VALIDATION OF A HYDRAULIC PULSE CONTROLLER ON AN OFF-HIGHWAY MACHINE

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ABSTRACT

There are different ways of controlling hydraulic variable displacement pumps, mostly the choice of control depends on the application and the respective requirements. With a focus on open circuit off-highway applications, the classic control structures such as pressure cut-off, load sensing, positive or negative flow control or power control are standard. The design of these controllers is usually the responsibility of the pump manufacturer and is often solved purely hydraulically. With the current trends, such as connectivity and increased efficiency of systems, a development into electrically actuated pumps becoming apparent. The preferred technology for this is usually a modified or improved version of the mechanical controller. Using proportional technology for oil flow control of the control system of a pump is one possibility but does not use the entire potential of an intelligent and efficient pump. With the approach of system optimization, a new controller based on two independent digital valves was already presented at the IFK in 2022. This hydraulic pulse controller (HPC) has since been developed further and a system comparison with different pumps has been tested on a modified test rig. In addition, its suitability for controlling an application was tested as part of a proof of concept on a real working machine. Furthermore, the development is now focused on the definition of this as a new smart subsystem and optimal system integration into existing architectures.

Keywords: digital hydraulic, Hydraulic Pulse Control (HPC), digital valve, variable displacement pump, efficiency, pump control, working maschine, flexibility, smart subsystem

1. INTRODUCTION

According to the current state of the art, the adjustment of variable displacement pumps is mostly managed by proportional valves. The in- or outflow of the control chamber is influenced by spool valves. In most cases, the system is controlled purely hydro-mechanically (passive), based on the area ratio, mechanical components like springs and the respective pressure level. Electro-hydraulic control valves can also be found in such applications; these are based on identical spool technology but are operated and positioned by a solenoid and controlled according to electrical signals.

In difference to the state of the art, the Hydraulic Pulse Controller (HPC) uses two independent fast-switching digital valves, which are controlled in a digital hydraulic manner (**Figure 1**). The valves have a hydraulic switching time of 2ms and are triggered with a PWM signal, which defines the opening duration and the level of opening. Rapid opening and closing results in a corresponding cross-section, which can be changed quasi arbitrarily by adjusting the PWM dutycycle. This makes it possible to achieve control accuracy analogous to proportional valves.

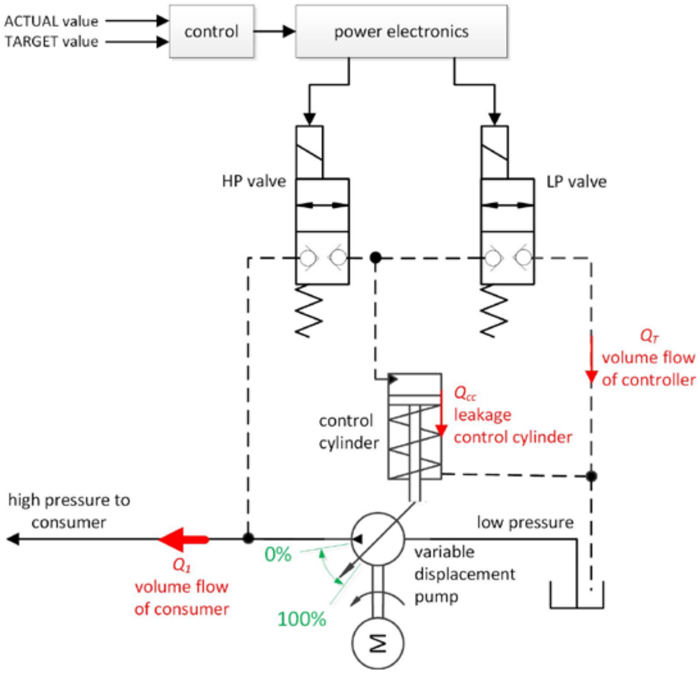

Figure 1: left: HPC (Hydraulic Pressure Control), digital pressure control schematic; right: HPC realized with 2 PULSE valves

This new way of acting on the control system results in significant advantages in terms of performance and energy savings. For example, the leakage or power loss of a previous spool valve is almost completely eliminated. Also, the two independent digital valves no longer represent a compromise between inflow and outflow of the oil flow, as it is on a spool with multiple control edges. This system is also less susceptible to contamination and offers enormous potential for energy savings by changing the operating mode and reducing the control pressure in the overall system. This controller was published for the first time on IFK2022 [1] and its functionality and advantages were proven through testing on various test benches. What is new in the meantime, however, is the extended control architecture, further conclusive tests with different pumps on a modified test rig in the laboratory and a test carried out on a real working machine in cooperation with the Innovative Hydraulics and Automation (IHA) of Tampere University, Linz Center of Mechatronics GmbH (LCM) and Andreas Lupold Hydrotechnik GmbH (ALH). Before the HPC was installed, comparative measurements were performed on an existing system with electro-hydraulic control (Rexroth). With a definition of certain test cycles, it was possible to evaluate the performance achieved and the general suitability of the system on a forwarder.

2. LABORATORY TESTS

For the established test rig [1] a hydraulic drive was used for powering the test pump (UUT, Unit Under Test) equipped with different controller types including the digital HPC controller. At the beginning the standard A10 DFR-Controller made by Lupold were applied and tested as some sort of reference and subsequent the idea of the digital control alternative was realized with on the market available 2/2-way seat type valves. They showed up to be too small for the aimed performance of the pump control and are therefore not suitable for such applications. Bigger valves were either not available or too slow in reaction time and not reaching the desired dynamic performance, so the digital valve PULSE was developed and optimized [2] upon which the current HPC (Hydraulic Pulse

Control) is based.

Figure 2 shows a picture of the PULSE valve and its duty cycle flow-rate behaviour at different pressure levels. It can be seen, that nominal down to very low flow rates can be achieved by choosing the appropriate duty cycle (50Hz excitation frequency), even at high pressure drops. Due to the fact, that the valve is not pressure equalized, the moment of opening respective minimal reasonable duty cycle to achieve useful flow rates varies depending on the differential pressure. Higher pressure drops lead to lower necessary minimum valve excitation times.

