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MULTI-PRESSURE ACTUATOR IN ENHANCING THE ENERGY BALANCE OF MICRO-EXCAVATOR

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ABSTRACT

This paper presents results of an experimental study conducted with an electrified small sized excavator, whose boom swing function is equipped with digital hydraulic multi-pressure actuator. Highlighted and discussed are the performance, energy efficiency and controllability of the function in varying inertia loads. In addition, component-specific energy losses are analysed. This data is compared with the results gained with the excavator's original control arrangement, a load-sensing system based on pressure adjustment valve. Results manifest high energy saving potential of the multi-pressure system in mobile machinery.

KEYWORDS: digital multi-pressure actuator, micro-excavator, energy efficiency, control, experimental

1. INTRODUCTION

Tightening international legislation regarding the allowed NO_x and particle emissions of fossil fuel burning machines along with the rising fuel prices are directing the machine manufacturers to develop more environmentally friendly and energy efficient machines to keep their products both competitive and in accordance with the valid regulations.

When considering mobile work machinery, one of their most essential systems considering machine's performance and energy efficiency are their hydraulic systems that are used to operate various functions of the machine. If the drive transmission is left aside, the hydraulic systems, the work hydraulics, in these machines are typically realized as open circuit valve controlled central hydraulic systems. In these, the individually valve controlled actuators that operate the functions of the machine are connected in parallel and fed by a single pump or pump combination run by a combustion engine, typically a diesel-engine. Because of the direct interconnection between actuators and prime mover, the diesel-engine has to be dimensioned according to the combined maximum power required by the actuators at any instant. However, this maximum power may be required only at minor portion of the machine's work cycle, and most of the time the combined power requirement of the actuators can be significantly lower and have much variation. From the diesel-engine's point of view, this variation of power requirement means ever changing operating point and thus not constantly operating in the most energy-efficient one, which in turn significantly increases the fuel consumption and the emission of pollutants of the engine.

Feeding parallel connected and simultaneously operated actuators from a single supply line, as it is in the open circuit valve controlled central hydraulic systems, leads to a need to throttle the input and output flows of at least one of the actuators if the ratios of the actuator sizes and loads between the simultaneously operated actuators are not identical. This throttling is required to produce pressure losses that balance the existing differences in supply line pressure and load pressures between the actuators, and is therefore required always regardless whether the pump of the system is of fixed delivery flow type or variable delivery flow type, e.g. a Load-Sensing (LS) pump. Although the valve actuated throttling allows for very accurate control of actuator velocity and position, it induces high pressure losses and therefore also high power losses leading to low energy efficiency of the system and because of this, typically a need for a cooling system that further reduces the energy efficiency of the system.

Since the energy-efficiency related weaknesses of open circuit valve controlled central hydraulic systems are due to incapability to match the prime mover produced power with the power requirements of parallel connected actuators without intentionally induced power losses, which in turn are due to the direct connection between the actuators and prime mover, the solution for higher system energy efficiency has to be sought by reducing or even totally removing both the interdependency between actuators and the direct connection between the prime mover and actuators. During recent years, some new system architectures have been proposed to reach these targets.

Of these, the quite widely studied electro-hydraulic actuators (EHAs), e.g. [1, 2, 3] are based on one motor – one pump – one actuator –principle, where the actuator is directly controlled by the pump without any valves. The control of the pump flow is typically implemented through the control of the rotational velocity of the motor running the pump, although pump displacement control or combined velocity-displacement control are also possible. Even though there is a direct interdependency between the prime mover and actuator, this solution enables very good matching of the actuator required power and the motor produced power and thus reaching high system efficiencies in absence of throttling losses. Another advantage is that from hydraulic point of view each actuator constitutes a self-sufficient power pack unit not dependent on the other actuators of the machine, and requires only the electric energy and control signal from outside of the pack. This autonomy enables the dimensioning of machine's each power pack according to the requirements of the function that they are intended to operate. It also enables their relatively easy adoption for various functions in different machines. The major disadvantage of this system structure is the relatively high commissioning price compared to traditional open circuit valve controlled central hydraulic systems, since each actuator requires its own pump, prime mover and control drive.

Some alternative solutions for reducing the interdependence both between system's actuators and between prime mover and actuators were presented in [4]. Amongst these was one, which was based on several pressure sources for each of the actuators of a hydraulic system. In following research works this concept was developed further and named digital hydraulic multi-pressure actuator (DHMPA). In this concept, each of the available pressure sources can be optionally connected to whichever actuator chamber. This enables a variety of actuator produced forces in both directions of movement and thus also actuator's adaptation to several operating points, the number of which naturally depends on the number of source pressures. The selectable pressures are generated through an actuator specific high-pressure accumulator and a set of pressure transformers between the accumulator and actuator, and the pressure sources to be connected to the actuator chambers are selected via a group of low pressure loss 2/2 on/off -valves, a detailed structure of the system is presented in Chapter 2. The charge of the high-pressure accumulator is maintained by a pump that is common to all of the digital hydraulic multi-pressure actuator units of the machine, Figure 1. Since the occasionally required actuator power peaks can be dealt with using the pressure transformers, this pump and its drive motor can be dimensioned according to lower power requirements. Despite of the common pump, each actuator can be operated without mutual dependence and the pressure transformers practically remove the direct connection between the actuators and the prime mover. This enables both reaching high energy-efficiency of the system and possibility to run the prime mover in its most energy-efficient operating point while charging the pressure accumulators. As a consequence, the energy (fuel) consumption and the pollution emissions of the machine are lowered compared to a machine with open circuit valve controlled central hydraulic systems.

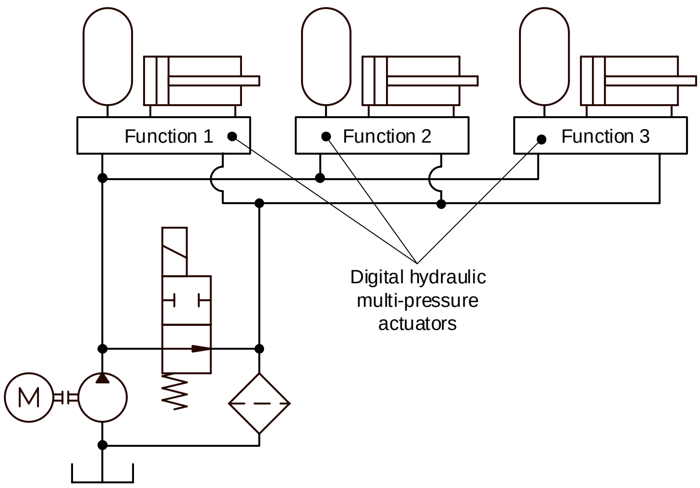

Figure 1. Circuit diagram of a system implemented with digital hydraulic multi-pressure actuators.

