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Energy Balance of Electro-Hydraulic Powertrain in a Micro Excavator

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***Abstract*—This paper presents the experimental results of the study performed with an electrified small sized excavator, a 1.1-tonne JCB Micro, equipped with conventional hydraulics. The highlighted points in this study are the overall energy balance of the electro-hydraulic powertrain of this excavator and the power losses in individual components. The measured energy balance of the electric motor powered system is compared with the simulation data obtained from a preliminary simulation model of the system. The empirical evidence and the results of the preliminary simulation model will be in future research utilized to discover and compare new alternatives for powertrain architectures.**

Keywords*—*excavator hydraulics; energy efficiency; simulation

I. INTRODUCTION

The construction machinery, alike all other machinery whether stationary or mobile, is today faced with ever tightening demands for higher energy efficiency. To meet these demands, new energy-saving technologies have been developed for powertrain optimization over the last decades. In addition, variety of different configurations and control strategies to modify conventional excavator to be hybrid or even hybrid plug-in have been proposed all around the globe. The purpose of this study is experimentally to chart the overall energy balance baseline and the power losses of individual components in the powertrain of an electrified small sized working machine, a 1.1-tonne JCB Micro excavator. This empirical baseline evidence will be used as a reference when developing new, more energy efficient powertrain solutions in the future. In future studies this data will also be used for finding out what effect the electrification of the excavator has had on the energy consumption compared to the original diesel engine powered configuration.

The studied excavator has originally been diesel engine powered producing 13.6 kW at 2200 rpm, but it has been modified to electric motor powered [1–3]. The excavator has also been equipped with sensors throughout the electro-hydraulic powertrain in such a way that the overall energy balance and the power consumption of individual components and work movements can be estimated in detail. The battery pack, electric motor and hydraulic pump can be measured and the power losses in control valves and transmission lines can be estimated on basis of the data gathered from the sensors.

Besides making measurements and analyzing the acquired data, a preliminary simulation model for the excavator including mechanics has been used in this study. The applicability and reliability of this model is enhanced based on the recently acquired measurement data. In the future stages of the study, the model will be further developed and used for comparing different alternatives for powertrain architectures as well as for optimizing the system components in regards of type and size. Since the next experimental step in the development of the excavator's powertrain architecture is to transfer from central hydraulics to direct drive hydraulics, where each of the machine functions will be individually powered, also this architecture will be investigated using the created model.

The remainder of this paper is organized as follows. Scheme and the working principles of the systems are described in detail in Section II. Section III describes experimental procedure, and section IV describes the simulation model. Analysis of measurement results is described in Section V. Discussion and Concluding remarks are presented in Sections VI and VII, respectively.

II. SCHEME AND WORKING PRINCIPLES

The studied 1.1-tonne JCB excavator utilizes low cost load-sensing (LS) system comprised of a fixed displacement fixed speed pump and a pressure adjustment valve. The valve senses the highest load pressure and adjusts the system pressure accordingly by directing the excess pump flow to the tank.

In the following text, the term conventional hydraulic system refers to the current hydraulic setup of the excavator, which is powered by an electric motor and controlled with electro-hydraulic proportional directional control valves. The original diesel engine powered system was otherwise similar, but the proportional directional control valves were manually controlled. For description of the excavator modification, refer to [1–3]. The current conventional hydraulic system of the studied excavator is presented in Figure 1. In the system, the reference rotational velocity for the electric motor (3) is set by a Sevcon Gen 4 motor controller (2). The motor runs a gear type fixed displacement Parker PGP511 dual pump (4) that produces a flow rate that depends on the rotational velocity of the prime mover and the displacements of the pumps, which in this system are both 6 cm^3/rev. Proportional directional control valves (8) control the three actuators that run the functions of Bucket, Arm, and Boom. Pressure adjustment valve (6) senses through shuttle valves the highest prevailing load pressure of the actuators and adjusts the system pressure to a level that is approximately 20 bar higher than the highest load pressure. Simultaneously it also directs back to tank the portion of the fixed pump flow that is not needed in the actuators. The pressure relief valves (5, 7) fulfil a safety function preventing the pressure from rising to a level that would damage the system. During normal operation of the system, these valves should be in closed position.

