. Materials and Methods

In this section, the proposed approach for optimal component sizing is presented. The workflow of the approach as well as the necessary steps are shown in Figure 3.

\[
\text{Figure 3. Approach for optimization-based supply component sizing.}
\]

On the input side, load and motion cycles (Figure 2) of the consumers are fed into a forward model. These cycles contain the design-relevant operating scenarios of the system and were obtained from measurements in the framework of the presented investigations. Alternatively, consumer cycles can be scaled from similar applications or fully synthesized. The forward model is a transient simulation model.
that represents the topology to be investigated. It is used to calculate the loads and motions of the supply components to be dimensioned while respecting the consumer cycles. The supply component cycles along with the design parameters form the input variables of the backward model. In contrast to the forward model, it is quasi-static and thus more computationally efficient. This property significantly reduces the computational effort as it is evaluated numerously during the following optimization process. It consists of scalable supply component models that can be varied in their dimensions by means of the design parameters. These models describe the costs and the energy consumption of the components based on the design parameters considering the supply cycles. The costs and the energy consumption are the objective functions that are minimized in the following multi-objective optimization. The optimization algorithm iteratively evaluates the values of the objective functions and selects new design parameter sets. After this iteration, optimal component specifications are available.

3.1 Forward Model

The forward model is a 1D dynamic simulation model built in the MATLAB/Simscape [29] environment from the hydraulic and mechanical library. This model of the electrified wheeled excavator assuming ideal power sources is used to determine the supply component load and motion cycles, which are shown in Figure 4 in the form of mechanical and hydraulic loads, from the specified consumer cycles.

In the case of the bucket and stick cylinder drive, an ideal pressure source supplies the system. The pressure of this source is set by the current maximum consumer load and offset by an LS pressure. The cylinder strokes are controlled by proportional valves which are modeled by variable orifices. An operator model, implemented as a PI-controller, transforms the difference between the actual and the desired cylinder stroke into a valve actuation. The so released volumetric flow leads to a pressure buildup in the cylinder chambers, which is considered by the pressure buildup equation. To calculate the cylinder motion, a force balance between pressure and external forces is performed at the cylinders. The external forces consist of inertial forces of the cylinder piston and rod as well as its attachments, friction, digging and weight forces. The sum of the external forces is derived from the cylinder chamber pressure difference in the reference vehicle measurements. The pressure and flow rate provided by the ideal power source is used in the following backward model to identify optimal component specifications of the hydraulic pump, EM, and PE.

In the case of the travel drive the excavator is modeled as a single mass under the influence of propulsion torques from ideal torque sources and resistances from the track gradient, rolling resistances, air drag resistance and external forces (e.g. soil forces during grading). Here the operator model, likewise implemented as a PI-controller, transforms the difference between the actual and the desired excavator velocity into propulsion torques. The wheel torque and speed are used for the component sizing of the single-wheel drive. The supply component load and motion cycles for the travel and working drive determined in the forward model are shown in Figure 4.

![Figure 4. Supply component load and motion cycles. Left: Wheel torque $T_W$ and speed $n_W$. Right: Pump flow rate $Q_P$ and pressure $p_P$.](image-url)
3.2 Backward Model

A backward model consists of scalable component models. Based on the design parameters, it describes the costs and, considering the component loads, the energy consumption of a component taking into account the energy losses. In the optimization process, backward models are evaluated numerously and are therefore modeled in a quasi-static approach to reduce computational effort. Losses are thus described by constant factors, maps or analytical equations. The level of detail of the models is selected according to the authors’ available database (parameters) of the components.

3.2.1 Scalable component efficiency models

Electric motor and power electronics

To evaluate the energy efficiency of the electrical system consisting of EM and PE, a model is used that calculates the electrical energy demand (objective) at a given mechanical load cycle (input from forward model) as a function of the installed electrical power (design parameter). Therefore, a precalculated reference efficiency map for a power class of 60 kW, which takes into account the operation point dependent losses of both components, is used. The map is available for motor operation in the first quadrant and is adopted to the remaining quadrants under the assumption of symmetrical loss behavior. The design parameter of the EMs and PEs is their power. For the scaling of the electrical system, the torque axis of the reference map is stretched or squeezed and thus the loss behavior is adapted to the required power class. By evaluating the mechanical operating points in the efficiency map, the efficiency of the electrical system $\eta_{EM,PE}$ and ultimately electrical power is obtained. The reference map used is shown in Figure 5. The highest overall efficiency is achieved in the low power range.