The digital hydraulic multi-pressure actuator was proven functional and energy-efficient already with the measurements with the first prototype in [5]. Later it has been tested with four different control strategies [6, 7] with promising results; the energy savings compared to traditional proportional valve based Load-Sensing (LS) system varied between 58–77 %, it has high dynamics, small position tracking error, but on the other hand, at slow actuator velocities the control resolution was found to require improvement to enable smooth tracking, accurate positioning and to avoid oscillations during stopping of the movement. Although the system efficiency was proven to be high, there is room for improvement considering the pressure losses of 2/2 on/off -valves, which were the highest in the system. The proof-of-concept prototype used in these measurements was constructed using separate components connected and by hoses, which highly likely had impact on the gained results.

In this study, the second prototype of the digital hydraulic multi-pressure actuator with more compact structure is presented and applied to operate the boom swing function of an electrically driven micro excavator. The challenge of this function is that the boom inertia is not constant, but changes depending on the position of the boom, i.e. how extended or folded the boom structure is, and also on the state of the boom bucket (empty-full). The effect of the changing inertia has to be taken into account in the control of the swing motion.

Focus of this study is in experimental research considering the energy efficiency, controllability, and performance of the multi-pressure actuator based system. The acquired results are compared to measurement results gained with the original control arrangement of the excavator, which consisted of fixed displacement pump running with constant velocity and a LS-system realized purely with valves. In addition, the component wise energy losses are analysed. The study ends with recommendations for further development and research.

2. METHODS

The methods used in this paper include experimental boom turning tests in an actual working machine, involving both the original pressure adjustment valve-operated LS-system and the digital hydraulic multi-pressure actuator system. Analytical power loss analyses of both systems have also been conducted in order to determine most important targets for energy saving.

2.1. Test rig

The digital hydraulic multi-pressure actuator was installed to drive the boom swing function of a small-sized working machine, a 1.1 tonne JCB Micro excavator. Originally the machine had been equipped with 13.6 kW diesel-engine, but in conducted electrification projects it had been replaced with a 10 kW variable speed electric motor and a battery pack. The work hydraulics for the four boom functions is a low cost Load-Sensing

(LS) system comprising of a fixed displacement pump and a pressure adjustment valve, which senses (using shuttle valves) the highest prevailing load pressure of the actuators and adjusts the system pressure such that it is approximately 20 bar higher than the highest load pressure. At the same time, it directs back to tank the portion of the fixed pump flow that is not needed in the actuators.

This system had been modified by replacing the original manually controlled proportional directional valves with electrically controlled valves, but otherwise the control structure remained as original. A simplified system schematics presenting only one function is shown in Figure 2, a more detailed description of this system structure with measured characteristics is presented in [8], although those measurements did not include the boom swing function, but these were conducted separately for this study for baseline reference. Despite of the possibility to vary the rotational velocity of the electric motor and thus also the pump outflow, in the measurements presented in this paper the motor was run with fixed speed, the purpose of this being to emulate a simple valve-based LS-system used typically in small work machines. Although not used in this study, it would be possible with this test rig to imitate the operation of a conventional LS-system by controlling the rotational speed of the electric motor.

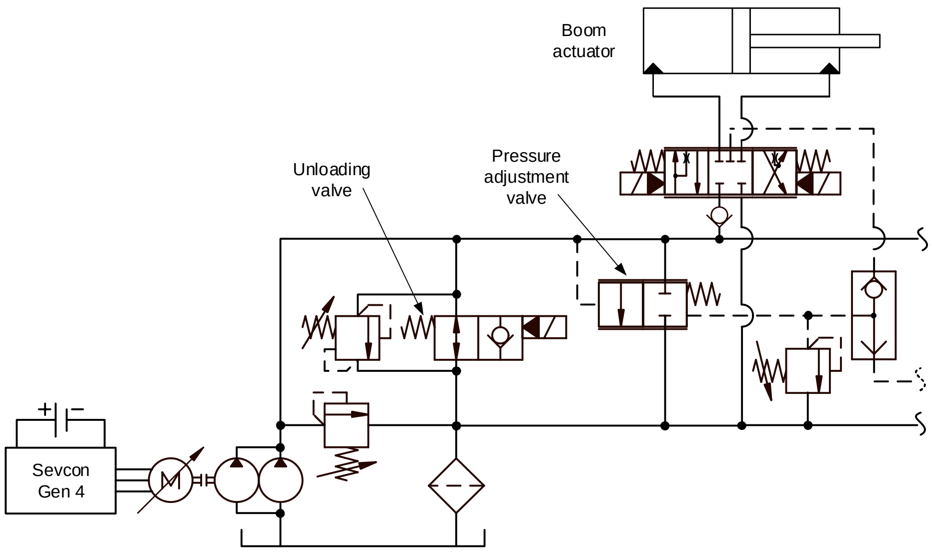

Figure 2. A section of the original valve-based LS-system of the excavator.

The excavator itself is presented in Figure 3. As the excavator is used as test rig in several studies, the individual boom functions are modified according to the need of the research. In the figure, the arm actuator is operated by an EHA unit attached to the boom, while the digital hydraulic multi-pressure unit used for the boom swing actuator is installed on the backside of the excavator and presented from above in the photo on the left.