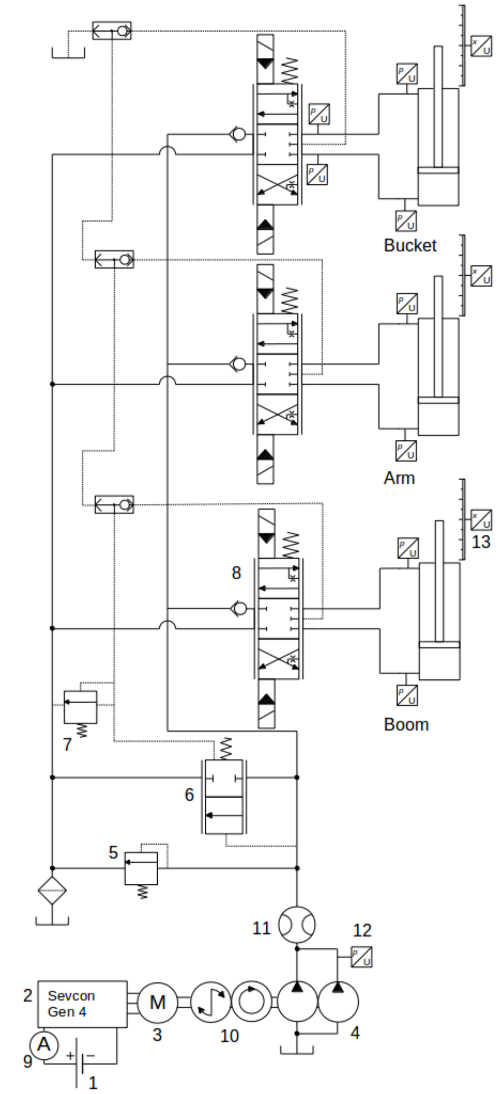

Fig. 1 Schematic of the excavator's hydraulic system with sensors

The load sensing circuit in each proportional valve (8) reads the load pressure only when the spool is moved from its center position and furthermore only from the port directing the flow to the actuator, which is to be moved (Boom, Arm or Bucket). This means that lowering a heavy load by throttling will not raise the pressure level of the entire system. Simultaneous movements of the actuators with high load pressures will therefore not always lead to high power consumption.

The system is equipped with several pressure sensors in the lines between actuators and their controlling valves and also in the pump outlet, which is also equipped with flow rate sensor. The actuators are additionally equipped with position sensors. The power input axle between electric motor and dual pump is equipped with torque and rotational velocity sensors. Table I lists the components of the hydraulic system as well as the sensors applied to it.

III. Experimental Procedure

In order to be able to compare the measured performance of the studied excavator with other machines of similar type and size, a generally accepted test procedure was needed. For this, the procedure and duty cycle defined in the standard H 020:2007 of the Japan Construction Mechanization Association (JCMAS) [4], originally meant for testing the fuel consumption of hydraulic excavators, was applied. The measurements are conducted without external loading as defined in the standard.

TABLE I. Components and Data Acquisition System

Number	Description	Details
1	Battery pack	72V Lead-acid
2	Motor controller	Sevcon Gen4
3	Electric motor	10 kW
4	Dual pump	Parker PGP511
5	Pressure relief valve (PRV)	
6	Pressure adjustment valve	
7	PRV LS-circuit	
8	Proportional valve	Danfoss PVG-32
9	Current sensor	LEM DK 200
10	Torque and tachometer	Kistler 4502
11	Flow sensor	Kracht VC 0,4
12	Pressure sensor	Hydac HDA
13	Position sensor	Siko SGH10

The swing and bucket motions of the excavator are not included in the scope of this work; therefore, only boom and arm movements were studied. Figure 2 visualizes the duty cycles of the arm and bucket actuators. The duration times of these cycles were 10 s each.

