SOLUTIONS FOR ENERGY-EFFICIENT AND EASY IMPLEMENTABLE ELECTRIFIED VARIABLE-SPEED PUMP DRIVES IN MOBILE APPLICATIONS

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ABSTRACT

Mobile machinery is currently undergoing a transformation towards an increasingly electrification of its powertrain systems, which is subject to ever-increasing demands in terms of performance, functionality, noise behaviour, integration and – above all – energy efficiency. Especially in the latter case, solutions on a system level appear more and more attractive as the advantages of electric drives can be fully revealed this way – most of all the high torque build-up dynamics and speed variability across a much wider speed range compared to combustion engines that also involves standstill as well as negative speeds. Following this idea, the contribution introduces a model-based and holistic approach to achieve a customer-specific system design that is targeted at a given individual load specification for the energy-efficient supply of the hydraulic implement systems with a demand-oriented volume flow at an optimal speed. Both, the application-optimized dimensioning of the drive components and the use of an operating strategy, lead in combination to significant energy savings during operation. These improvements are presented in this work, which is based on simulations and test bench measurements. Furthermore, this paper concludes with an investigation of the thermal behaviour of the electric drive using different speed operation strategies.

Keywords: Electrification, Mobile Machinery, Optimal Speed Strategy, Energy Efficiency, Model-Based Design

1. INTRODUCTION

1.1. Motivation and system solutions

Next to zero local emissions, the potential for much higher energy efficiency is a main motivation and driver for the increase in numbers of electric drives in mobile machines. Both aspects can be seen as an important basis for future CO_2 neutrality [3]. For battery-electric systems, saving energy plays a special role as well, because battery capacities are still rather limited and the amount of energy carried is currently still significantly smaller than in diesel-driven machines [2]. For the supply of hydraulic implement systems in these electrified machines, electric motor-driven pumps become more and more available. In addition to the already high efficiency of the electrical components, they allow a more consistent speed adjustment in a much larger range compared to diesel-driven systems due to their advantages in dynamics, like providing maximum torque from zero speed [7]. This results in further potentials for energy savings at system level.

The adjustable speed of an electrified pump system combined with commonly used variable displacement pumps (demand-based volume flow supply) provide a degree of freedom which allows a load and system state depending optimal utilization of all drivetrain components involved during the operation [1]. In order to achieve these goals and at the same time to take into account the very high variance of mobile applications themselves as well as their different application profiles, this contribution introduces a model-based and holistic approach to achieve a customer specific system design that is targeted at the customers' load specifications for the supply of their hydraulic implement systems with a demand-oriented volume flow at an optimal speed.

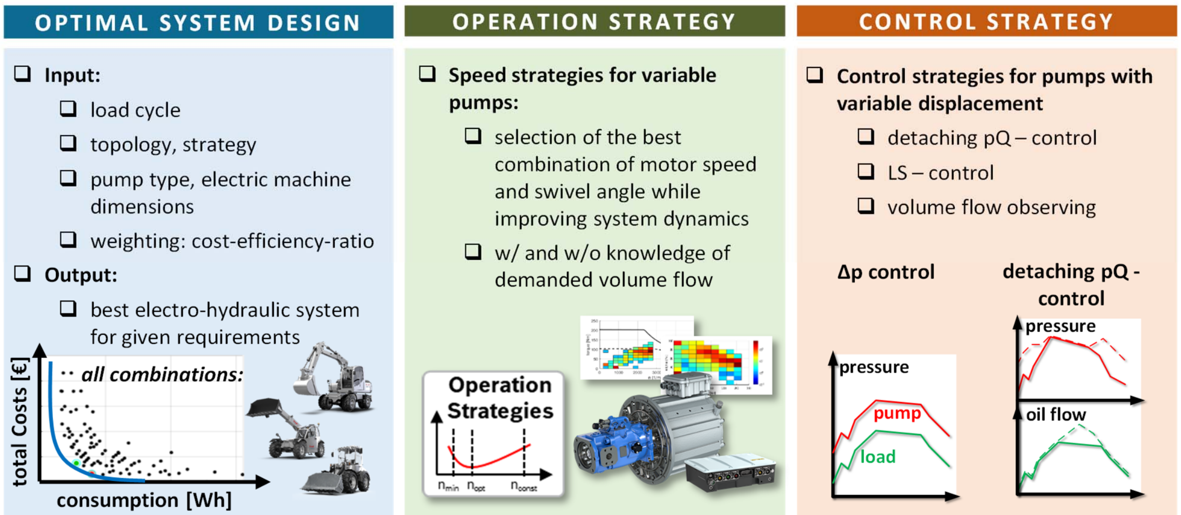

Figure 1: System solution approach for electrified pump drives

For the ideal dimensioning as well as the subsequent operation of an electrified pump drive, a model-based design tool is used, as shown in the left field in **Figure 1**. With the knowledge of the application, the considered load cycles, the relevant component data as well as the preferred operating strategy, a picture of all feasible combinations can be generated in a very efficient way.