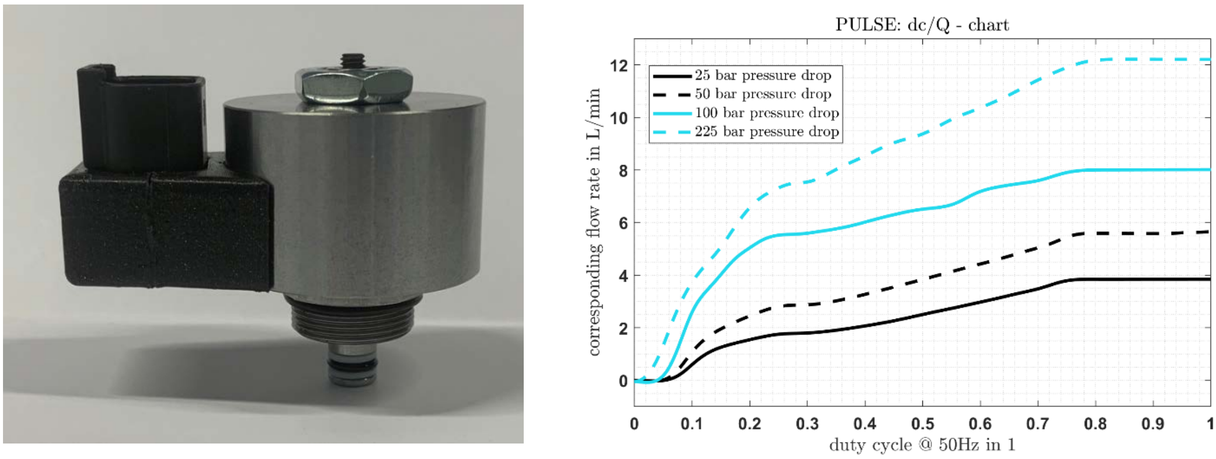

Figure 2: left: picture of the PULSE valve, right: characteristic diagram of flow rate at different duty cycles and pressure drops (based on 50Hz PWM-frequency)

2.1. The new measurement setup:

During completion of the former measuring objectives some issues and new requests showed up: The different controller types naturally lead to different results in performance and efficiency factor. Clearly determining the efficiency factor of the UUT in every working point would have been desirable. On the primarily side, in fact, only the pressure for the hydraulic drive was measured and the oil flow just estimated by the hydraulic source’s swivel angle sensor. Due to the unknown characteristic of the hydraulic drive itself, it was impossible to clearly determine the mechanical power fed to the UUT. Comparing different controllers was of course possible but not the overall change of the efficiency factor of the UUT. Another issue was the motor-pump-load constellation, combining several hydraulic controllers, UUT and hydraulic load including the pressure control of primary oil source and hoses, which led to oscillations at some operating points and so fragmentary efficiency factor charts.

To get ‘cleaner’ results and additional information the test rig was advanced (**Figure 1Figure 3**). To reduce the hydraulic complexity the hydraulic drive was replaced with an 22kW electric asynchronous motor with frequency converter and to measure the mechanical power, driving the UUT, and a torque transducer with integrated measurement for the rotation speed was applied. To provide tank pressure oil for the UUT it was necessary to actively pump oil from the oil power unit below the laboratory upstairs. To keep the pressure at a level of about 3 bar a check valve was integrated in the back flow hose. This minimum pressure is also checked to prevent the UUT to be rotated without proper tank oil supply.

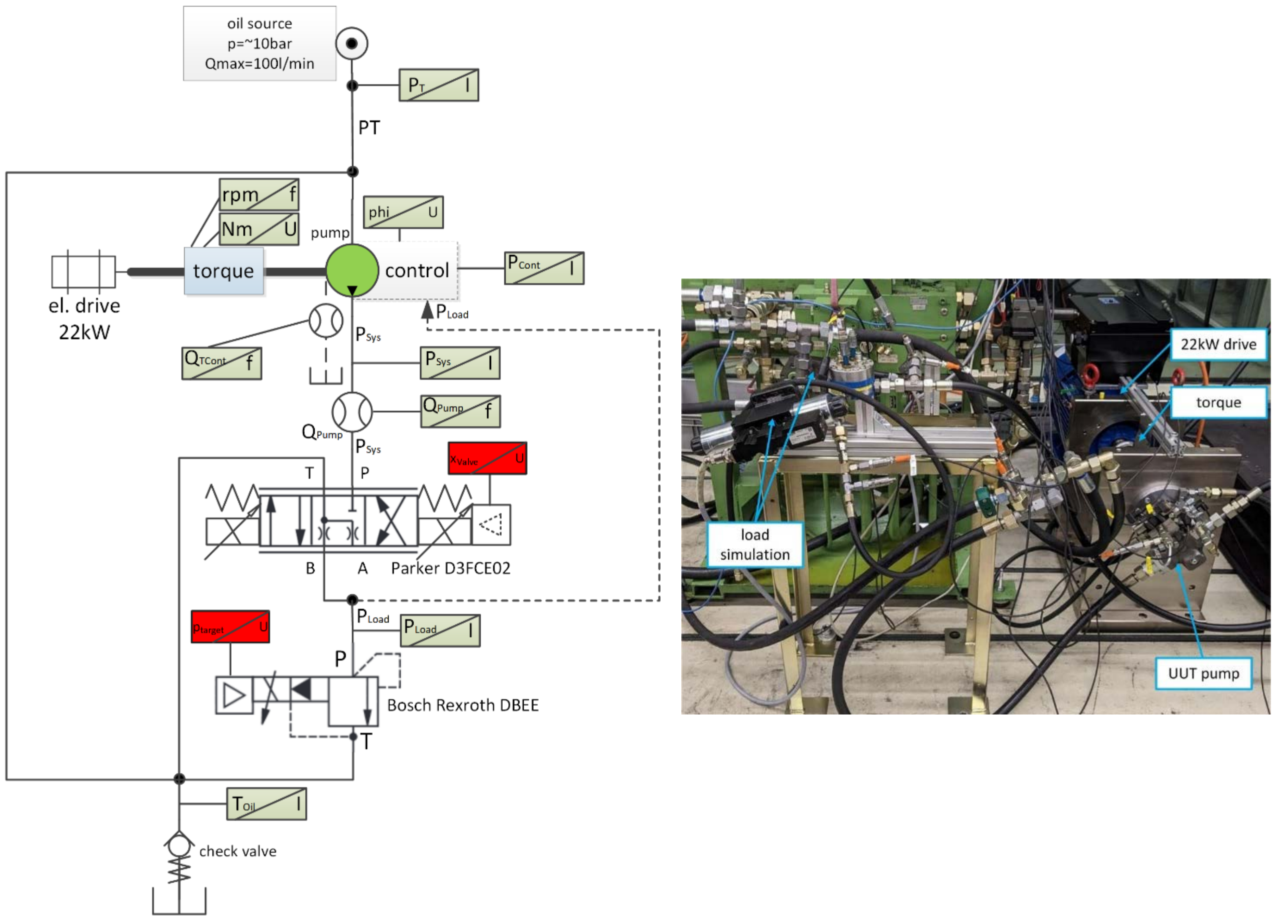

Figure 3: left: test rig setup schematic; right: picture of test rig at laboratory