**Figure 5.** Reference efficiency map of the electrical system consisting of EM and PE depending on the mechanical operating point of the EM.

**Hydraulic pump**

To describe the efficiency of the hydraulic pump, efficiency maps of a complete product series were available to the authors. On the basis of the known efficiency maps, parameters were extracted that enabled the use of a more detailed analytical efficiency model, which is described in the following. According to Equation (1) the total efficiency of a hydraulic pump $\eta_{tot}$ is the product of the volumetric $\eta_{vol}$ and the hydraulic-mechanical efficiency $\eta_{hm}$.

$$\eta_{tot} = \eta_{vol} \cdot \eta_{hm}$$

The volumetric efficiency is the ratio of the effective $Q_{eff}$ and theoretical volumetric flow rate $Q_{th}$ (Equation (2)). The effective volumetric flow is smaller than the ideal volumetric flow by a leakage flow $Q_{leak}$. Here, $\alpha$ is the relative displacement, $V_0$ the maximum displacement in cm$^3$ and the speed of the pump $n_p$ in rpm.

$$\eta_{vol} = \frac{Q_{eff}}{Q_{th}} = \frac{Q_{th} - Q_{leak}}{Q_{th}} = \frac{\alpha \cdot V_0 \cdot n_p - Q_{leak}}{\alpha \cdot V_0 \cdot n_p}$$
The leakage flow $Q_{\text{leak}}$ in $\text{m}^2 \cdot \text{s}^{-1}$ is calculated using Equation (3). $K_{HP}$ is the Hagen-Poiseuille coefficient in $\text{m}^3 \cdot \text{s}^{-1} \cdot \text{Pa}^{-1}$ and $\Delta p$ the pressure difference of the pump ports in Pa.

$$Q_{\text{leak}} = K_{HP} \cdot \Delta p \quad (3)$$

The hydraulic-mechanical efficiency is the ratio of theoretical $T_{th}$ and effective torque $T_{eff}$ at the pump shaft (Equation (4)). The effective torque is higher than the theoretical torque by a friction torque $T_{\text{fric}}$.

$$\eta_{hm} = \frac{T_{th}}{T_{eff}} = \frac{T_{th}}{T_{th} + T_{\text{fric}}} = \frac{\alpha \cdot V_0 \cdot \Delta p}{2 \cdot \pi \cdot T_{\text{fric}}} \quad (4)$$

The total frictional torque $T_{\text{fric}}$ in Nm is composed of a constant component $T_0$ and a pressure-dependent component $T_p$ (Equation (5)). Here, $K_{TP}$ describes the proportionality factor for determining the pressure-dependent component in $\text{N} \cdot \text{m} \cdot \text{Pa}^{-1}$.

$$T_{\text{fric}} = T_0 + T_p = T_0 + K_{TP} \cdot \Delta p \quad (5)$$

A parameter estimation approach is used to evaluate five efficiency maps of a pump model series and fitted the parameters of the loss model described above to satisfy the efficiency maps. The parameters obtained are depicted in Figure 6 as a function of the corresponding maximum displacement. In the optimization, parameters of pump sizes which are located between the available data are determined by interpolation using neighboring points. Pump sizes beyond the boundary values are not considered. The model allows the loss behavior of hydraulic pumps to be continuously scaled using the displacement as a design parameter.

**Figure 6.** Parameters of the efficiency model of a pump series as a function of the maximum displacement volume. Left: $K_{HP}$ and $K_{TP}$. Right: $T_0$.