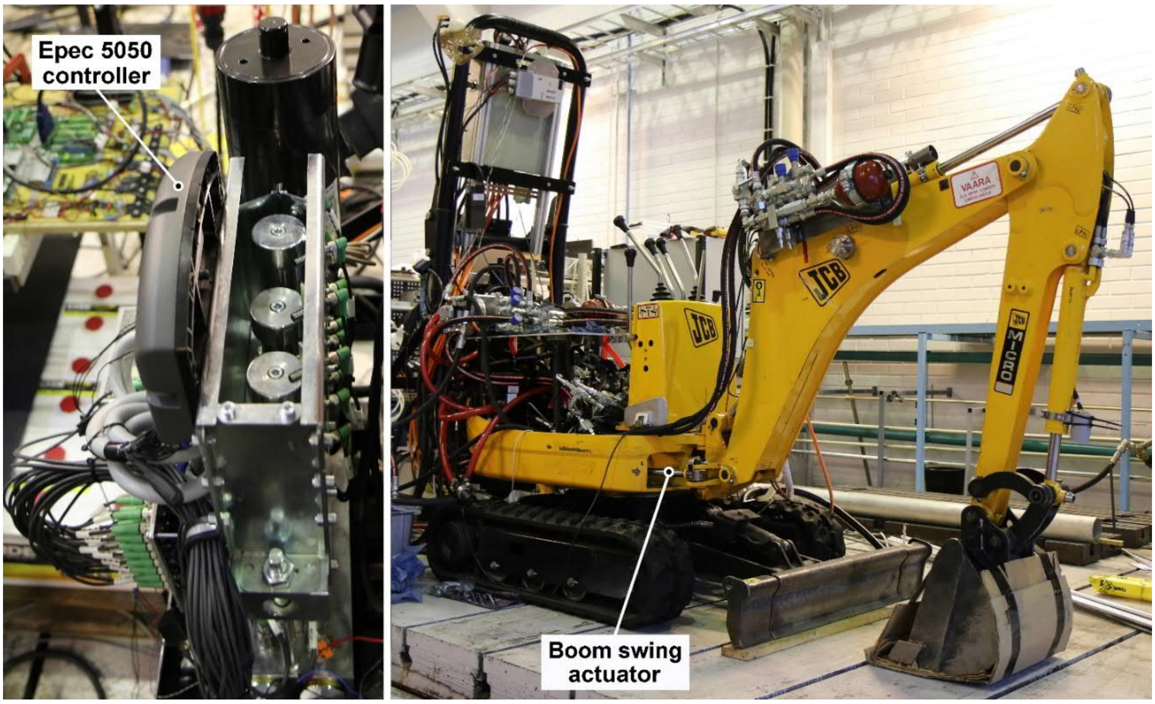

Figure 3. Digital hydraulic multi-pressure unit and excavator test rig.

The structure of the multi-pressure unit and its adaptation to boom swing function of the excavator is presented in Figure 4 with the main system components listed in Table 1. All the components of the multi-pressure unit are installed in a single manifold.

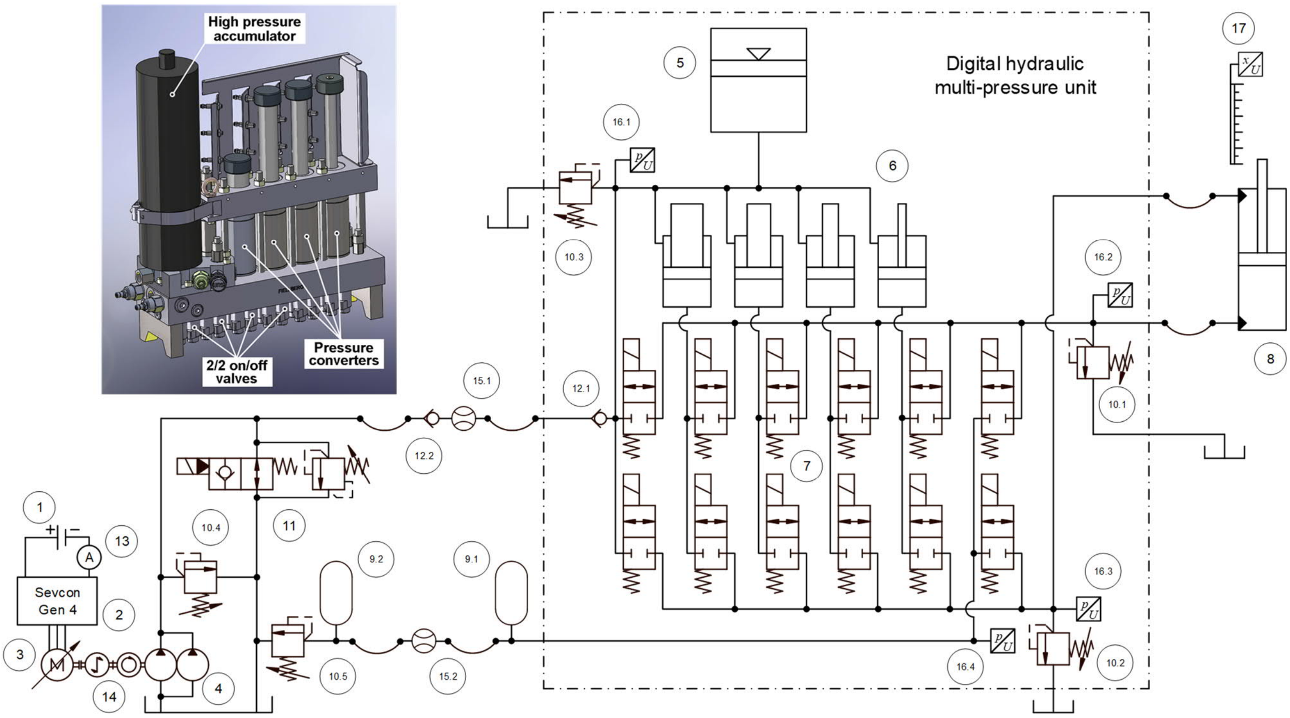

Figure 4. Digital hydraulic multi-pressure unit and its adaptation to boom swing function.

Table 1. Components of measurement system.