Together with the volume flow sensor measuring Q_T (**Figure 1**) of the controller it is not only possible to distinguish between the mechanical power needed by the UUT to generate the required hydraulic power but also to identify the losses of the controller itself. The leakage Q_{CC} of the control cylinder is not considered as a loss of the controller but only the leakage of the low pressure valve. With a leakage free control cylinder at a stationary operation point the oil coming (leakage) from the controller's high pressure side would raise the pressure. To keep the operation point the oil would have to be dumped whether by the low pressure valve's leakage undesignedly or actively by piloting the low pressure valve. Either way, the amount of oil dumped by the low side valve comes from the high side valve at the pressure of p_S and so the power loss of the controller is estimated by:

$$P_{cont} = p_S \cdot Q_T \tag{1}$$

When it comes to comparing different controllers mounted on the same UUT the leakage of the control cylinder would be the same at the same operation points due to the leakage of the control cylinder. This loss is included in the overall efficiency factor as well as the hydraulic power needed for the controller.

$$P_{mech} = M \cdot \omega \tag{2}$$

$$P_{hydr} = p_S \cdot Q_1 \tag{3}$$

$$\eta_{tot} = P_{hydr}/P_{mech} \tag{4}$$

With the present setup the overall efficiency of the UUT and the characteristic of the controller can be determined (e.g. **Figure 4**, **Figure 5**). Especially the profiles in the right plots show the high benefit in the overall efficiency using the HPC controller in comparison to the established hydro-mechanical controller, reaching an increase of up to 10% at lower loads. Some investigations with proportional valves can be found in [5]. Our measurements with similar valves showed comparable behaviour and the high leakage like the hydro-mechanical controller (DFR), so the digital seat valve based HPC noticeable exploits its strength vs. both technologies. This energy saving will also help to reduce losses at the upcoming electrification of mobile machines [6].

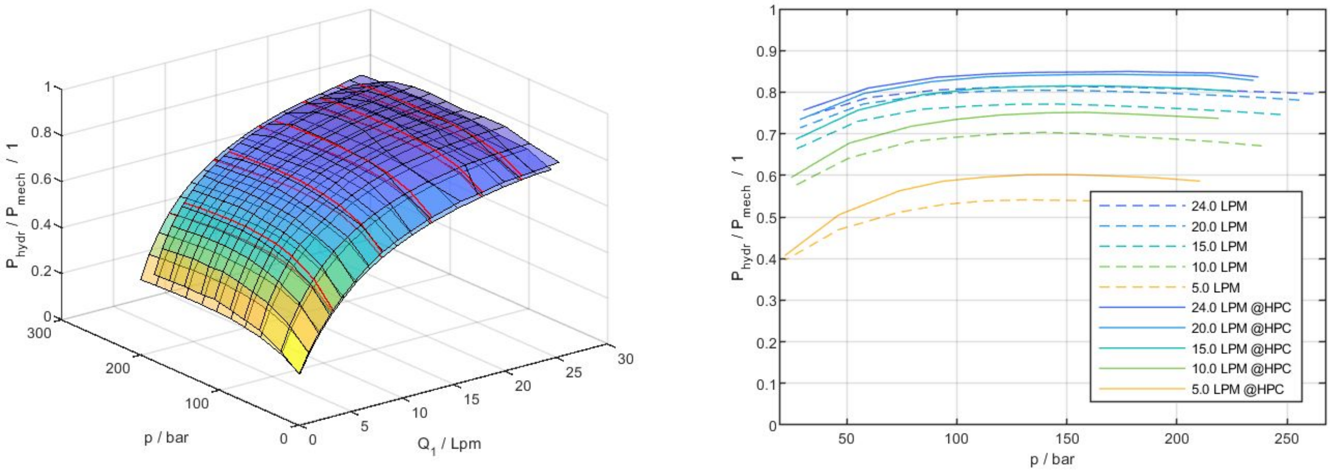

Figure 4: efficiency of 28ccm Bosch Rexroth A10 pump; hydro-mechanical vs. HPC controller

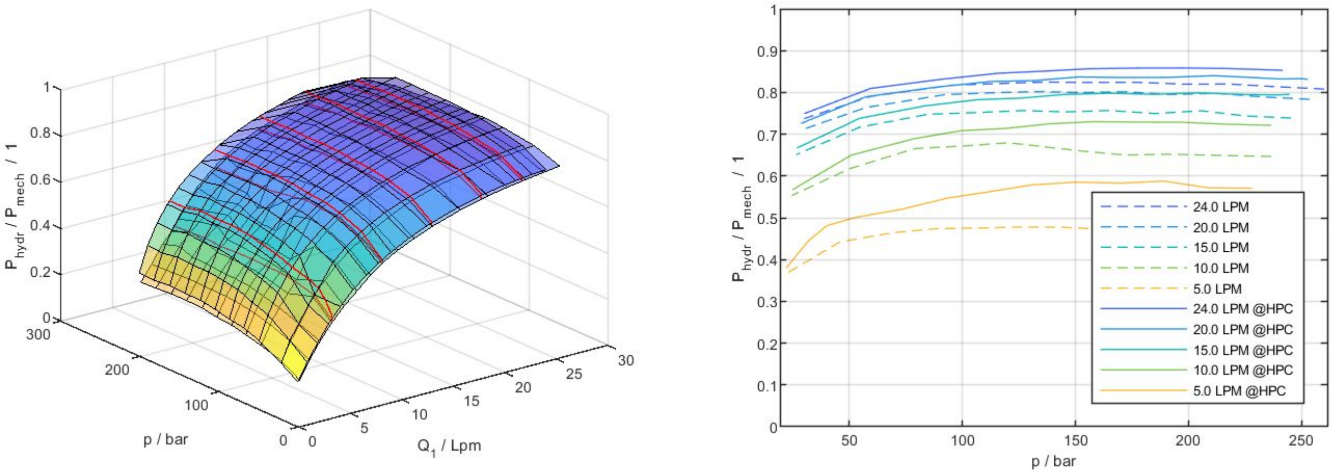

Figure 5: efficiency of 28ccm Liebherr pump; hydro-mechanical vs. HPC controller

2.2. Dynamic Behaviour

Besides the efficiency also the dynamic behavior of the pump-controller combination could be performed as it was formerly done [1] but in a more exact way (**Figure 6**, **Figure 7**, **Figure 8**).