The efficiency model is used to calculate the mechanical operating points of the connected electric motor based on the hydraulic operating points determined in the forward model. Figure 7 shows an exemplary overall efficiency map of a 200 cm$^3$ pump at 1,800 rpm which was derived from the analytical model. The highest overall efficiency is achieved at maximum displacement and a mid-range pressure level.

**Figure 7.** Exemplary overall efficiency map of a 200 cm$^3$ pump at 1,800 rpm
Planetary hub gearbox

The design parameter of the gearbox is the gear ratio. Thus, a model is required that determines the efficiency of the gearbox over the selected ratio. In a first approach, the efficiency of the gearbox is modeled by a fixed efficiency depending on the selected gear ratio. For an economic design of gearboxes, Niemann, Winter [30] identified a maximum ratio of 1s = 6 for a single-stage, 1s = 35 for a two-stage and 1s = 150 for a three-stage gearbox. For each gear stage, an efficiency of ηPl = 0.981 is assumed for a planetary gearbox at its rated operating point [30]. This results in the equation shown in Equation (6) for the efficiency of the gearbox ηG. More detailed models to calculate the gearbox component and operating point specific losses such as bearing and gear losses both load dependent and independent can be used in future works [31].

\[
η_G = \begin{cases} 
η_{Pl} & \text{if } 1_s \leq 6 \\
η_{Pl}^2 & \text{if } 6 < 1_s \leq 35 \\
η_{Pl}^3 & \text{if } 1_s > 35 
\end{cases}
\]  

(6)

3.2.2 Scalable Cost Models

Electric motor, power electronics and hydraulic pump

The costs of EM C_EM, PE C_PE and hydraulic pumps C_Hyd are described in € using linear approaches according to Equations (7), (8) and (9). The costs of EM and PE result from the product of their powers P_EM and P_PE in kW and the respective constant cost factors k_EM and k_PE in € · kW⁻¹. The costs of hydraulic pumps scale with the constant cost factor k_Hyd in € · cm⁻³ and the maximum displacement V_0 in cm³. The cost factor of EM and PE is assumed as k_EM = 15 € · kW⁻¹ and k_PE = 16 € · kW⁻¹ respectively according to [22]. For the combination of EM and PE, other sources give similar numerical values (30 € · kW⁻¹ [11], [12]). Due to confidentiality, the cost factor for the hydraulic pumps is omitted.

\[
C_{EM} = k_{EM} \cdot P_{EM}
\]  

(7)

\[
C_{PE} = k_{PE} \cdot P_{PE}
\]  

(8)

\[
C_{Hyd} = k_{Hyd} \cdot V_0
\]  

(9)

Planetary Hub Gearbox

For the calculation of the gearbox costs, there are approaches in the literature that are based on the gearbox mass [32] as well as approaches that are based on the number of required components [33]. In order to be able to consider different gearbox costs within a selected number of gear stages depending on the transmitted torque, a mass-based approach is chosen in the context of this contribution. To estimate the gear mass, a mass factor f_g, mass is determined which allows the mass of a reference gearbox to be scaled to other gear ratios. This mass factor was derived in [34] and used in [25] for a single-stage planetary gearbox. It can be determined according to Equation (10) where N is the number of planets and i_pl is the overall gear ratio of a single planetary gear stage. The derivation of the equation is based on the assumption of a maximum allowable surface durability factor of the gears [34].

\[
f_{G,\text{mass}} = \frac{1}{N} + \frac{2}{N \cdot (i_{pl} - 2)} + \frac{i_{pl} - 2}{2} + \frac{(i_{pl} - 2)^2}{(i_{pl} - 2)^2} + \frac{0.8 \cdot (i_{pl} - 1)^2}{N \cdot (i_{pl} - 2)} + \frac{0.4 \cdot (i_{pl} - 1)^2}{N}
\]  

(10)

For a reference gearbox, the mass factor can be used to scale the gearbox mass according to Equation (11), where i_Refer is the overall gear ratio of the reference gearbox.

\[
M_{G,\text{mass}|i=i_{pl}} = \frac{f_{G,\text{mass}|i=i_{pl}}}{f_{G,\text{mass}|i=i_{Refer}}} \cdot M_{G,\text{mass}|i=i_{Refer}}
\]  