Nr	Description	Details
1	Battery pack	72 V Lead-acid
2	Motor controller	Sevcon Gen4
3	Electric motor	10 kW
4	Dual pump	Parker PGP511 (2* 6 cm^3/r)
5	High-pressure accumulator	2 litres
6	Pressure converters (piston Ø / piston rod Ø – stroke)	50/40-120 mm 50/36-120 mm 50/30-120 mm 50/22-120 mm
7	Doubled 2/2 on/off -valves	Bucher W22-D-5
8	Boom swing actuator (piston Ø / piston rod Ø – stroke)	52/25 - 293 mm
9.1 - 9.2	Low-pressure accumulator	1.0, 0.7 litre
10.1 - 10.5	Pressure relief valve	Rexroth VSC-30
11	Unloading valve	Rexroth VEI8A-VSBN-08A
12.1 - 12.2	Check valve	HBS ½” 0.5 bar
13	Current sensor	LEM DK 200
14	Torque and tachometer	Kistler 4502
15.1 - 15.2	Flow sensor	Kracht VC 0.4 - VC 1
16.1 - 16.4	Pressure sensor	Trafag NAH 8254
17	Position sensor	Siko SGH10

When started, the dual pump (4) driven by the speed-controlled electric motor (3) charges the high-pressure piston type accumulator (5) through check valves (12.1, 12.2) until the highest set charge pressure ~55 bar is reached, after which the controller of the multi-pressure unit shuts down the electric motor. As a result, there are six selectable pressures to connect to the boom swing actuator (8) chambers; the pressure prevailing in the high-pressure accumulator (5) directly, the four pressures generated by the accumulator and the four converter cylinders (6), and the pressure prevailing in the low-pressure accumulator (9.1) placed in the return line. When the boom swing actuator is given a command to move with certain velocity, the controller of the multi-pressure unit (Figure 5), which uses measured system pressures and actuator velocity as feedback signals, opens appropriate 2/2 on/off valves (7, representing six pairs of parallel connected valves) and connects pressures optimal to prevailing operating condition to the actuator chambers. The pressure relief valves (10) secure the system against excessive pressures, although the function of valve (10.5) is to maintain a back pressure of ~8 bar in tank line.

During the motion of the actuator, the controller keeps track of the positions of the converter cylinders and prevents them from running to their ends by changing the pressures connected to actuator chambers when necessary, i.e. when the required actuator movement requires more fluid volume than one converter cylinder can supply. The positions of the converter cylinders are maintained in acceptable regions by connecting their piston sides to appropriate pressure source when necessary. The control principle of the multi-pressure unit is described in detail in [5, 6, 7]. When the pressure of high-pressure accumulator decreases and reaches set minimum value of ~42 bar, the electric motor is started and the accumulator is charged again to its highest set pressure. In the measurements of this study, the electric motor was run with fixed rotational velocity, although the possibility to control the velocity could be utilized to increase the energy-efficiency of the accumulator charging process.

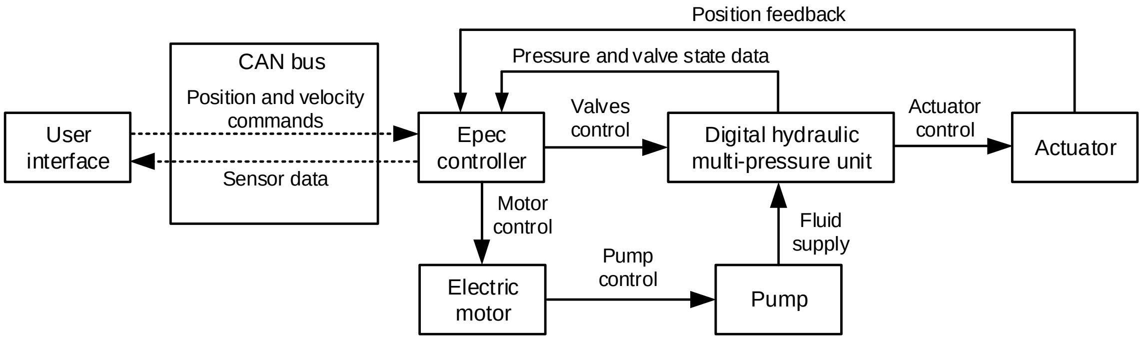

Figure 5. Control structure of the digital hydraulic multi-pressure system.

Control of the digital hydraulic multi-pressure actuator is built on Epec 5050 controller (see Fig. 3), which is able to control both all the 2/2 on/off -valves and the electric motor and thus also the pumps of the system, and in addition acquire sensor data and convey it to the user interface via CAN bus.

2.2. Experimental procedure

The characteristics and behavior of the digital hydraulic multi-pressure actuator were determined with measurements with excavator's boom swing function. Each single measurement consisted of velocity controlled boom deflection (i.e. swing actuator extension) from the excavator's centreline, followed by a short-timed immobility, after which the boom was returned (i.e. swing actuator retraction) to its starting position with controlled velocity. The loading of the system was varied by folding and extending the boom structure, two different positions were used. For comparison data, same measurements were repeated with a system that emulated the excavator's original valve-based LS-system.

During the measurements the following quantities were recorded; input current and voltage of the electric motor, output torque and rotational velocity of the electric motor, pump outlet flow rate and pressure, high-pressure accumulator's pressure, actuator chamber pressures, actuator position, low-pressure line pressure and flow rate. These were used in calculating the input and output powers and energies of system's components.

The positions of the pressure converter pistons and 2/2 on/off -valves were not recorded and are thus not presented here, but a description of their operation during actuator movements is explained in [6, 7].

3. RESULTS

This chapter presents the measurement results for both DHMPA and LS systems gained with low (folded boom) and high (extended boom) inertia, representing an inertia ratio of 1:10.

Figure 6 shows position trajectory command signals and actual actuator outputs for both directions of boom swing in the low and high inertia cases for both the digital hydraulic multi-pressure system and the LS-system.