The test cycle consists of several system excitations to determine the controller and UUT dynamics but also the accuracy. Starting with constant flow rates at 2 different states using constant xValve positions and the controller keeping the pressure drop at 16 bar the system pressures are ramped up and down at different pressure rates. In the second part of the test cycle (time > 120s) the system pressures should be kept at constant pressure levels while the flow rates are stepwise changed. Due to these impacts it is possible to show whether the controller works fast enough and if there are any oscillations at some operation point. Another benefit of such a challenging test cycle is the possibility to compare the different controller settings and pump reactions at extreme stress.

The overall goal of the comparison with the existing hydro-mechanical controller was not only to be more efficient but also to reduce oscillations (as shown in **Figure 6** between the time period of 80s to 145s) without being less dynamical (rising or reducing pressure rates). Of course, another great advantage of an electrically actuated system is the possibility to change the controller's characteristic just in software without any mechanical modification even in already operating conditions.

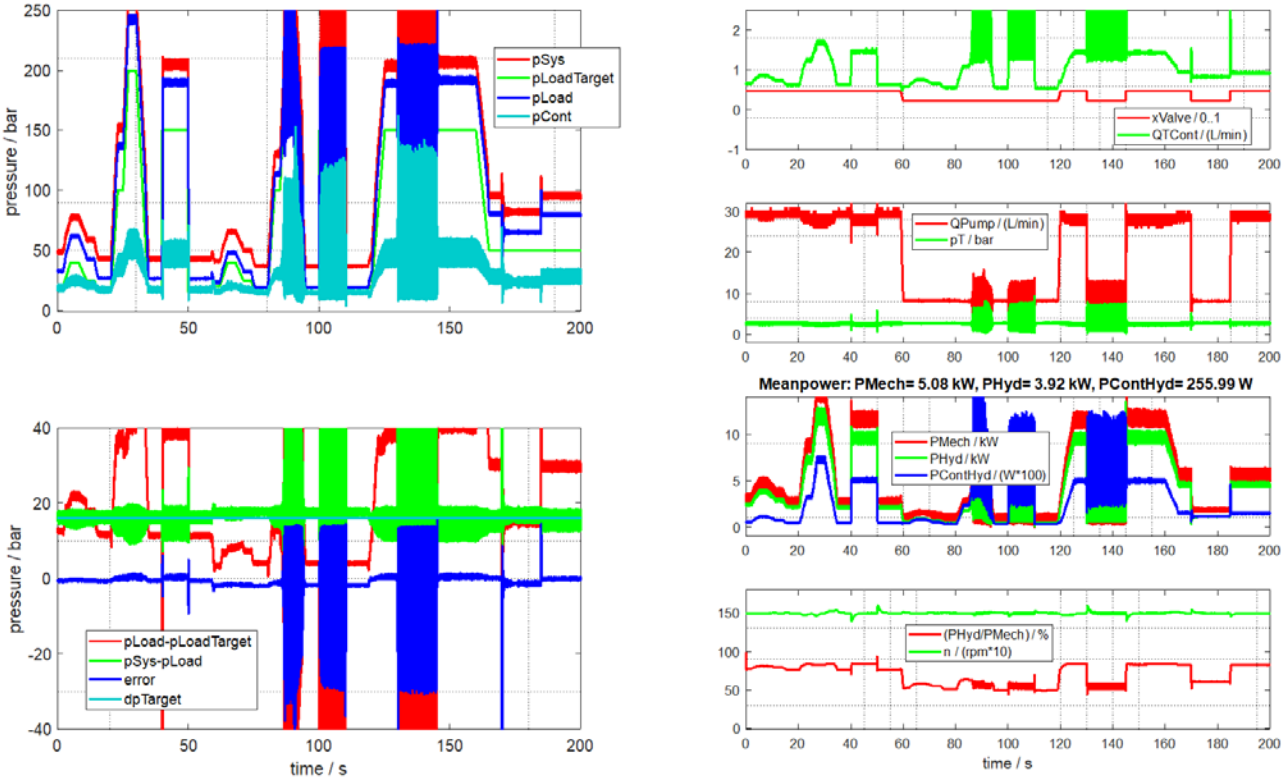

Figure 6: test cycle with Liebherr 28ccm and mechanical DFR controller

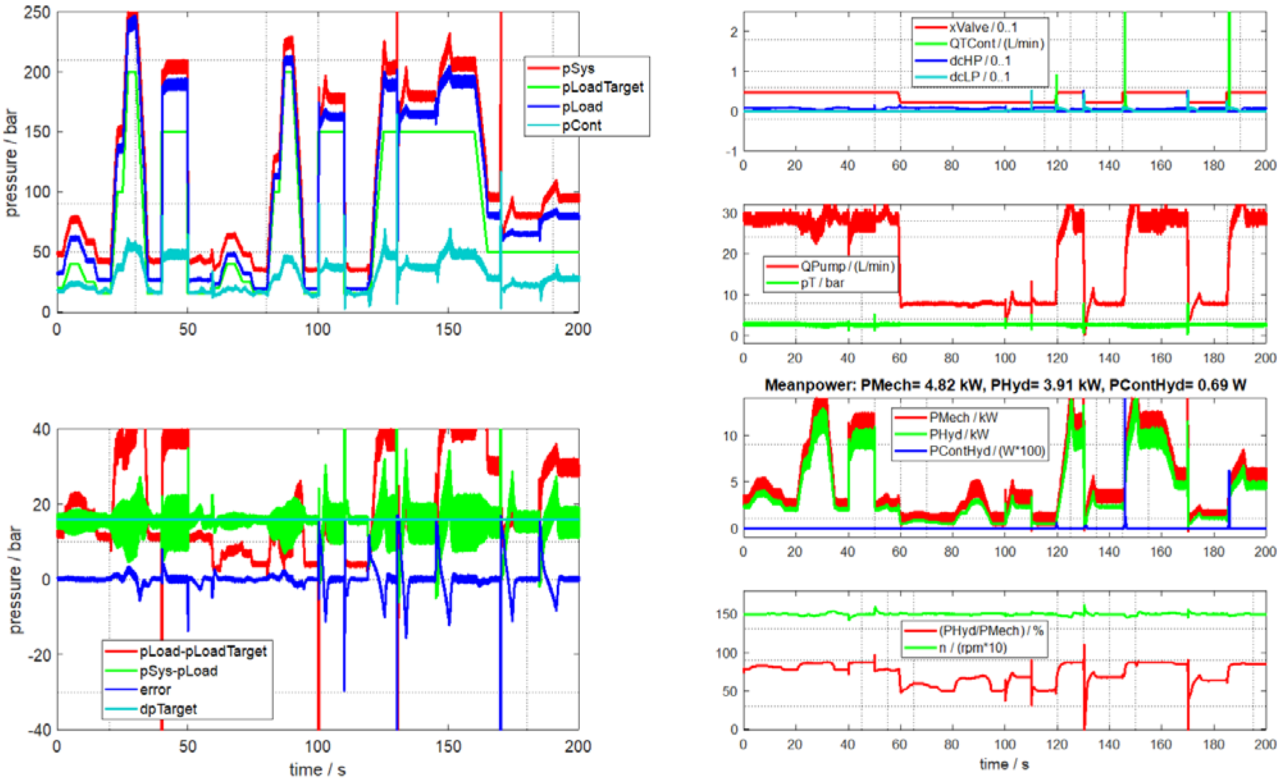

Figure 7: test cycle with Liebherr 28ccm and HPC controller

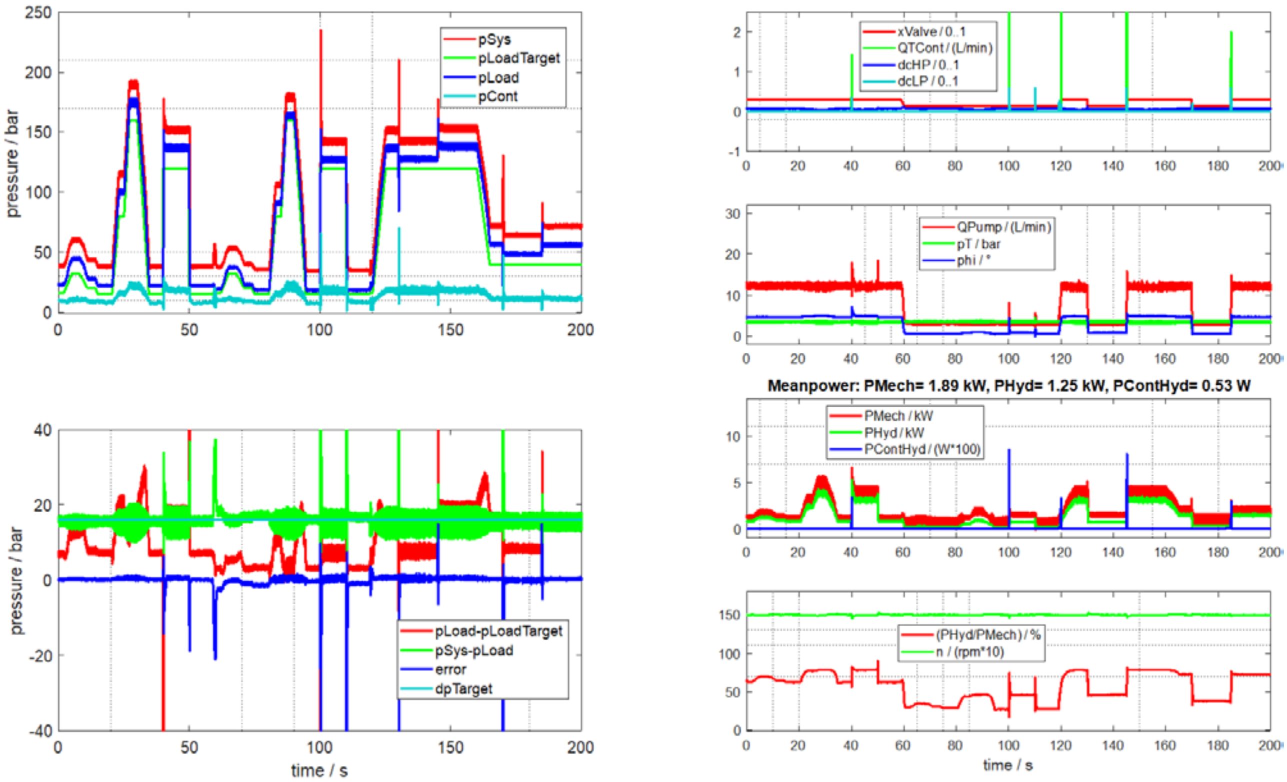

Figure 8: test cycle with Bosch Rexroth A10 28ccm and HPC controller with optimized parameters

The tests were performed with several variable displacement pumps from 18ccm to 32ccm of different suppliers which showed to be controllable out of the box without severe changing of the controller parameters. The performance could be improved with small adjustments during the measurements (e.g. **Figure 7** with standard; **Figure 8** with optimized parameters resulting in a shorter settling time and less noise). These promising results led to the next step, the proof of concept on a real working machine without the fear of serious problems.