(11)

The mass estimation of a single-stage planetary gearbox is not sufficient for a planetary hub gearbox and was therefore extended to two- and three-stage planetary gearboxes. For the extension to a two- or three-stage gearbox, the total gear ratio is assumed as equally divided among all stages. A separate mass factor is determined for each stage. The additional ratio per stage and the additional mass per stage are
used from catalogue references of a planetary hub gearbox manufacturer [35]. The parameters used for the reference gearboxes are summarized in Table 1.

| Parameter                  | First Stage | Second Stage | Third Stage |
|----------------------------|-------------|--------------|-------------|
| Additional Gear Ratio      | 6.09        | 6.19         | 4.51        |
| Additional Weight          | 30 kg       | 56 kg        | 39 kg       |

A relative weight cost of 14.25 € · kW⁻¹ according to [32] is assumed. The nonlinear shape of the cost function for two- and three-stage planetary hub gearboxes is shown in Figure 8. The costs of a gearbox $C_G$ are determined by an evaluation of the cost function for a given gear ratio.

![Figure 8. Cost function for planetary hub gearboxes](image)

Gearbox costs are calculated by estimating the gear mass. In addition to the pure gearbox mass, other factors such as the number of gear box components installed have an influence on the costs. Taking into account the number of gearbox components would result in a different cost leap between a two-stage and a three-stage gearbox design. The three-stage gearbox has less gear mass due to smaller gear diameters, but more components. The interchangeability of the component models allows a more detailed cost model for the gearbox to be added at a later point in time and the influence on the component specification to be investigated.

### 3.2.3 Model Structure

The scalable component efficiency and cost models are combined to backward models of the travel and working drive subsystems (Figure 9). The model inputs are the precalculated supply component cycles ($T_W(t)$, $n_W(t)$ and $p_p(t)$, $Q_p(t)$) and the design parameters ($i$, $V_0$, $P_{EM}$, $P_{PE}$). To expand the solution space the constant pump speed $n_p$ is treated as a design parameter as well. The model calculates and outputs the costs ($C$) and the electric energy consumption ($E_{El}$) of the subsystem.

![Figure 9. Structure of travel (left) and working drive (right) backwards models](image)

The costs of a subsystem are the sum of the component costs (see Section 3.2.2) and calculated using Equation (12).
The consumption of electrical energy is determined by integration of electrical power, which is given by Equation (13).

\[ E_{El} = \int P_{El}(t) \, dt \quad (13) \]

The electrical power is evaluated along the efficiency path (see Section 3.2.1) of the respective subsystem (Equations (14) and (15)). \( T_P \) and \( n_P \) are the torque and speed of the pump shaft respectively.

\[ P_{El,\text{travel}}(t) = \eta_{EM,PE}(T_W, n_W) \cdot \eta_g \cdot 2\pi \cdot T_W \cdot n_W \quad (14) \]

\[ P_{El,\text{working}}(t) = \eta_{EM,PE}(T_P, n_P) \cdot \eta_P(p_P, Q_P) \cdot p_P \cdot Q_P \quad (15) \]

### 3.2.4 Multi-Objective Optimization

To find an optimal component specification, a multi-objective optimization problem is formulated. \( J \) is the multi-objective optimization function consisting of the costs and the electrical energy consumption of the respective drive.

\[ J = [C, E_{El}] \]

\[ \min_x J \quad (16) \]

\[ x \text{ is the optimization vector. For the travel drive it is } x_{\text{travel}} = (P_{El}, P_{PE}, i) \text{ and for the working drive } x_{\text{working}} = (P_{El}, P_{PE}, V_0, n_P). \]

In preference to combining the objective functions for costs and energy consumption Pareto optimization is used to find trade-offs between the two objectives. As the algorithm a state-of-the-art implementation paretosearch in MATLAB is used [29].