Furthermore, real-time capable operating and control strategies adapted to the system can be implemented as software functions, as depicted in the middle field of **Figure 1**. These strategies allow the operation of variable-speed pump drivetrains to supply any common valve systems, such as for example load sensing (LS) and electronic positive control (ePC) systems, considering the efficiency maps and fundamental dynamic effects of all involved drive components.

In addition, a pump control system that is optimally selected for the respective application ensures the supply of a demanded volume flow and – if applicable – also a specified pump pressure limitation. Such control strategies can be, for example, a pressure difference control for LS valve systems or an alternating pressure or flow control for electronic flow matching (EFM) or ePC systems, like shown exemplarily in the right field of **Figure 1**. However, both control strategies can result in situations where the volume flow demand does not mandatorily correspond to the actual volume flow of the pump (pressure or power limitation in intervention and/or volume flow demands generally not known). For these cases, developed mechanisms ensure a stable behaviour and energetically efficient operation at the same time. All of these measures enable the best possible operation of the mobile machine in terms of performance, functionality and energy efficiency and thus also makes a notable contribution to energy, CO_2 and cost savings.

1.2. Related work and contribution of this paper

Electro-hydraulic pump systems which are using the inherent two degrees of freedom – variable speed and variable displacement – have been in focus especially for stationary industrial applications for a couple of years already. An example is hydraulic presses, whose energetic behaviour has been experimentally studied by Huang et.al. [1], varying speeds and loads. Furthermore, Zheng [9] proves the feasibility of variable speed drives also for pressure-controlled pumps in the same context.

Reidl et.al. [5] show that energy savings of up to 29% can be achieved for electro-hydrostatic axes.

Therefore, a loss optimal displacement trajectory is determined in this study, which presupposes the knowledge of the load cycle. Complex models of the pump, in which neural networks are used, form the basis for the dynamic optimization algorithm. The specific application with all its drive and transmission components is known and all their characteristics are considered within the optimization model. Moreover, neural networks, here in backpropagation form, can be found in Rui et al. [6] to set the optimal speed and swivel angle based on the measured pressure and flow rate of the system.
Furthermore, many solutions provide an offline determination of the trajectories for optimal swivel angle and optimal speed – see for example [4] and [5]. In [4], two different algorithms are investigated by Reichert et.al. One algorithm looks purely at the quasi-stationary states during operation and the second algorithm is a model-based approach that also includes dynamic effects. With Willkomm [8] the consideration of very dynamic processes is in the foreground, whereby an approach is pursued with the help of model predictive methods, to achieve – above all – very high process dynamics with simultaneous energy efficiency.
In contrast to the related work mentioned above, this contribution will take up the aspect of the speed variability of a pump drive and examine especially its energetic advantages in the context of mobile machines. The focus is on a model-based approach, which is used both in the already mentioned speed strategies but also in a dimensioning tool. With this tool, dimensioning becomes automated and much easier, as a variety of different component size combinations can be compared with each other regarding several criteria that go beyond energy efficiency. In addition, both, an optimal design and the use of an optimal speed strategy have beneficial effects on the thermal behaviour mainly of the electrical components. In this contribution, the results of an exemplary design for the application mini excavator are presented and validated by measurements on a test bench. In the referenced works, the corresponding load cycle is often very specific and well known. For this cases, offline calculated trajectories are often used. This is generally not possible with mobile machines, as the loads and speeds typically depend on the machine type, work task, operator, and potentially many other influences. Therefore, online strategies or approximations can be used.

2. MODEL BASED APPROACH FOR OPTIMAL ELECTRIFIED PUMP DRIVES

2.1. Implementation of a speed-variable electrified pump drive system

Figure 2 shows the schematic representation of the relevant assembly and top-level functional architecture of a variable-speed variable-displacement pump drive for electrified mobile applications, which can be seen here in the implementation for a typical digitized control structure of a todays hydraulic implement system. The essential relationships of the assembly and the function groups with each other but also in the context of the implement system are indicated in the form of physical interfaces and signal flows. Here, an implement control unit evaluates the operator demands, for example joystick signals, and determines the control signals for the individual valve units of the main control valve (MCV). In the following, these valve units measure the desired flow rates for the hydraulic loads and the available pump flow rate is distributed according to the operator demands and the valve system characteristic. The exemplary valve system shown here is a LS system, but other valve systems such as ePC or EFM systems are also possible, in which the nominal volume flow of the pump can be determined from the operator demands.
The task of the electrified pump drive system is now to provide the required pump volume flow according to the operator demands and with regard to further criteria in an optimal way. For this purpose, the volume flow request derived from the operator must be made available as a signal. Based on the selection of one of the operating strategies, which are implemented as software on a control unit, this signal is used for the calculation of the desired speed of the electric drive unit driving the pump. This generated speed setpoint signal is simultaneously transferred - alternatively also the signal

of the actual speed - to the pump controller, which, depending on the selected control type (e.g., volume flow control, LS control, etc.) adjusts the corresponding swivel angle so that the pump can provide the requested volume flow.