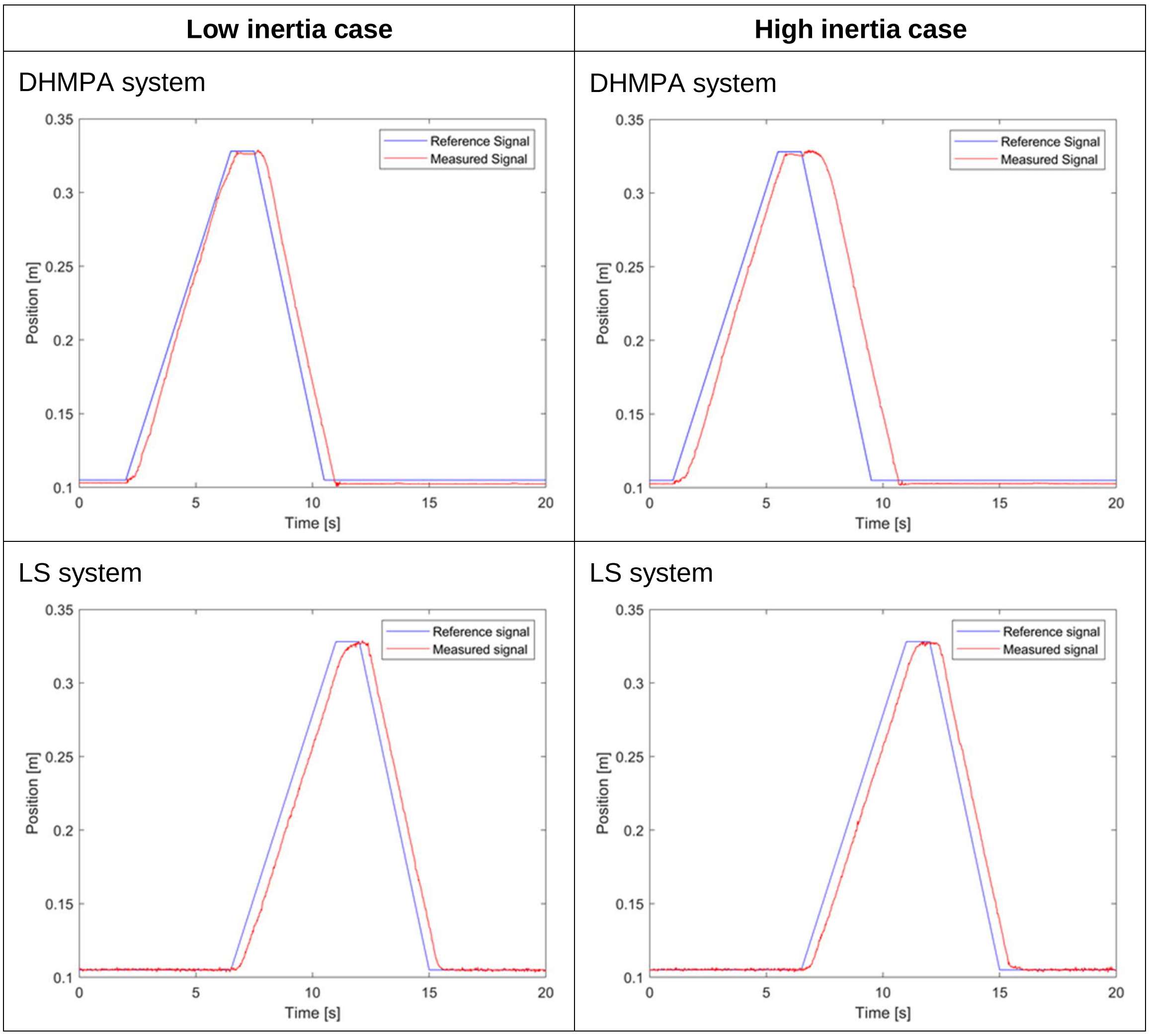

Figure 6. Position trajectories for the boom swing actuator with low and high boom inertia.

Figure 7 presents the multi-pressure system's high-pressure accumulator pressure, dual pump flow rate and boom swing actuator displacement during one work cycle. Accumulator charging control is based on pressostat (pressure switch) type of operation, the pump is turned on when the accumulator pressure reaches the minimum set pressure and it is turned off as the maximum set pressure is achieved. Only low inertia case is presented since differences between low and high inertia cases were negligible.

Figure 8 presents the power usage of the two studied systems in low and high inertia cases, i.e. with folded and extended boom structure. The plots include the input power of motor's inverter, the hydraulic pump's mechanical input power, and the pump's hydraulic output power. The electric motor had a tendency to creep with low velocity following the end of swing motion, i.e. during commanded immobility phase, which manifests itself in the measured electric input power. Since this unwanted phenomenon is later to be removed, the plots also present electric motor's input power calculated without this creeping.

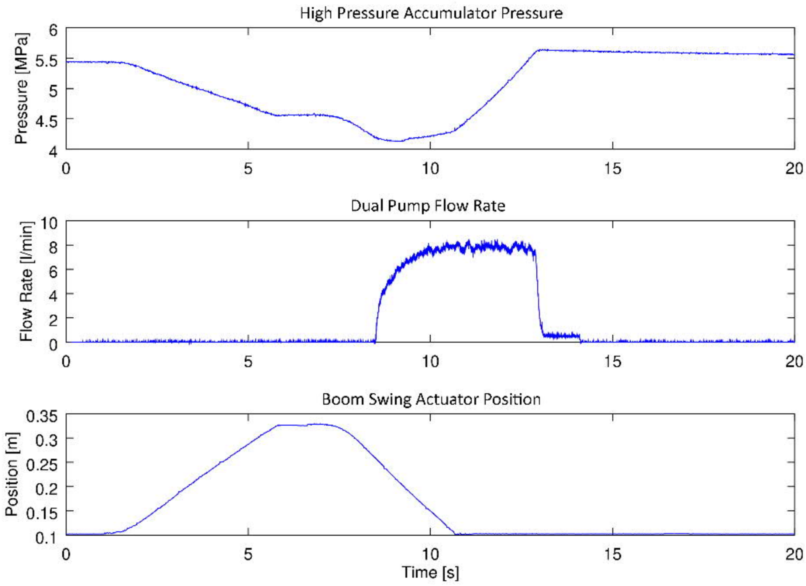

Figure 7. High-pressure accumulator pressure, dual pump flow rate and boom swing actuator position during work cycle, DHMPA system in low inertia case.