Stopping criteria are defined for the optimization, which discard unsuitable solutions. In the case of the working drive, for example, these are solutions that exceed the comparatively low maximum speeds of the pump or cannot provide the required flow rate due to low pump speed and maximum displacement. For both the travel drive and the working drive, exceeding the maximum torque of the \( EM \) is not permitted.

### 4. Results

In the following section, the results of the multi-objective optimization are presented and discussed. The results are divided into travel and working drive. Since no weighting of costs and energy consumption was made, the results are presented as Pareto fronts.

#### 4.1 Travel Drive

Figure 10 shows the optimization results of the single-wheel drive system. Each circle represents one set of design parameters whose energy consumption and costs are displayed against each other. The size of the circles reflects the installed electrical power, with larger circles representing a higher installed power. The color of the circles represents the required gear ratio.

![Figure 10. Result of the multi-objective optimization visualized as a Pareto front for the travel drive. The size of a circle is proportional to the installed electrical power of a solution. The color represents the gear ratio.](image-url)
For the travel drive, two separate Pareto fronts are obtained, which result from the discontinuous cost and efficiency functions of the planetary hub gearboxes. The first Pareto front at the top left of the diagram is characterized by high relative costs and low energy consumption. Within this Pareto front, only design parameters of two-stage planetary hub gearboxes are present. Variants with high installed electrical power lead to higher costs. Due to more efficient operating points in the efficiency map of the EM and PE, these variants have a lower energy consumption. The second Pareto front, which is characterized by low costs and a comparatively high energy consumption, shows only variants that utilize a three-stage planetary hub gearbox. The current gearbox cost model considers the material mass only, so that if more extensive influences are taken into account in the future, the results may vary. The results indicate that the costs for providing the required high torque by means of a third stage are lower than implementing a more powerful electric motor. However, the lower energy efficiency due to the additional gear stage is disadvantageous. A permission of additional system costs of 26% can reduce the energy consumption of the most efficient three-stage solution (1) to the most cost-effective two-stage solution (2) by 3%. A further 36% increase in cost between the most cost-effective two-stage solution (2) and the most efficient two-stage solution (3) improves energy consumption by only 1%.

4.2 Working Drive

Figure 11 shows the results of the multi-objective optimization for the working drive. In the left plot the size of the circles represents the installed electrical power while the color of the circles represents the maximum displacement of the hydraulic pump. In the right plot the circle color represents the required pump speed.

Figure 11. Result of the multi-objective optimization visualized as a Pareto front for the working drive. Left: The size of a circle is proportional to the installed electrical power of a solution. The color represents the required pump displacement. Right: The circle color represents the required pump speed.

Within the Pareto front, a gap is apparent, but unlike the travel drive, it is not due to a discontinuous component model. In fact, it shows a region where no solution satisfies the boundary conditions of the system. In the left plot of Figure 11 a clear correlation can be seen between the installed electrical power and the objectives cost and energy consumption. As the installed electrical power increases, the relative costs of the working drive increase and the energy consumption decreases due to more efficient operating points of the EM, PE and hydraulic pump. More efficient solutions utilize higher maximum pump displacements. In addition, they are operated at lower speeds (Figure 11, right). If 68% higher system costs of the most cost-effective solution (1) shown are permitted, the energy consumption can be decreased by 3.2% to the most energy efficient solution (2) shown.

5. Conclusion and Outlook

Due to stricter legislation, the demand for zero-emission drive systems in the field of mobile machinery will increase today and in the future. One possibility for operating construction machinery in zero emission zones is the use of battery-electric drive systems. Compared to the automotive sector,
however, the high power requirements of mobile machinery over long periods of time lead to high energy demands, which are reflected in heavy, large and expensive batteries. The efficiency of the drive concept consisting of topology, component sizing and control strategy thus moves to the center of the development.