2.3. Comparison of power consumption

Table 1 shows the power distribution of a Bosch Rexroth A10 (18ccm) with different controller types. The mean powers over one test cycle are listed and the corresponding power consumed by the controller split in electric and hydraulic power. Of course, the DFR controller doesn't consume any electrical power. It can be seen that the HPC controller is roughly 13% better than the DFR and about 9% better than using a standard proportional valve, in terms of energy efficiency.

Table 1: comparison of mean powers over a test cycle using Bosch Rexroth A10, 18ccm

	HPC with PULSE valves	Standard proportional valve	hydro-mechanical DFR
P_{mech} / (kW)	2.01	2.32	2.37
P_{hyd} / (kW)	1.5	1.54	1.47
$P_{electric,controller}$ / (W)	0.76	6.1	-
$P_{hydraulic,controller}$ / (W)	1	252	385
η_{tot} / (1)	75%	66%	62%

3. PROOF OF CONCEPT ON A REAL WORKING MACHINE

Even though majority of development and testing of a pump controller could be carried out in laboratory, proof of concept testing on real applications is necessary. LCM, IHA and ALH decided to begin real application testing at IHA mobile machine test area located in Tampere Finland. Forwarder was selected as a test platform for the pump controller studied. The Ponsse Caribou S10 forwarder (**Figure 9**) is a machine used in cut-to-length (CTL) timber harvesting. The CTL method utilizes two machine types: 1) the harvester felling, delimbing and cutting, 2) the forwarder picking and transporting the piles of logs to roadside [3].

Figure 9: Ponsse Caribou forwarder as a test platform for the pump controller

The work functions of the forwarder include: slew, lift, luffing, extension, grapple rotator and grapple. These functions are realized using hydraulic actuators each powered by a common open circuit variable displacement pump (Bosch Rexroth 71 ccm A10 series) in load sensing configuration. The actuators are controlled by Parker K170LS mobile proportional directional control valve as presented in the simplified hydraulic circuit diagram in **Figure 10**.