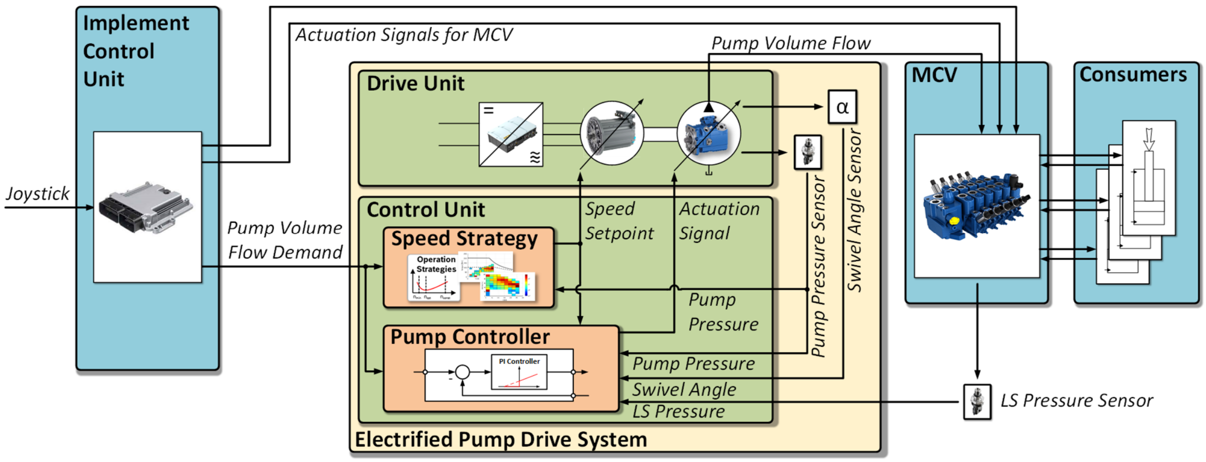

Figure 2: Schematic representation of the electrified pump drive system

By considering the different characteristics related to the energetic behaviour of the electric drive (inverter and electric machine) and the hydraulic pump, the different strategies for speed adjustment can be derived. For this purpose the respective input and output powers of the components involved are to be considered. The following relationship applies to the hydraulic pump:

$$P_{hyd} = Q_{eff} \cdot \Delta p = \eta_V \cdot \frac{\omega}{2\pi} \cdot V_g \cdot \Delta p = \eta_V \cdot \omega \cdot T_{ideal} = \eta_V \cdot \eta_{mech} \cdot \omega \cdot T_{eff} \tag{1}$$

$$P_{hyd} = \eta_V(n, \alpha, \Delta p) \cdot \eta_{mech}(n, \alpha, \Delta p) \cdot P_{mech} = \eta_{Pmp}(n, \alpha, \Delta p) \cdot P_{mech} \tag{2}$$

According to Equation (2), the hydraulic power at the pump outlet results from the mechanical power at the pump shaft multiplied by the volumetric efficiency η_V and the mechanical efficiency η_{mech}. These efficiency values are very often available as measured lookup tables for the leakage (η_V) as well as for the friction (η_{mech}). The input variables of both tables are the speed n and the swivel angle α of the pump as well as the pressure difference Δp between suction and pressure ports. The mechanical power at the input of the pump can be assumed to be equal to the mechanical output power of the electric machine, so no equally lossy transmission is arranged between the electric machine and the pump. The output power P_{mech} of the electric machine can also be determined by an efficiency ratio as shown below:

$$P_{mech} = P_{EmOut} = \eta_{Em}(i_{Ph}, u_{DC}, n) \cdot P_{EmIn} \tag{3}$$

The efficiency map covers both, the occurring mechanical losses as well as the electrical iron and copper losses of the permanent magnet synchronous machine (PMSM) underlying these investigations. To drive the synchronous machine, an electric rotating field is required, which is generated in its three windings. The necessary three-phase current is provided by the electrical inverter, wherein the variable frequency of the for this generated three-phase voltage is responsible for the variability of the speed of the motor and thus also of the pump. Like the pump and electrical machine, a relationship between the input power and the output power of the inverter can also be find out via the determination of a characteristic efficiency map:

$$P_{InvOut} = P_{EmIn} = \eta_{Inv}(i_{Ph}, u_{DC}, n) \cdot P_{ElecIn} \tag{4}$$

The loss behaviour of the semiconductor power elements used in the inverter is mainly described by the phase current i_{Pha} and by the active power factor $cos\ \varphi$, which depends on the utilized electric machine and is mainly characterized by its inductive behaviour. Since $cos\ \varphi$ as well as the efficiency of the electric machine can also be determined by the physical quantities phase current i_{Phi}, DC link voltage u_{DC} and speed n, the equations (3) and (4) can be combined and the corresponding powers can be described by the overall efficiency characteristic η_{eDrv} of the electric drive. For the overall drive system including the hydraulic pump, the following equation results:

$$P_{hyd} = \eta_{eDrv}(i_{Ph}, u_{DC}, n) \cdot \eta_{Pmp}(n, \alpha, \Delta p) \cdot P_{ElecIn} \tag{5}$$

These relationships are depicted in the following **Figure 3**. Shown are the corresponding losses of the pump, the electric machine and the electric inverter. The latter can be combined into an electric drive and be handled in the further course of the explanations in this contribution.