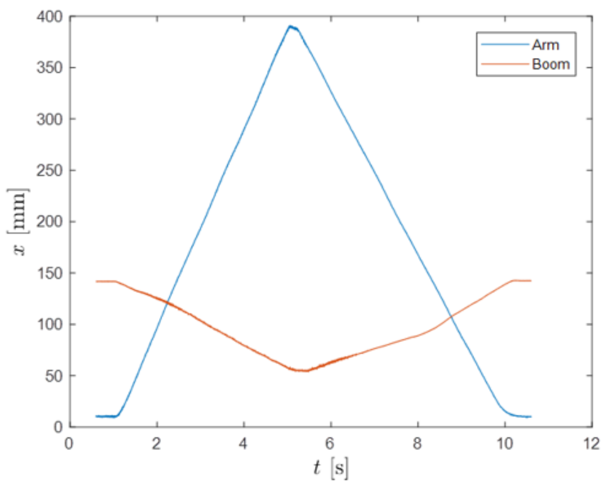

Fig. 2 Levelling cycle according to JCMAS H020:2007 when only boom and arm movements are taken into account.

In realization of this levelling cycle, position feedback controllers were used to control the proportional valves via CAN bus to reach the reference position of each actuator. The CAN communication with the valves was implemented with Simulink and furthermore the controller was ran in Simulink. The controller itself was a proportional loop, which was tuned for the levelling cycle ensuring proper movement of each actuator.

IV. Simulation model

To solve the total energy consumption of the excavator and the power losses of its individual components, a model interlinking the subsystems of mechanics, hydraulics, electrics and control was needed. This was created in Matlab/Simulink. The dynamics of the multibody structure was modeled in PTC Creo and imported to Matlab through Simscape Multibody Link Plug-In, [5]. The top level structure of the excavator model is presented in Fig. 3.

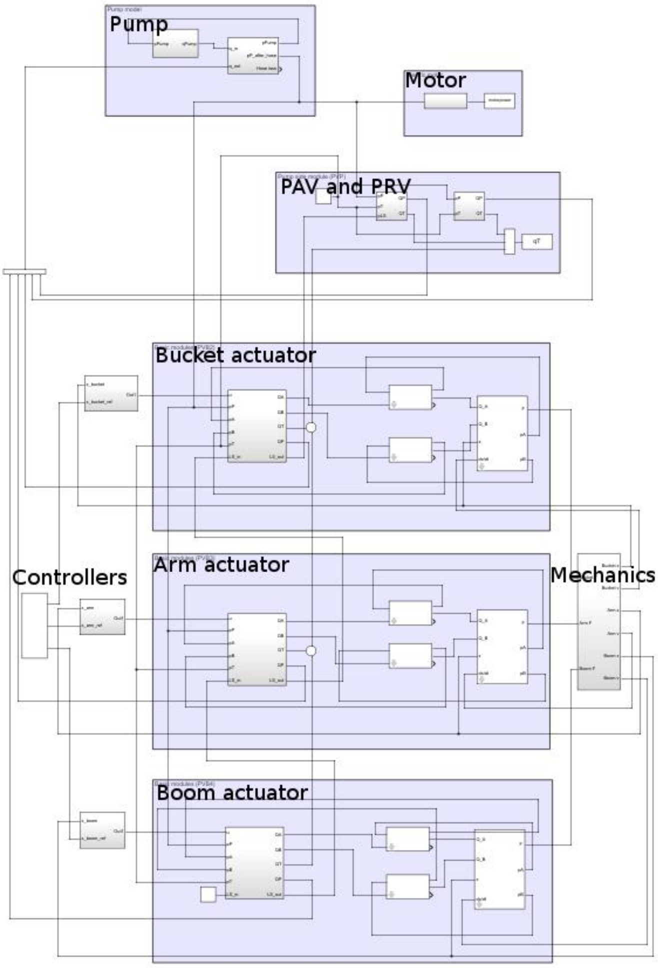

Fig. 3 Top level structure of the Simulink based simulation model of the studied excavator visualizing the interconnections between the subsystems.

The description of the mathematical models of the hydraulic components built in Matlab/Simulink and used in simulation of the excavator is presented in [6]. However, in the present study some of these models were developed further or their parameters were updated on basis of the gained measurement results. This was done to get the simulation results to comply more accurately with the measurement results.