In order to answer the question of how an efficiency- and cost-appropriate drive concept can be found, this paper presented a novel optimization-based method that allows the identification of an optimal component sizing for a given topology. The novelty of the presented method results from the combination of a computationally intensive and therefore for optimization methods unsuitable transient dynamic simulation, which makes it possible to consider the interaction of parallelly operated consumers in the component specification, and a performant quasi-static simulation, which is necessary to apply known optimization methods for the component sizing. The component models used in the simulations are characterized by scalability in the design parameter space and are arbitrarily interchangeable and extendable in the level of detail. Using a mobile excavator as an example system, the design parameter space was defined first. Corresponding to the selected design parameters, suitable models for the description of the objective criteria energy consumption and costs for the components EM, PE, hydraulic pump and planetary gearbox were presented. In a transient dynamic simulation, component loads were derived from measured load and motion cycles. These component loads serve as input for backward models, which implement the objective functions for a multi-objective optimization using scalable component models. Based on the results of the multi-objective optimization, which are presented in the form of Pareto fronts, a quantifiable trade-off of the objective criteria energy consumption and costs was derived. This trade-off helps the OEM to identify suitable component specifications in early stages of development by determining the additional costs they are prepared to accept in order to achieve an increase in efficiency. Using the example of the travel drive, it was possible to quantify the influence of the number of gear stages on the manufacturing costs and the energy consumption. By using a two-stage planetary gearbox paired with a more powerful EM, a 3% reduction in energy consumption can be achieved with additional costs of 26%. Using the example of the working drive, a reduction in energy consumption of 3.2% for additional costs of 68% could be demonstrated.

In the future, the presented approach and given framework should be extended and detailed. For this, the cost and efficiency models can be refined to provide an expansion of the current design parameter space. Using the example of the of the planetary gearbox a predimensioning can be added to the approach. This would expand the parameter space with quantities such as number of teeth, tooth module as well as shaft (diameter, length) and bearing (diameter, type) properties. In addition to higher accuracy in the mass estimation, analytical loss models can be used to calculate the gearbox component and operating point specific losses such as bearing and gear losses both load dependent and independent.

The sizing of the EM can be extended to the specification of the number of pole pairs as well as the length and diameter of the rotor and stator. The efficiency model can be expanded to more fidelity using the well-known Willans approach. The presented method thus can be adapted to the specific development task as needed and can be used in different phases of the product development.

6. Acknowledgements
The authors thank Liebherr-Hydraulikbagger GmbH for providing data, insight and expertise. The project is funded by the Federal Ministry for Economic Affairs and Energy (BMWi).
Abbreviations

| Abbreviation | Description               |
|--------------|---------------------------|
| EM           | Electric motor            |
| GB           | Gear box                  |
| OEM          | Original equipment maker  |
| PE           | Power electronics         |

Nomenclature

| Symbol | Description                                      |
|--------|-------------------------------------------------|
| α      | Relative displacement                           |
| η_EM+PE | Efficiency of the electric system              |
| η_G    | Efficiency of the gearbox                       |
| η.hm   | Hydraulic-mechanical efficiency                |
| η_pl   | Efficiency of a single planetary gear stage    |
| η_tot  | Total efficiency                               |
| η_vol  | Volumetric efficiency                          |
| C      | Costs of a subsystem                           |
| C_EM   | Costs of electric motor                        |
| C_G    | Costs of gear box                              |
| C_Hyd  | Costs of hydraulic pumps                       |
| C_i    | Costs of a component                           |
| C_PE   | Costs of power electronics                     |
| E_EI   | Energy consumption                             |
| f_G.m.f | Gear mass factor                               |
| i_pl   | Overall gear ratio of a single planetary stage  |
| i_St   | Gear ratio of a gear stage                     |
| J      | Multi-objective optimization function          |
| k_EM   | Costs factor of electric motor                 |
| K_HP   | Hagen-Poiseille coefficient                    |
| k_Hyd  | Costs factor of hydraulic pumps                |
| k_PE   | Costs factor of power electronics              |
| K_TP   | Proportionality factor for pressure-dependent  |
|        | frictional torque                             |
| M_G.m.f| Gear mass                                      |
| N      | Number of planets                              |
| n_mech | Rotational speed of the engine                 |
| n_P    | Rotational speed of the pump                   |
| n_W    | Rotational speed of the wheel                  |
| p      | Pressure                                       |
| Δp     | Pressure difference                            |
| P_EI   | Power demand                                   |
| P_EM   | Installed power of the electric motor          |
| p_L    | Load pressure                                  |
| p_P    | Pressure at the pump                           |
| p_S    | Stick cylinder pressure                        |
| P_PE   | Installed power of the power electronics       |
| Q_eff  | Effective volumetric flow rate                 |
| Q_leak | Leakage flow rate                              |
| Q_P    | Volumetric flow rate of the pump               |
| Q_th   | Theoretical volumetric flow rate               |
| S_S    | Stick cylinder stroke                          |
| T_0    | Constant frictional torque of the pump         |
| T_eff  | Effective torque of the pump                   |