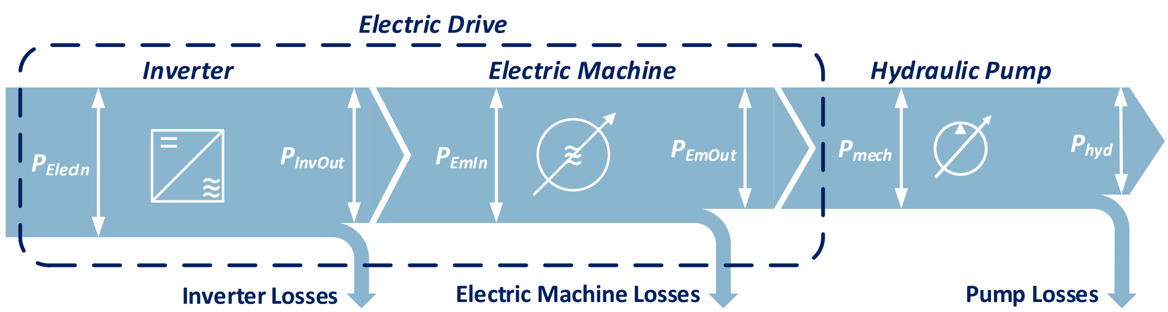

Figure 3: Schematic representation of the power flow within the electrified pump system

2.2. Speed-variable strategies to increase energy efficiency

The following **Figure 4** shows exemplarily the characteristic efficiency behaviour of both the electric drive and the hydraulic pump:

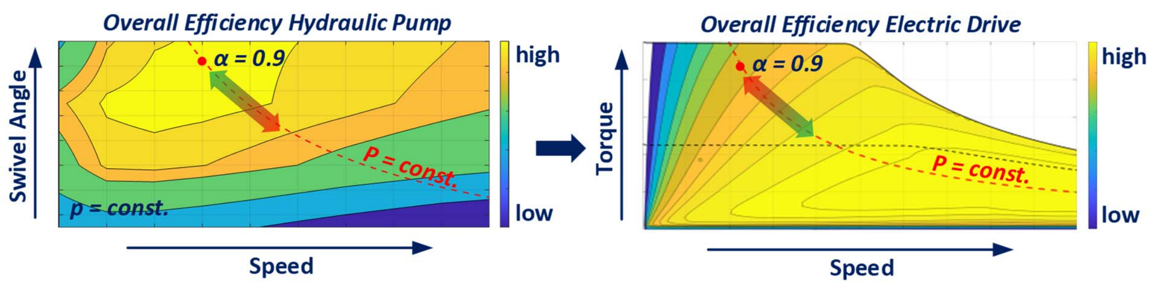

Figure 4: Exemplary efficiency maps of the hydraulic pump (left) and the electric drive (right)

The illustration on the right shows the typical torque behaviour of a permanent magnet synchronous machine in motor operation. The solid line shows the course of the peak torque, and the dashed line represents the course of the permanently available torque. Both torques are almost constant up to the nominal speed of the electric machine, in order to then fall off when the field weakening operation is reached at constantly increasing speeds. With increasing speeds, however, the energy efficiency of the electric drive system increases. The left illustration of **Figure 4** shows the characteristic efficiency behaviour of an axial piston pump with variable displacement volume that was used in this study. Assuming a constant pressure, one can see that high efficiency values are achieved, especially for larger swivel angles at lower speeds (which result in higher torques for the given hydraulic power demand). For the electric drive it is the opposite in that case. From this, two elementary strategies for setting a speed can be derived, which should enable a more energy-efficient overall behaviour of the electric pump drive compared to constant speed. If only the pump behaviour is considered, one would always choose the largest possible swivel angle in order to be able to operate at least the pump in an energy-optimal range. Such an implementation can be achieved by applying the following equation for determining the pump volume flow with α close to 100%:

$$n_{min} = \frac{Q_{eff}}{\eta_V \cdot \alpha \cdot V_{gmax}} = \frac{Q_{des}}{\eta_V \cdot \alpha \cdot V_{gmax}} = \frac{Q_{des}}{\eta_V(n, \alpha, \Delta p) \cdot \alpha \cdot V_{gmax}} \tag{6}$$

Starting from the equation for describing the effective volume flow Q_{eff} of the pump, which simultaneously corresponds to the desired volume flow Q_{des}, the necessary speed is determined according to equation (6). This is the minimum necessary speed at which the volume flow can be realized and is therefore referred to as the minimum speed n_{min} in the further course. For the volumetric efficiency η_V, either a simple constant or the efficiency map of the pump can be used, which must be determined in advance and implemented on the control unit.