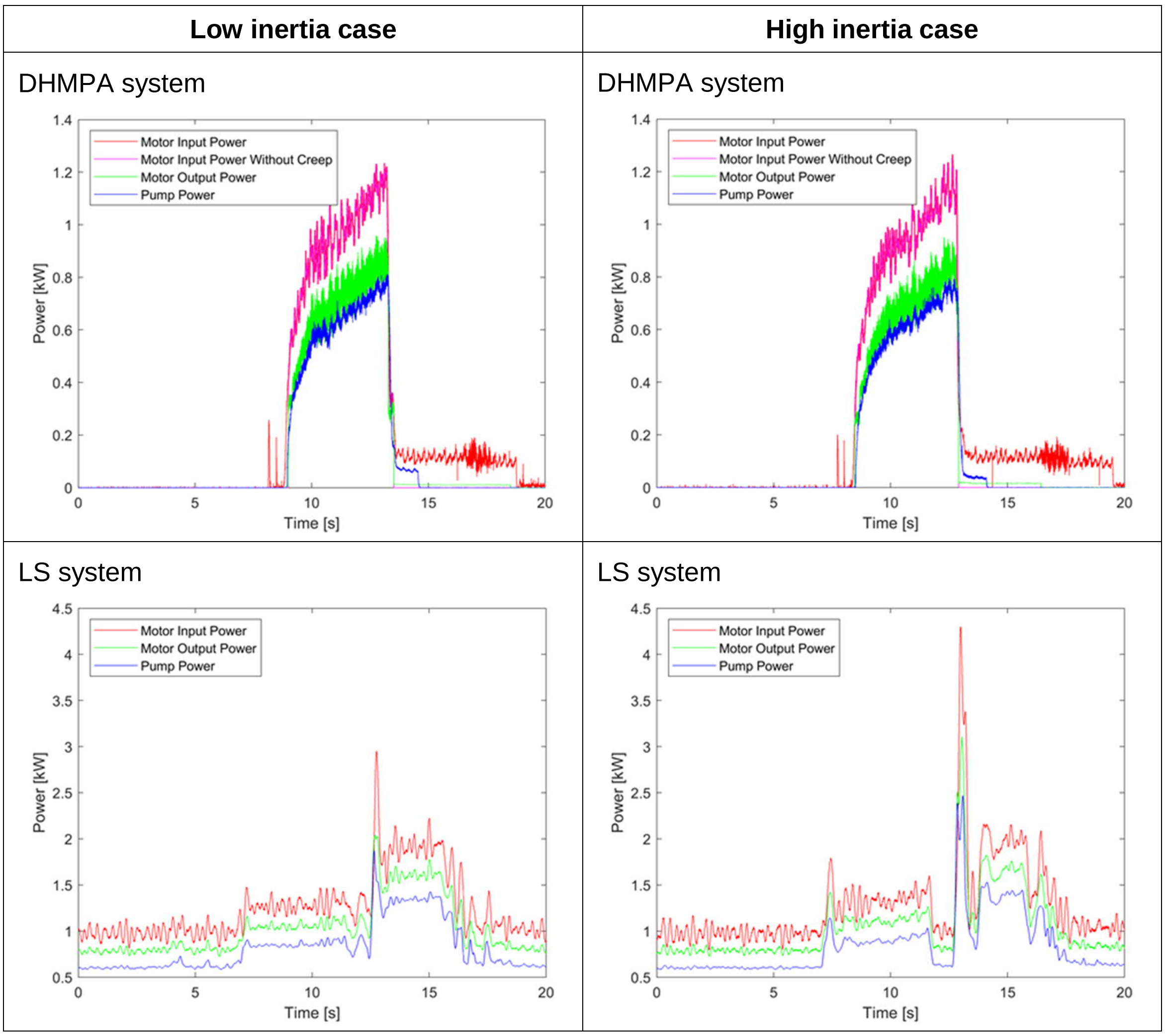

Figure 8. Power usage of studied systems during one work cycle with low and high boom inertia. Note the different power scales in upper and lower figures.

Figure 9 presents the cumulative energy consumption of the various sub-systems in low and high inertia cases. Electric motor's energy consumption is calculated both with the effect of creeping motion and without it.

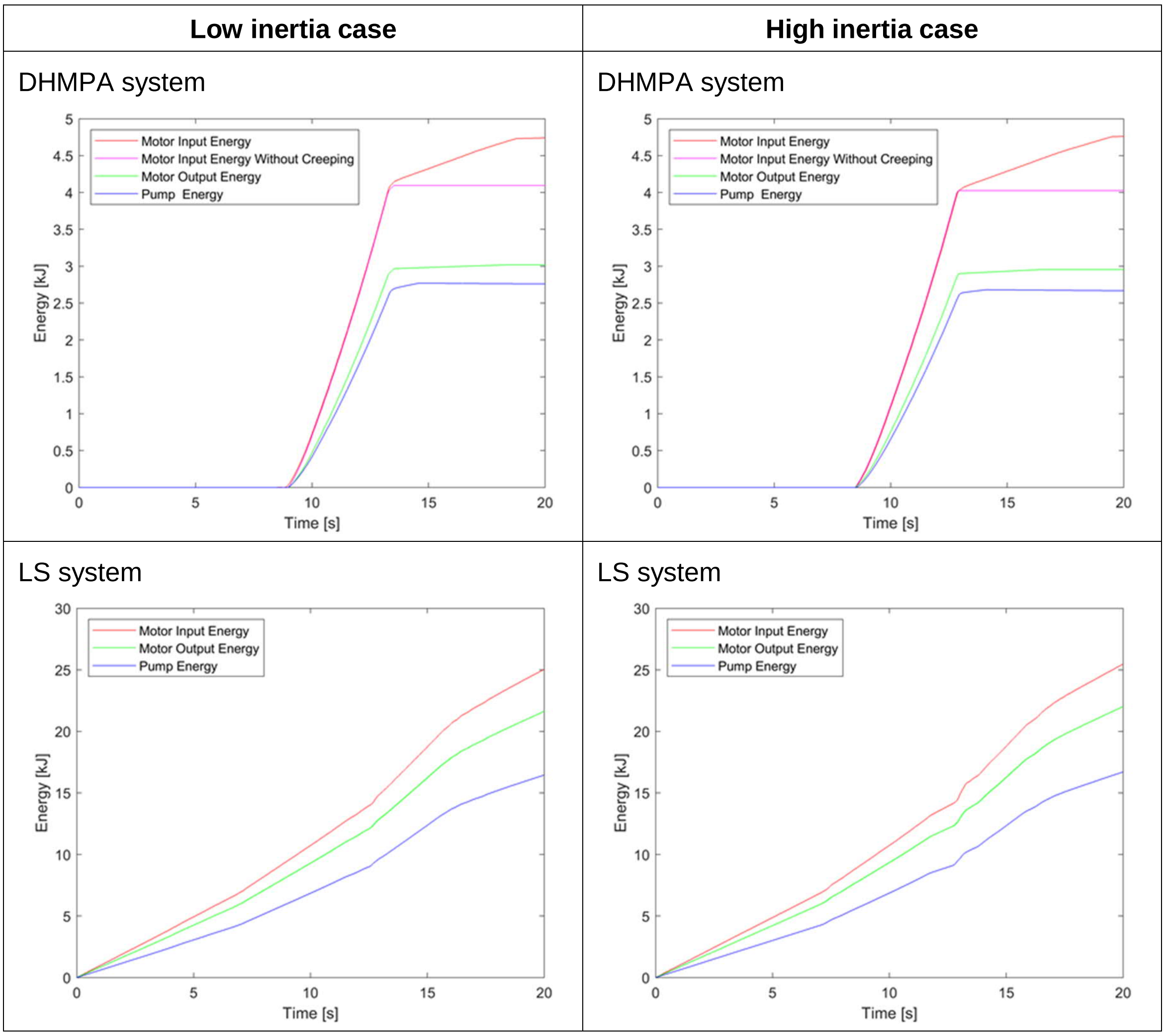

Figure 9. Cumulative energy consumption of studied systems during one work cycle with low and high boom inertia. Note the different energy scales in upper and lower figures.