The pump model used in simulations assumes the pump leakage to be solely dependent on the pressure difference between the flow ports of the component, and the effects of the rotational velocity of the pump and the fluid temperature are omitted. The leakage parameters of the pump model were tuned in order to give better match between the simulated and the measured effective output flow of the pump. The cylinder friction was modeled with LuGre model [6], however, the pressure difference between the cylinder chambers was not taken into account and nor were the fluid temperatures. The parameter values for the cylinder model were obtained from measurements. The model of proportional valve block presented in [6] was outfitted with leakage flow in the present study. The pressure adjustment valve was modeled as linear relation between the pressure difference over the valve and the flow rate through it. The pipe model includes the pipe friction that causes pressure loss in the system as well as the effective volume of the pipe that brings elasticity to the system.

V. Results and analysis

A. Simulation results validation

This section presents simulation results and their validation with performed measurements. Figs. 4–5 and 6–7 illustrate the position, and cylinder force for boom and arm for the 10 s long cycles, respectively. The command(s) for actuator positions are realized in such a way (oversized) to ensure that at least one of the proportional valves is fully open for (practically) most of the time.

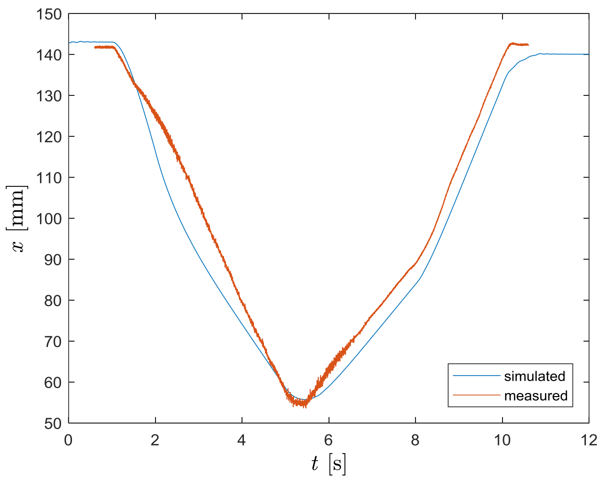

Fig. 4 Simulation and measurement results for boom position.

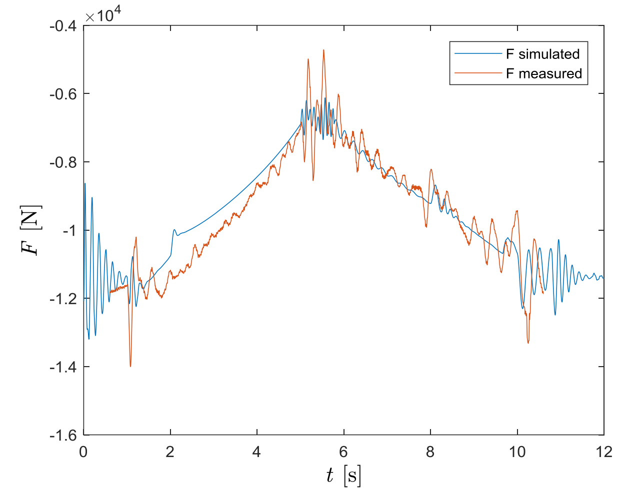

Fig. 5 Simulation and measurement results for boom force.

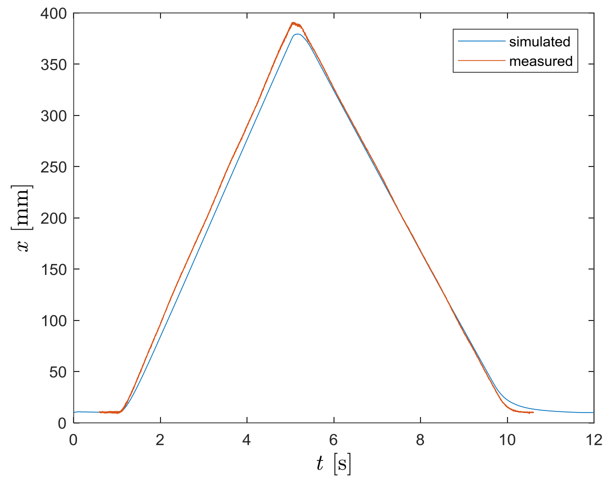

Fig. 6 Simulation and measurement results for arm position.