The minimum speed strategy impresses with its simple implementation and corresponds to the preference to adapt the speed to the actual volume flow demand. However, the losses caused by the electric drive are not considered. If you take a closer look at the efficiency behaviour of the pump and the electric drive, you can see that increasing the efficiency of both drive technologies requires opposite speeds. This suggests an optimum, which can be described by the following optimization problem P, in which the electrical input power P_{ElecIn} based on equation (5) is minimized online assuming that the hydraulic input power P_{hyd} and further system states are known:

$$n_{opt} \rightarrow P: \quad \min_{n} \left(\frac{p_{pmp} \cdot Q_{des}}{\eta_{Pmp}(n, \alpha, \Delta p) \cdot \eta_{eDrv}(i_{Ph}, u_{DC}, n)} \right) \quad n \in R, n_{min} \leq n \leq n_{max} \tag{7}$$

3. APPLICATION AND VALIDATION OF THE MODEL-BASED DESIGN TOOL

3.1. Dimensioning and selection of an optimal solution based on a concrete example

In order to be able to select an optimal combination from a variety of inverter, motor and pump sizes that are suitable for a specific application, it is necessary to know the typical load profile for the application, like you can see in the following **Figure 5**:

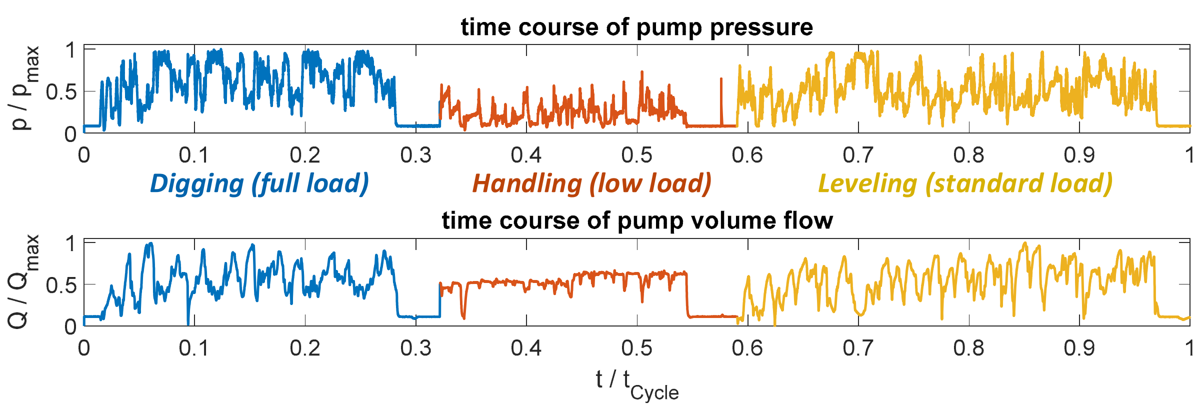

Figure 5: Composition of a load profile using the example of a mini excavator

This load profile represents the temporal pump pressure and the temporal pump volume flow of a combination of typical work tasks for an exemplarily selected mini excavator application. The values are normalized to the occurring maximum values of the cycle course. In the specific example, this would be the digging cycle as a representative of a full load task, the handling cycle, which stands for a low load, as well as the levelling cycle as standard or rather partial load task. The use of a mix of typical work tasks is particularly advisable for mobile applications, which have a large degree of generalization regarding a range of tasks and can usually be used in their everyday work for many different activities. While the full load cycle defines reasonable performance requirements for the components, the combination of different cycles is necessary for a more realistic energy consumption evaluation. This is especially true for a design that focuses on maximizing energy efficiency.

A mathematical model of the electrified pump drive system shown schematically in **Figure 2** was developed in which the essential stationary relationships, mainly the energetic behaviour in the form of efficiency maps, but also the important dynamic effects, such as swivel angle and speed build-up dynamics, have been implemented. In addition, the speed strategies presented in Section 2.2 are implemented within this model. With the help of the provided load cycles, which act as inputs for the model, the behaviour of the electrified pump drive system can be simulated. The developed framework prepares the data of the corresponding components and generates a graphical representation of all simulated possible and permissible drive combinations. The result of such a simulation run for the exemplary selected application mini excavator and the compilation of a representative load cycle shown in **Figure 5** can be seen in the following **Figure 6**:

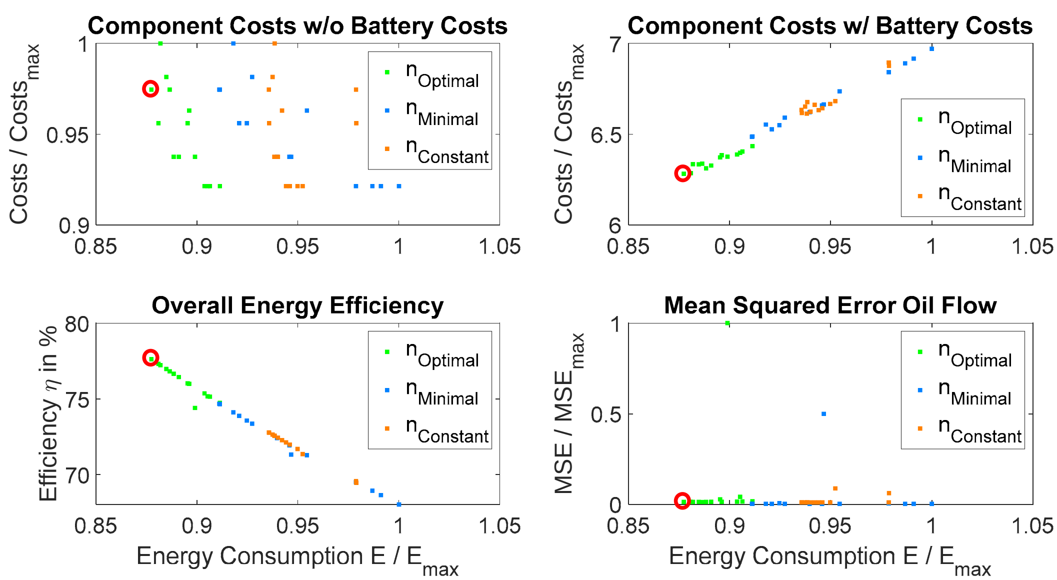

Figure 6: Presentation of the results of a simulation run of different combinations of the drive components

The results of the individual simulation runs are shown as corresponding points. The representation is again normalized here. The x-axes of all diagrams show the simulated energy consumption at the electrical input of the inverter. Further possible evaluation criteria for the design of a drive system, such as component costs with or without consideration of the battery (assumption: 8 h working time without charging), are selected as examples. The normalization of costs is carried out to the respective component cost maximum. It turns out that the selection of more expensive components of the actual powertrain may be worthwhile if a significant proportion of the total costs caused by a larger battery can possibly be saved (see drive combination marked in red in **Figure 6**).

Further statements can be made about the simulated operating strategies. The most obvious finding from the analysis of the results is that the optimal speed strategy achieves in some cases significantly higher energy savings compared with a minimum speed strategy. The contrast is even higher when looking at a constant speed strategy. Constant speed refers here to the speed at which the maximum volume flow of the cycle can be achieved. The representation of the mean squared error shows a measure of the deviation between the simulated volume flow and the predetermined volume flow Q_{des}. In this case, there are some outliers caused by effective torques that are too high. Depending on the design of the system, the pump controller can limit the torques that are too high by swivelling back the pump, so that the drive does not stall. However, this has no effect on a possible compensation of the missing volume flow by increasing the speed. This speed adjustment can be considered, for example, when implementing the optimal speed strategy by adding a corresponding constraint during optimization.

3.2. Validation of the simulation results by measurements on a test bench

In order to validate the model-based methods, models and operating strategies under realistic operating conditions, the electrified pump drive system was set up on a test bench. The test bench setup can be taken from the following **Figure 7**:

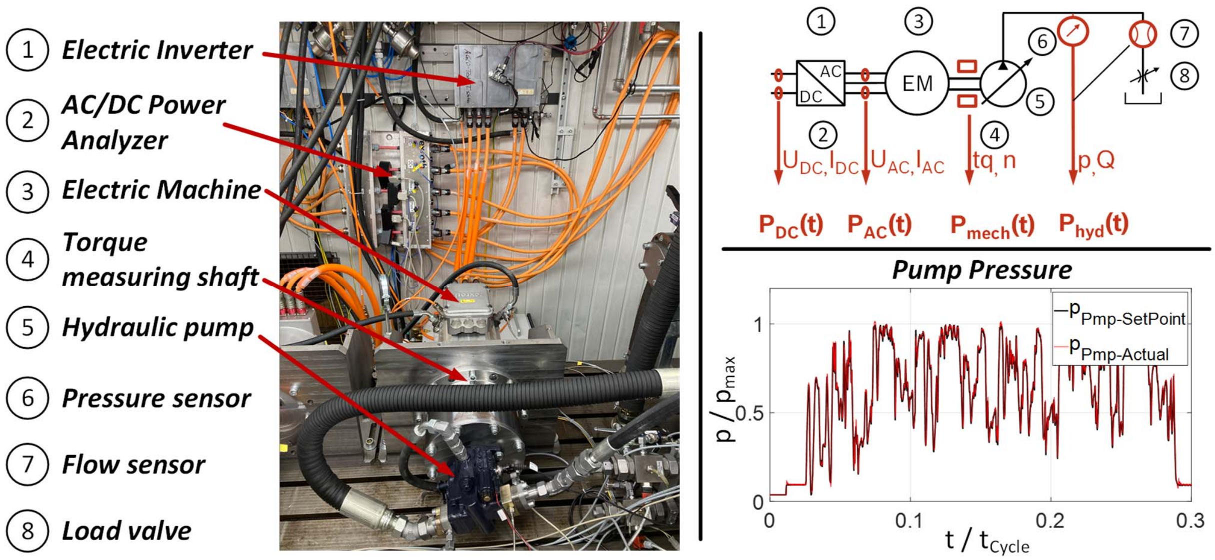

Figure 7: Test bench setup and test results

Bosch Rexroth components were used for the drive train. Specifically, this comprised an eLION EDS1 inverter, an eLION EMS1 permanent magnet synchronous machine and an A10VO axial piston pump with variable displacement and eOC Control. The software with the different operating strategies and the pump control were implemented on a BODAS RC5-6/40 control unit. The control unit received the required volume flow Q_{des}, which was set by the pump controller. The pump pressure was controlled by a pilot-operated proportional valve (8 in **Figure 7**). The trajectory of the load pressure control can be taken from the time course of the pressure (also in **Figure 7**). The following **Table 1** shows in a comparison the efficiency results obtained by simulations and measurements compared to a constant speed operation.