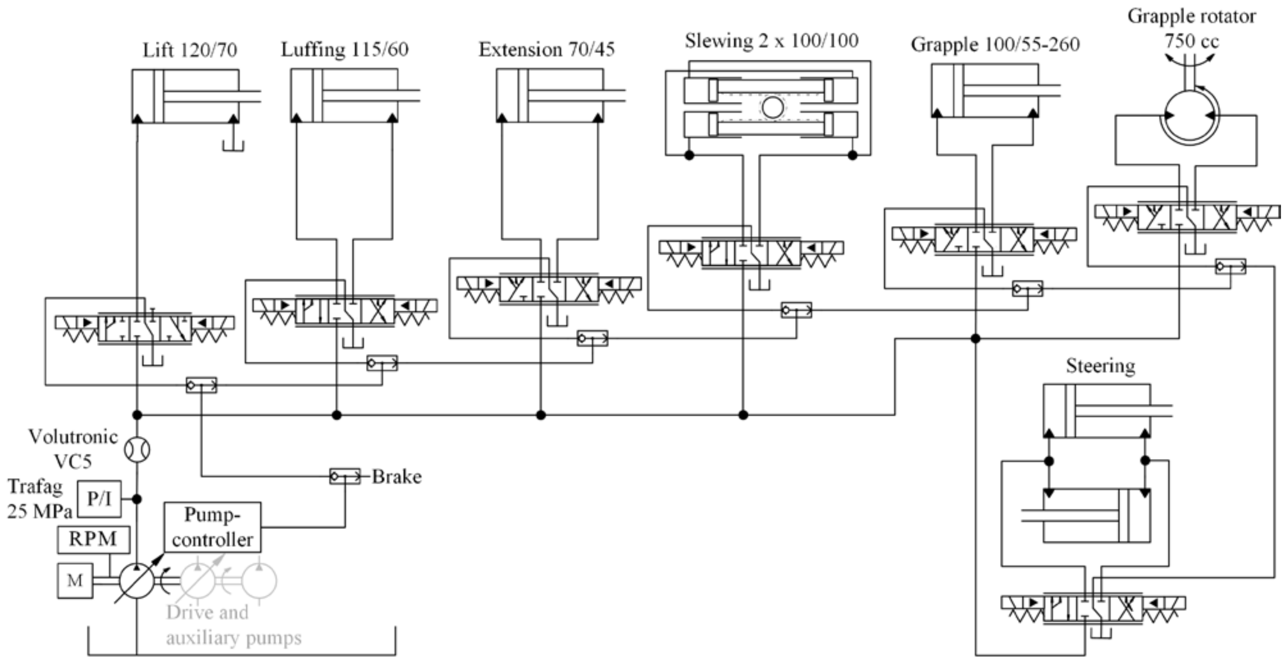

Figure 10: Simplified hydraulic circuit diagram of the forwarder

The data acquisition and control system is presented in **Figure 11**. The 4/3 control valves of the hydraulic actuators are driven by IFM current controller which receives the coil current references from the Epec 5050 main controller. The main controller measures joystick signals, boom angles, cylinder positions and part of the actuator pressures. The rest of the pressures are measured using CBX-CAN converter. Crane slewing angle is measured with resolver which sends angle information to main controller via CAN bus.

The second Epec 5050 unit (Pump controller) measures diesel engine rotational speed, LS-pressure and pump angle. All the measured data is transmitted via CAN buses to Kvaser Memorator which is used here for data acquisition. Measurement results are post-processed and presented in MATLAB.

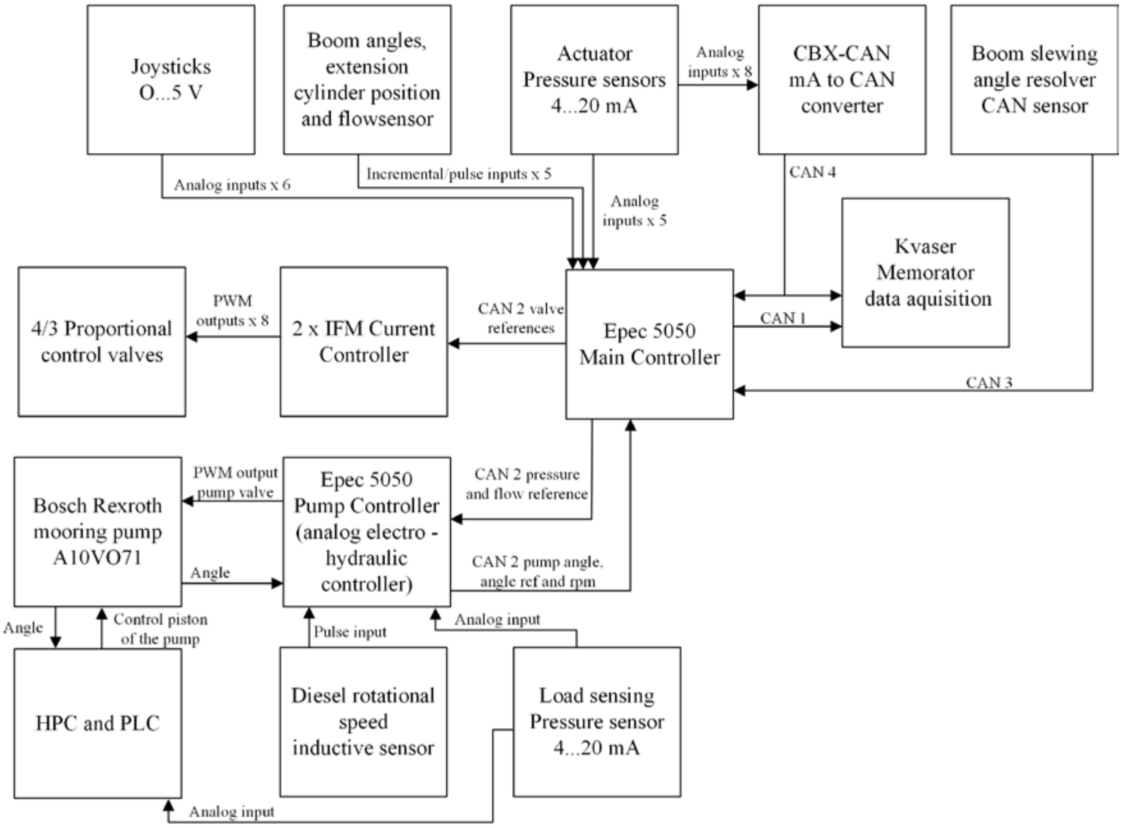

Figure 11: Data acquisition and control system used in the proof-of-concept testing