Tables 2–4 present calculated peak powers, energy consumption and differences in efficiencies between digital hydraulic multi-pressure system and LS system.

Table 2. Peak powers of one back and forth swing motion in DHMPA and LS systems.

Power type	**Low inertia case**		**High inertia case**	
	DHMPA (kW)	**LS system** (kW)	**DHMPA** (kW)	**LS system** (kW)
Peak electric power	1.23	2.93	1.26	4.30
Peak mechanical power	0.95	2.03	0.95	3.10
Peak hydraulic power	0.81	1.85	0.79	2.46

Table 3. Energy consumption of one back and forth swing motion in DHMPA and LS systems.

Energy type	**Low inertia case**		**High inertia case**	
	DHMPA (kJ)	**LS system** (kJ)	**DHMPA** (kJ)	**LS system** (kJ)
Electric energy	4.09	14.92	4.01	15.43
Mechanical energy	3.01	12.91	2.95	13.33
Hydraulic energy	2.66	10.14	2.66	10.48

One reason for the substantial difference in power and energy consumptions between DHMPA and LS systems is that in order to be able to follow the position command, in LS system a pressure difference is needed across the proportional valve controlling the actuator. This necessitates that in the LS system the pump is constantly run at velocities that produce higher flow rate to the system than required by the actuator, whilst in DHMPA system, where there is no direct connection between pump and actuator, the pump is run only according to the real flow rate need of the accumulator. In LS system, the produced excess flow is directed back to tank through the pressure adjustment valve (see Fig. 2), whose setting value for the pressure difference between the pump and the actuator pressures is 20 bar. Despite of maintaining the system-specific pressure difference across the proportional valve, in the measurements the valve opening had to be kept close to its maximum value in order to achieve the demanded actuator velocities. For these reasons, a straight comparison of the net energy consumptions between the two systems can be misleading.

Table 4. Reduction of energy consumption with DHMPA compared to LS system.

Energy type	**Low inertia case** (%)	**High inertia case** (%)
Electric energy	72.6	74.0
Mechanical energy	76.7	77.9
Hydraulic energy	73.8	74.6

An analytical calculation of component-specific power losses were conducted for both systems to locate points of potential energy savings. In LS system, the losses caused by the mismatch in pump and actuator flow rates were excluded in this calculation. Thus the outcome of analysis are valid for both (ideal) pump and pressure adjustment valve operated LS-systems. The results related to the power consumptions for a one-directional boom swing motion (half cycle) are presented in Table 5. Table 6 presents the energy losses caused by charging of the high-pressure accumulator, which applies only to the DHMPA system.

Table 5. Estimated power consumption for one-directional swing motion.

Loss source	**DHMPA** (W)	**LS system** (W)
High-pressure accumulator seal friction	3.4	0.0
Pressure converter seal friction	5.8	0.0
Boom swing actuator (including cylinder seal)	220.0	220.0
2/2 on/off valves actuator inflow pressure loss	7.0	0.0
2/2 on/off valves actuator outflow pressure loss	3.2	0.0
Proportional valve actuator inflow pressure loss	0.0	233.0
Proportional valve actuator outflow pressure loss	0.0	180.0
Hose; actuator inflow pressure loss	26.8	26.8
Hose; actuator outflow pressure loss	15.8	15.8
Pressure relief valve pressure loss	71.8	0.0
Volume flow sensor (Kracht VC 1) pressure loss	4.5	4.5
Volume flow sensor (Kracht VC 0.4) pressure loss	0	8.2
Total power consumption in half cycle	**358.3**	**688.3**

Table 6. Estimated losses in high-pressure accumulator charging for one-directional swing motion.

Lost energy in accumulator charging	**DHMPA** (J)	**LS system** (J)
High-pressure accumulator thermal losses	56	0.0
High-pressure accumulator friction losses	7	0.0
Volume flow sensor (15.1 - Kracht VC 0.4) loss	29	0.0

The relatively low losses listed in Table 6 can be contrasted with the total energy consumption for the half cycle (obtained by time integration of the total power consumption and, for the DHMPA, by adding the high-pressure accumulator related losses): 1525 J for the DHMPA, and 2754 J for the LS system.

Note that if hose and volume flow sensor losses are omitted in both systems, the estimated energy consumption in DHMPA is circa 53% of the energy consumption in an LS system (with perfectly matched flow rate).

4. DISCUSSION

Both studied systems, LS and DHMPA, proved good trajectory following capability (Fig. 6), although in case of the latter system, the actuator load had greater impact on the tracking accuracy. The measurements with DHMPA were run with fixed control gains, and the noted dependency of accuracy on load could supposedly be answered by using load adaptive control gain.

The power and energy demands of the systems are clearly at different levels (Figs. 8 and 9; Tables 2 and 3). This is due both to the different operational principles of the systems and to the different magnitudes of systems' pressure losses. Since in DHMPA system there is no direct connection between the actuator and the pump, the latter is run only when the high-pressure accumulator requires charging and is shut down otherwise, whereas in LS system, where the direct connection exists, the pump runs constantly and thus also requires continuous, although not constant power input since the system pressure varies according to the operating situation. However, it must be noted, that the power and energy consumption of the DHMPA system during a boom swing motion depends on the dimensioning of the high pressure accumulator. The smaller the size the more frequent charging is required, and vice versa. Also if the load increases substantially, it leads to more frequent charging of the accumulator because of its fixed pressure and the fixed sized pressure converters. These differences in operational principles also give rise to the result that the magnitude of load inertia has negligible effect on the power and energy consumption of DHMPA system, but in LS system it has a major impact on these.