Table 1: Comparison of the efficiency gain relative to constant speed using the mini excavator cycles

Load cycle	**combined**		**digging**		**handling**		**levelling**	
Speed strategy	n_{Min}	n_{Opt}	n_{Min}	n_{Opt}	n_{Min}	n_{Opt}	n_{Min}	n_{Opt}
Simulation	1.6 %	3.3 %	-0.2 %	2.3 %	8.3 %	7.8 %	1.2 %	2.9 %
Testbench	1.7 %	3.1 %	-0.1 %	2.0 %	9.3 %	8.8 %	1.1 %	2.5 %

The results of the efficiency gain in percentage points show a very good agreement of the efficiency behaviour between the simulated and the measured data. As with the derivations to the operating strategies (section 5), the energy efficiency results from the division of the measured hydraulic power by the measured electrical AC power at the input of the inverter. In order to obtain a more differentiated statement, both the combined cycle and the three individual cycles from **Figure 5** were simulated and measured. Variable-speed drives show their advantages especially in cycles with low loads, but also in cycles with high loads, significant efficiency advantages could be achieved.

3.3. Impact of speed strategies on thermal behaviour

In the following consideration, the influence of the selection of the speed strategy on the thermal behaviour of the drive system is examined in more detail. For this purpose, **Figure 8** shows characteristic maps with the operation points of the load cycles of the electrical machine and the inverter. The bottom plots show the thermal behaviour of the inverter over time. To achieve a thermally balanced state, the single digging cycle was simulated several times.

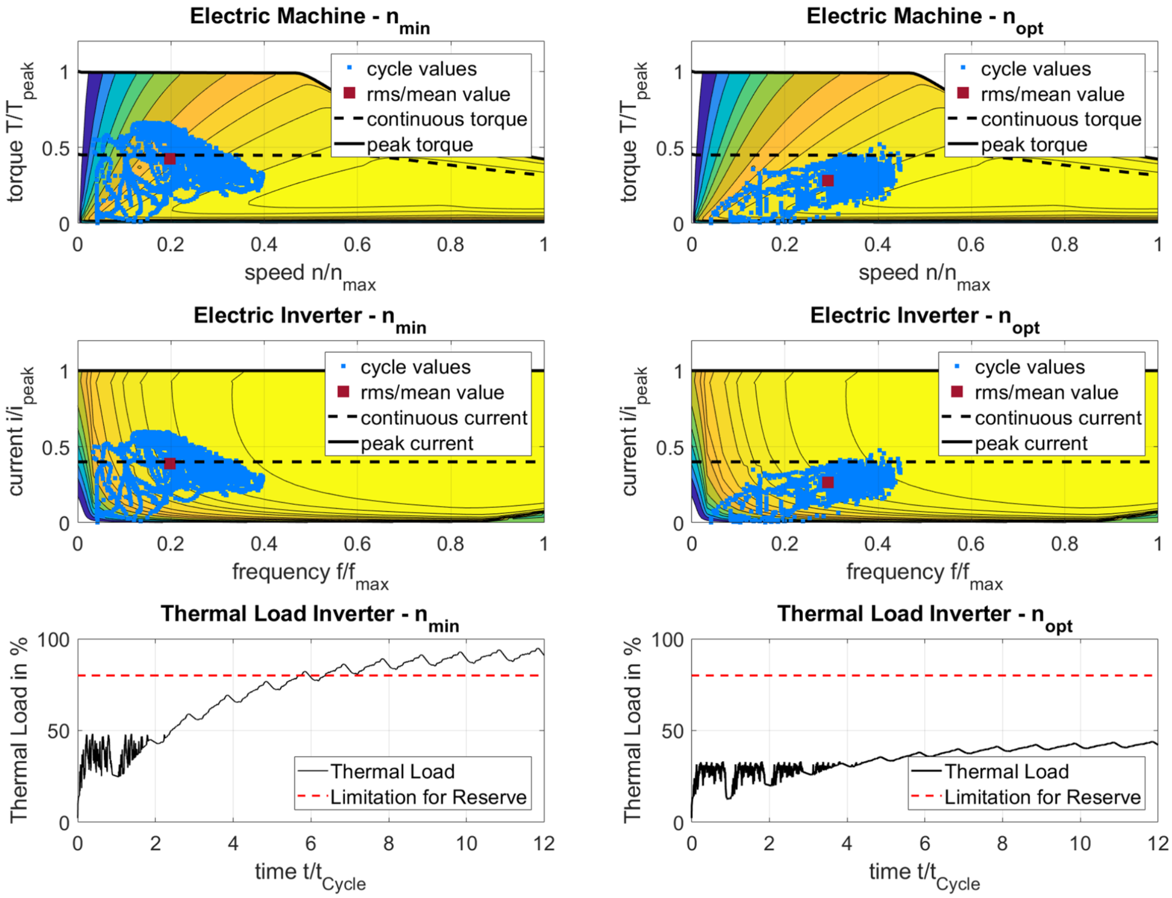

Figure 8: Effects of the speed strategies on the thermal behaviour of the drive components

The upper two figures show the stationary torque over speed behaviour of the electric machine in normalized representation. The behaviour at minimum speed is shown in the left illustration and at optimum speed in the right illustration. The point clouds reflect the resulted speed and torque operation points during the simulation, which are sampled with 10 ms. Since the thermal design should reasonably be carried out using the maximum expected loads, the consideration is based only on the digging cycle shown in **Figure 5**.