Figure 11 show both the HPC controller and the EPEC 5050 controller unit for the analog electro-hydraulic pump controller (EHC) using proportional valve from Bosch Rexroth. Naturally, only single pump controller is active at a time during the tests. The controller unit for the EHC uses the LS-pressure signal measured from the control valve LS-port to generate the pump pressure reference. The pressure reference is shaped using non-symmetrical rate limitation which allows fast increase of pressure, but slows down the decrease of pressure significantly. This results in relatively fast reaction to increased load pressure, while maintaining sufficient damping characteristics of the actuators and good operator feel when operating the crane. The pump pressure controller is of cascaded structure, where the inner loop controls the pump displacement. The reference value for the pump displacement is a sum of a flow feedforward and the output of the outer control loop, which controls the pump pressure. The flow feedforward is generated based on the coil currents of the actuator control valves. More information of the data acquisition and control system is given in [4].

The measurements performed with the machine include step responses of the lift actuator. It is worth to note that the actuator command is rate limited thus resulting in fast ramp-wise command. The purpose of these is to enable repeatable tests with less operator influences. The tests are carried out with luffing and extension actuators at minimum position and without load in the grapple. Initial position of the lift actuator is set such that grapple touches the ground.

To prove the applicability of the proposed pump controller in real mobile machine operation, also loading of logs from ground level into the load space of the forwarder is tested. These work cycles include simultaneous operation of more than one actuator and represent typical use of the machine.

Figure 12 and **Figure 13** present the step responses of the lift actuator with the baseline analog electro-hydraulic pump controller EHC and the proposed HPC controller. First column of diagrams presents piston position and velocity of the lift actuator and its valve command. The recorded valve command curve shows the compensation of the dead zone and the ramp-wise rate limitation. The second column of diagrams presents piston side pressure of the lift actuator, pump pressure, LS-pressure, pump flow rate, pump displacement, mechanical output power of the lift actuator and the output power of the pump.

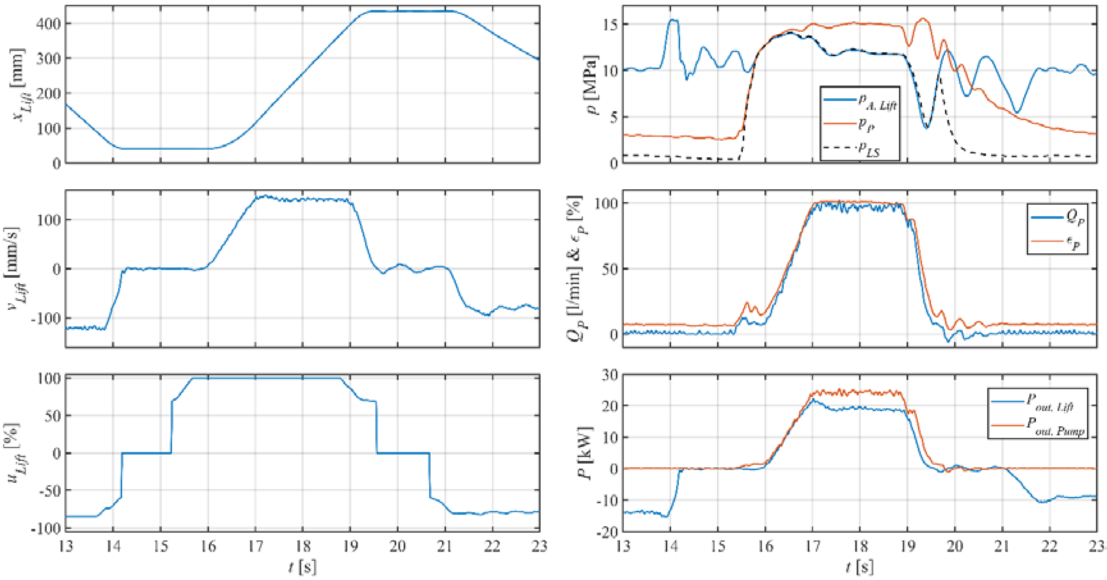

Figure 12: Lift actuator response with the EHC

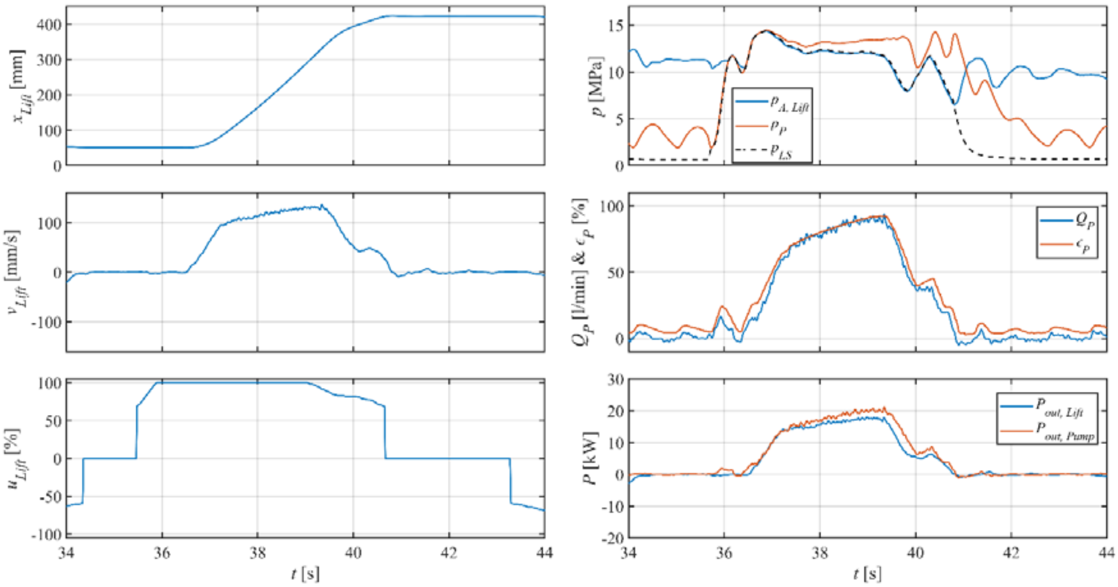

Figure 13: Lift actuator response with the HPC

Figure 14 presents an excerpt of the measured loading cycle. Piston positions and velocities of the four main actuators are presented as well as the pump pressure, LS-pressure, pump flow rate and displacement. The results show that the HPC controller is capable of controlling the load sensing pump pressure and the crane can be used for real loading work. At certain operating points, however, a some oscillation of the pump pressure can be observed, which will be further optimised. Despite slight oscillation of pump pressure in these first proof-of-concept tests, the overall operator feel of the machine equipped with the HPC and pressure compensated control valves is good.