\[
\begin{align*}
\alpha & \text{ Relative displacement} \\
\eta_{\text{EM+PE}} & \text{ Efficiency of the electric system} \\
\eta_G & \text{ Efficiency of the gearbox} \\
\eta_{\text{hm}} & \text{ Hydraulic-mechanical efficiency} \\
\eta_{\text{pl}} & \text{ Efficiency of a single planetary gear stage} \\
\eta_{\text{tot}} & \text{ Total efficiency} \\
\eta_{\text{vol}} & \text{ Volumetric efficiency} \\
C & \text{ Costs of a subsystem} \\
C_{\text{EM}} & \text{ Costs of electric motor} \\
C_G & \text{ Costs of gear box} \\
C_{\text{Hyd}} & \text{ Costs of hydraulic pumps} \\
C_i & \text{ Costs of a component} \\
C_{\text{PE}} & \text{ Costs of power electronics} \\
E_{\text{EI}} & \text{ Energy consumption} \\
f_{G,\text{mass}} & \text{ Gear mass factor} \\
i_{\text{pl}} & \text{ Overall gear ratio of a single planetary stage} \\
i_{\text{St}} & \text{ Gear ratio of a gear stage} \\
J & \text{ Multi-objective optimization function} \\
k_{\text{EM}} & \text{ Costs factor of electric motor} \\
k_{\text{HP}} & \text{ Hagen-Poiseille coefficient} \\
k_{\text{Hyd}} & \text{ Costs factor of hydraulic pumps} \\
k_{\text{PE}} & \text{ Costs factor of power electronics} \\
K_{\text{TP}} & \text{ Proportionality factor for pressure-dependent frictional torque} \\
M_{G,\text{mass}} & \text{ Gear mass} \\
N & \text{ Number of planets} \\
n_{\text{mech}} & \text{ Rotational speed of the engine} \\
n_P & \text{ Rotational speed of the pump} \\
n_W & \text{ Rotational speed of the wheel} \\
p & \text{ Pressure} \\
\Delta p & \text{ Pressure difference} \\
P_{\text{EI}} & \text{ Power demand} \\
P_{\text{EM}} & \text{ Installed power of the electric motor} \\
p_L & \text{ Load pressure} \\
p_P & \text{ Pressure at the pump} \\
p_S & \text{ Stick cylinder pressure} \\
P_{\text{PE}} & \text{ Installed power of the power electronics} \\
Q_{\text{eff}} & \text{ Effective volumetric flow rate} \\
Q_{\text{leak}} & \text{ Leakage flow rate} \\
Q_P & \text{ Volumetric flow rate of the pump} \\
Q_{\text{th}} & \text{ Theoretical volumetric flow rate} \\
S_S & \text{ Stick cylinder stroke} \\
T_0 & \text{ Constant frictional torque of the pump} \\
T_{\text{eff}} & \text{ Effective torque of the pump} \\
\end{align*}
\]
T\text{fric} & Frictional torque of the pump & Nm \\ T\text{mech} & Output torque of the electric engine & Nm \\ T_p & Pressure-dependent frictional torque of the pump & Nm \\ T_{th} & Theoretical torque of the pump & Nm \\ T_W & Torque at the wheel & Nm \\ v & Vehicle velocity & \text{km} \cdot \text{h}^{-1} \\ V_0 & Maximum displacement & \text{cm}^3 \\ x & Optimization vector & -- \\

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