Particularly noticeable is the distribution of a significant number of working points above the continuous torque curve when using the minimal speed strategy compared to the optimal speed strategy. This circumstance results from the high swivel angles and results in an increased residence area with smaller efficiencies of the electric drive, which both leads to an increased RMS value of the torques. This is only just below the continuous torque curve, which could speak for an increased thermal utilization of the electrical machine. The same behaviour you can find in the representation of the current-frequency behaviour of the inverter, in which the torques were converted into effective values of the phase current and the rotational speed into the electrical rotational frequency of the

phase current. Again, by using n_{Min} strategy the RMS of the effective phase current values is significantly closer to the continuous curve of the phase current than when using n_{opt} strategy.
Thus, in the simulation of the thermal behaviour of the inverter, there are now very big differences. Over time the thermal load reaches almost 100% when using n_{min} strategy. At this value, the current would be limited in the real operation of the inverter, which can lead to either a stall or at least to a reduction in the volume flow conveyed. For this purpose, however, the pump controller must be designed to reduce the swivel angle in such a way that the pump torque is not higher than the reduced drive torque during a thermal overload situation of the electric drive. In both cases, however, a reduced performance would appear. Such a situation does not occur with speed strategies which utilize the degree of freedom like the optimal speed strategy where the best swivel angle and speed combination is determined based on current circumstances. The thermal behaviour when using n_{opt} strategy is not critical and the thermal utilization is well below a reserve rate of 80%, which should be aimed for design purposes at most. An advantage of an uncritical thermal behaviour when using an optimum speed strategy can be seen in the fact that a greater overload capacity of the electric drive system is maintained and, if necessary, the drive system can be optimized to use smaller dimensioned components.

4. CONCLUSIONS AND OUTLOOK

In this article, a procedure for the design and energy-efficient operation of an electrified pump drive for supplying the hydraulic work equipment of a mobile working machine with a demand-oriented volume flow is presented. Variable speed strategies play an essential role here and aim either to maximize the swivel angle alone or to minimize the loss of the entire electrified pump drivetrain. With the help of the operating strategies and a model-based approach, many different combinations of component sizes can be simulated and evaluated in their behaviour in an efficient way. This represents a very powerful tool for the users to enable a principally efficient operation of their mobile machine in combination with the use of the speed control strategies. During operation, depending on the application and the work task performed, an average of 3 to a maximum of 9 percentage points of electrical energy can be saved compared to operation at constant speed. A further advantage is obtained, when using the optimum speed strategy – also regarding the thermal behaviour of the electrical components, which is uncritical and assures the full performance of the mobile application during its operation. In our current development, we continue to work on speed variable solutions that offer an user-friendly implementation of variable-speed pump drives including pump control, inverter device and communication as well as power limitation control.

NOMENCLATURE

α	swivel angle of the variable pump	%
$\cos\varphi$	active power factor	
EFM	Electronic Flow Matching System	
ePC	Electronic Positive Control System	
f	frequency, frequency of the three-phase rotation field of the electric machine	Hz
i_{Ph}	phase current	A
LS	Load Sensing System	
MCV	main control valve	
MSE	mean squared error (between demanded and simulated oil flow)	$(m^3/s)^2$
n	speed of pump / electric machine	rpm
n_{const}	constant speed strategy (referenced speed for maximum desired oil flow)	
n_{min}	minimal speed strategy	

n_{opt}	optimal speed strategy	
η_{eDrv}	overall efficiency electric drive	%
η_{Em}	efficiency electric machine	%
η_{Inv}	efficiency inverter	%
η_{mech}	mechanical efficiency of the pump	%
η_{Pmp}	overall efficiency of the pump	%
η_V	volumetric efficiency of the pump	%
p, Δp	pressure, pressure difference	Pa, bar
P_{EmIn}	input power of electric machine	W
P_{EmOut}	output power of electric machine	W
P_{ElecIn}	electrical input power at inlet of inverter	W
P_{hyd}	hydraulic power at output of electric pump drive	W
P_{InvOut}	output power of inverter	W
P_{mech}	mechanical Power at pump shaft	W
RMS	root mean square	
t_{Cycle}	cycle time	s
T_{eff}	effective torque at inlet shaft of pump	Nm
T_{ideal}	calculated ideal torque of pump based on displacement and pressure	Nm
u_{DC}	intermediate circuit voltage	V
Q_{Des}	desired / demanded oil flow	m^3/s
Q_{eff}	effective oil flow at output of pump	m^3/s
V_g	displacement of pump	m^3
ω	circular frequency	

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