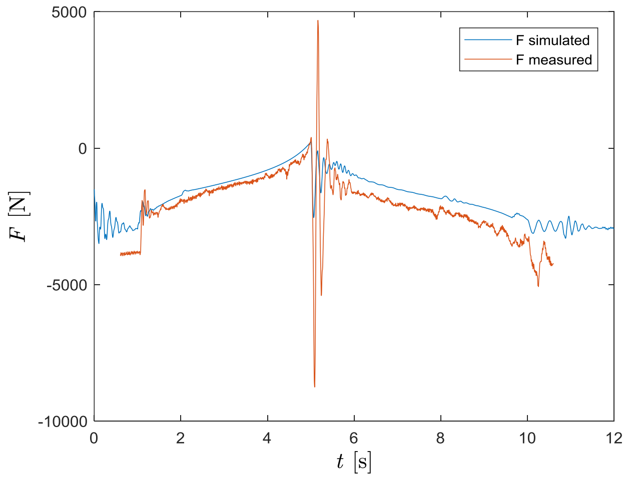

Fig. 7 Simulation and measurement results for arm force.

According to Figs. 4–7, validation of the model showed acceptable results for use in preliminary performance analysis.

B. *Measurement data analysis*

The system was equipped comprehensively with sensors that provided information on power usage in the electric, mechanical and hydraulic subsystems, including actuator hydraulics.

The electric power input from the battery to the electric motor was determined from battery voltage and current measurements. The mechanical power, which equaled the output power of the electric system and the input power of the hydraulic pump and thus the whole hydraulic system, was obtained from electric motor's torque and rotational velocity.

The hydraulic power is available from hydraulic pressure and flow rate measurements on the combined outlets of the hydraulic pumps (point 12 in Fig. 1). The cylinder chambers' (piston and rod) pressures and the piston position measurements give an estimate for the hydraulic power of each cylinder. The piston velocity was approximated by filtering the position signal and differentiating it numerically. The cylinders' flow rate estimates were calculated based on those calculated piston velocities. In the simulation, actuator piping friction losses were estimated by using pressure readings at opposite ends of pipe systems.

Overall, the power usage and power losses in the sub-systems could be estimated, making it possible to compare the contribution of the different subsystems to the total energy consumption.

C. *Power and energy consumption*

Fig. 8 illustrates the measured electric input and output power, pumps' output power as well as the actuators' power use during the 10 s long levelling cycle.

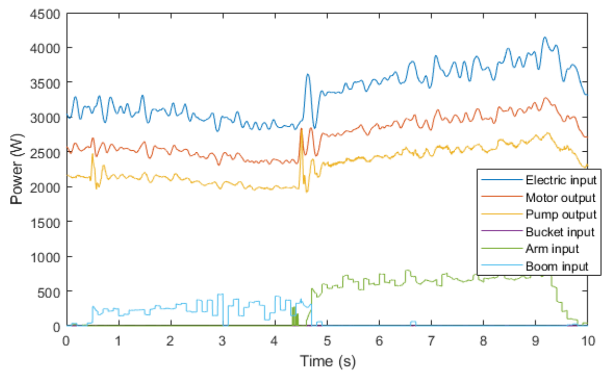

Fig. 8 Power usage during the levelling cycle.

Fig. 9 illustrates the levelling cycle's energy consumption in various sub-systems of the entire power transmission system as a function of time. For comparison and model validation purposes, the values computed from the measurements are presented together with the simulated estimates.

In Fig. 9 the electric energy is the energy drawn from the battery pack, i.e. the total energy consumption of the system, the mechanical energy is the energy needed to run the hydraulic pumps, and the hydraulic energy is the energy output from the pumps fed to the hydraulic actuator system. Since the levelling cycle is a zero energy process, all the hydraulic energy is wasted in the hydraulic system when the JCMAS non-load levelling cycle is run.