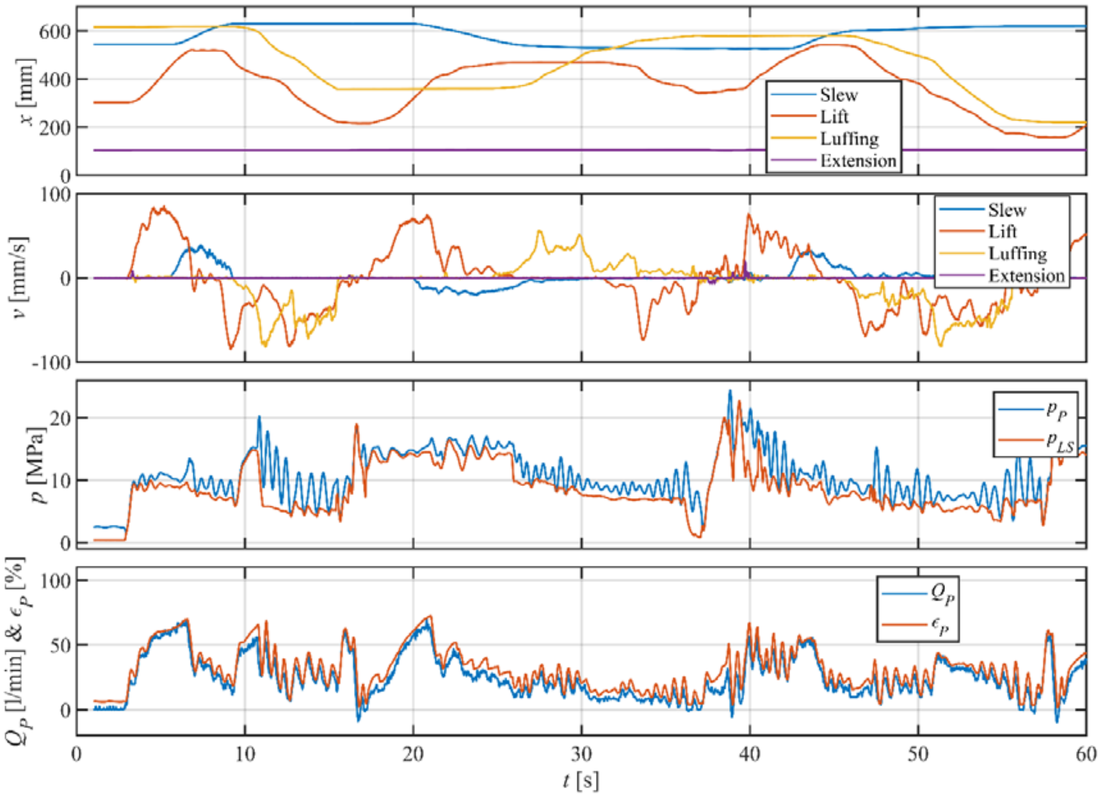

Figure 14: Excerpt of the measured loading cycle with HPC controller

4. CONCLUSION & OUTLOOK

The results of the tests in the laboratory have already shown that the Hydraulic Pulse Controller can be optimally combined with various pumps from different manufacturers. The structure of the controller is designed in such a way that tuning to a different pump is quickly possible. By modifying the test rig for better and measurable comparison of different controllers, it was also demonstrated that the efficiency increase was achieved. In the context of the tests in the laboratory the influencing variables from outside were less, but still these tests provide essential knowledge for the further development and the preparation of tests on complete machines.

The complexity to control a pump in a real working machine is a big challenge in many aspects. For example, the pump must supply oil to a number of actuators at the same time and react optimally to influences such as pressure changes, temperature or oscillations in order to ensure the operability of the machine. For this reason, it is also necessary to prove the suitability of the HPC in this environment. The pre-setup of the control parameters was originated from the lab tests and not specifically adjusted to the pump (71ccm A10 Bosch Rexroth) of the machine. From the first attempt, the HPC was able to control the pump according to the operator's requirements. The test period with the HPC installed on the machine was limited to a few hours, during which, with the help of some analysis, the controller parameters were further tuned until the performance was at a comparable level to the previously installed system. The full potential of the control quality was not yet fully exploited in this short time frame. However, the commissioning phase has already shown that the HPC can significantly reduce the time required here. In addition, adaptation to the respective application or a specific work cycle is much better possible. This is mainly realized by moving essential functions into software. Now it is possible to influence a characteristic or the behavior of the pump without adapting the mechanics. Another advantage is the connectivity between the machine and the pump, so that important information can be exchanged, such as information of the pump status.

In interaction with the machine, the HPC takes up the role of enabling the intelligence of the pump. The importance of software-based functions is growing, as this is the only way to react flexibly to new requirements according to [7]. In this paper, there was mostly only 'HPC' mentioned, but this has developed increasingly into a smart subsystem. An intelligent node in the bus-communication closes the gap in the network between the machine and the pump. This control system can be smoothly integrated into existing architectures. With this subsystem, further potentials are now available, such as optimized interaction of the entire hydraulic system to reduce the standby pressure. To validate these further advantages of the HPC, further tests will be carried out on the test bench and on mobile machines. The controller architecture will also be further developed and additional functions added. These include, for example model-based control including temperature dependencies and pump design parameters, or AI-based data evaluation including condition monitoring for early fault detection such as increasing leakage or aberrant behavior.

NOMENCLATURE

Variable	***Description***	***Unit***
Q_T	Volume flow of the controller measured at low pressure side	[L/min]
Q_1	Volume flow of pump to consumer (load simulation) at high pressure side	[L/min]
p_S, p_A, p_T, p_V	System pressure, controller p., lowside p., pressure used by consumer	[bar]
E_{hydr}	Hydraulic energy	[Ws]
P_{hydr}	Hydraulic power	[W]
P_{mech}	Mechanical power of the drive shaft	[W]
P_{cont}	Hydraulic power of controller	[W]
Q	Volume flow	[L/min]
M	Mechanical torque of the motor shaft	[Nm]
ω	Angular velocity	[rad/s]
p	Hydraulic pressure	[bar]
phi	Swivel angle of the variable displacement pump	[°]
dc	Duty cycle	[%]
η_{tot}	Overall efficiency factor	[1]

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