In LS system, the effect of increasing load force on energy consumption is due to the higher system pressure required, which increases both the power consumption of the fixed displacement – fixed speed pump and the pressure losses in the pressure adjustment valve of the studied one actuator LS system. In the studied case the high loads were predominantly associated with acceleration of the high inertia boom. In turn the deceleration of this boom structure would mean considerable energy losses since the braking mechanism in LS implementation is throttling by a proportional control valve. In DHMPA system the kinetic energy can be saved in the high-pressure accumulator and throttling losses are of minor importance, again.

In addition, in the studied system, the pump produced much higher flow rate than was needed by the boom swing actuator, which further increased the power losses in the pressure adjustment valve. In a multi-actuator system also the pressure losses of other actuators' control valves would change and probably contribute to the growth of power losses. On the other hand, if the pump could adapt to the changing need of flow rate (variable displacement pump or variable speed pump), then the effect of load on the power consumption of pump would be less. Also in DHMPA system, the magnitude of load naturally affects the power and energy need of the actuator, e.g. increasing load requires more power and energy, but this need is met internally between the high-pressure accumulator and the actuator through the pressure converters, and the effect of loading is seen from the outside of the hydraulic system only when the high-pressure accumulator requires re-charging. This in turn depends on the dimensioning of the system components and on the velocity and frequency on which the actuator is operated. In the studied DHMPA system the accumulator was charged once per one back and forth boom swing motion (see Fig. 7). In continuous operation, the re-charging of high-pressure accumulator would be repeated similarly every time, and from an energy consumption point of view it would require the same amount of energy. The only energy losses that would depend on the operational conditions of the actuator are the system's seal frictions and pressure losses (see Table 5).

In LS system, where the control of actuator is actualized with flow throttling proportional valve, the valve pressure losses required by the control are of magnitude 15–20 bar, but in DHMPA system, where the 2/2 on/off-valves are used only to control the direction of flow between pressure converters and actuator, not the magnitude, the pressure losses of the actuator controlling valves are significantly lower. When using two parallel connected on/off-valves, the pressure losses of valves can be well below 1 bar in the studied case. In applications, where higher flow rates are required due to, e.g. larger actuator, the system can be implemented with higher numbers of parallel connected on/off-valves to keep the pressure losses moderate. This approach does not work for LS systems, since a certain pressure loss across the actuator controlling valve (LS pressure) has to be ascertained to ensure the functionality of the system and the accuracy and sensitivity of the control.

The operational environment, a micro excavator, where the DHMPA unit was tested, was not optimized for this unit, which had some impairing effects on the measurement results. The unit had to be placed further away from the actuator than the proportional valve of the LS system, which necessitated using very long hoses, which in addition were of only 6 mm inner diameter (i.e. the size of the excavator's original hoses). Although the flow velocities in hoses were moderate (circa 4 m/s), the relatively high fluid viscosity gave rise to pressure losses that were estimated to be four times the losses of the on/off-valves. In spite of being still much lower than the pressure losses of proportional valve in LS system, this elevated pressure loss had a significant effect on DHMPA system, which otherwise proved to be very energy efficient. The remedy for this is easy and affordable, since replacing the narrow hoses with 10 mm or 12 mm inner diameter hoses does not mean major investment, but has major impact on the pressure losses of hoses.

In addition to the above, also the power loss induced by the pressure relief valve placed in the low-pressure line of the DHMPA measurement system (10.5 in Fig. 4) has considerable influence on the energy efficiency of the system. Although being low in absolute value, the pressure loss in this valve is high in relation to the accumulator and system pressures, and therefore has a significant impairing effect on the energy efficiency. Paloniitty et al. have studied this problem and introduce several remedies [9]. Furthermore, the DHMPA measurement system was equipped with flow sensors, which are normally not included in similar systems. Like the other pressure loss sources, also these had some impairing effect on the energy efficiency.

When considering high-pressure accumulator, its observed pressure losses after charging were moderate indicating high thermal and overall efficiency of over 90 %. The peak pressures during charging were circa 56 bar and the steady state pressure after thermal stabilization was close to 54 bar. The limits set for accumulator's minimum and maximum pressures were close enough to each other to obtain moderate compression ratio and advantageous thermal efficiency. In addition, in the studied DHMPA system the accumulator fluid volume was high enough to enable the whole work cycle with only one recharge. Smaller sized accumulators would require more frequent re-charging whilst larger sized less frequent re-charging. Choosing a size is a question of optimizing the system's acquisition operation costs.

The fluid used in the measured systems was of viscosity class ISO VG 32. Even though it is a quite low class by nature, the fluid viscosity in DHMPA system remained at relatively high level because of the systems low general pressure losses, which in turn did not cause significant increase in systems temperature, but induced the above mentioned relatively high pressure losses in hoses. This is a phenomenon, which could be widely associated with new energy efficient fluid power systems. The power losses in these systems may be so low that the fluid either achieves balance temperature very slowly or the final temperature might remain close to the surrounding temperature. The fluid temperature was not continuously recorded during the measurements presented in this paper, but it was monitored occasionally and remained at level 20–25 °C because of the short measurement runs.

Despite the above described shortcomings in the measurement system, the energy efficiency of DHMPA system was significantly higher than that of LS system, and the reduction of consumed electric energy was 72–74% depending on the loading of the measured systems. In the case where hose and volume flow sensor losses are omitted in both systems, the estimated energy consumption in DHMPA would still be only circa 53% of the energy consumption in an LS system with perfectly matched flow rate.

5. CONCLUSIONS

This study presents experimental results of digital hydraulic multi-pressure actuator (DMHPA) system and LS system applied to a boom swing function of 1.1 tonne JCB Micro excavator. In DHMPA system, the major sources of power losses were identified to be the pressure losses of on/off-valves controlling the flow between pressure converters and actuator, friction losses in pressure converters, friction and thermal losses in high-pressure accumulator, pressure losses in ducts, and pressure losses in the pressure relief valve of the low-pressure line. However, when compared to the pressure and power losses in LS system, these were minuscule both in absolute value and in relation to system's input power. The measurement results manifested a ~74% reduction in energy consumption when transferring from the valve-based LS system to DHMPA system.

The measurements were conducted with the second prototype of DHMPA and with original system architecture of the excavator, not optimized for the new control method. Therefore even higher system efficiencies are expected to be reached when the system components, mainly hoses and pipes, are dimensioned appropriately. In addition, if the charging process of the accumulator is also optimized to utilize the regions of highest efficiency of electric motor and pump, further enhancements in energy efficiency are expected. These remain to be studied in further research.

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