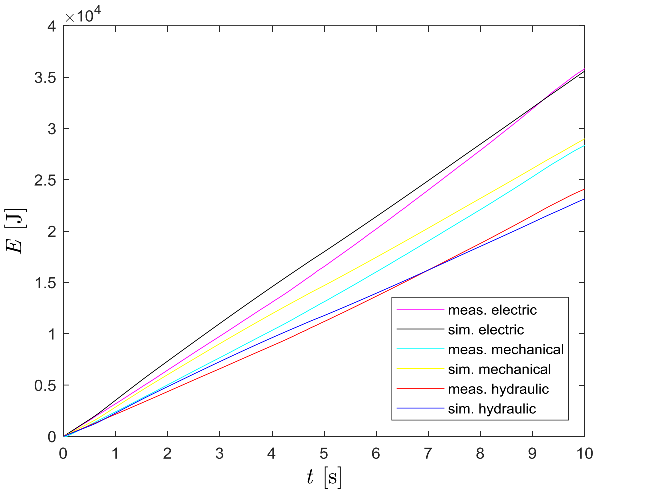

Fig. 9 Measured and simulated energy consumption of sub-systems for levelling cycle.

The energy consumptions presented in Fig. 9 can be converted to give the relative energy consumptions of the sub-systems as shown in Fig. 10.

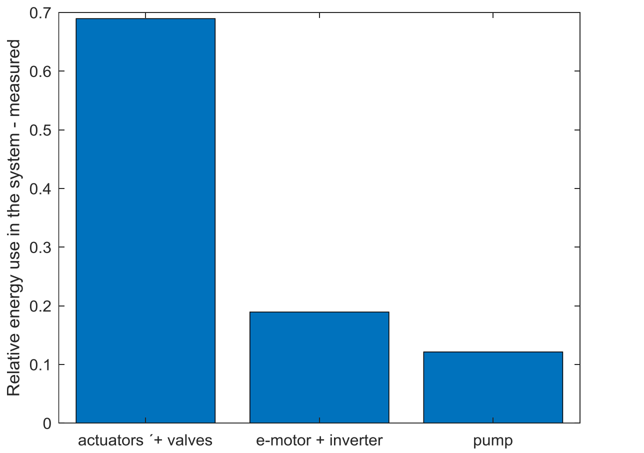

Fig. 10 Measured relative energy use in different subsystems of the power transmission.

Electric motor’s average energy efficiency corresponds to 79% and pump’s average efficiency to 85%. This means that on average approximately 67% of the energy provided by the battery pack is left for the actuator system. This energy can be further divided to component specific energy consumptions to reveal the points, which cause the most of the system losses.

Fig. 11 presents the simulated estimates for relative energy losses in the actuator system divided to cylinder seals’ friction, pressure losses in hoses, pressure losses in pressure adjustment valve and pressure losses in proportional directional control valves.

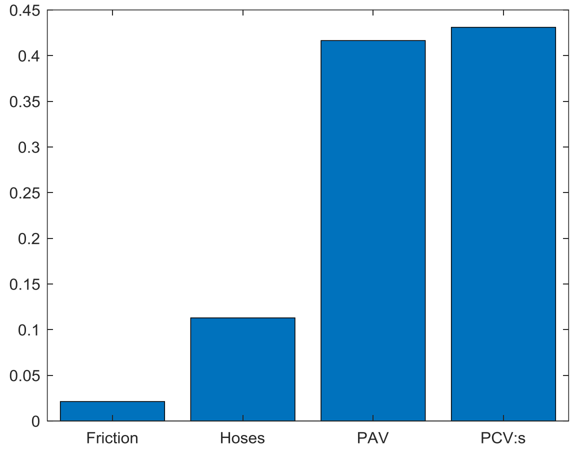

Fig. 11 Simulated relative energy use in the actuator system for levelling cycle.

The dominant position of the valves in the energy losses of the actuator system is explicit, their portion is over 80%.

Fig. 12 illustrates the flow rate provided by the two pumps and the sum of flow rates utilized in the cylinders.

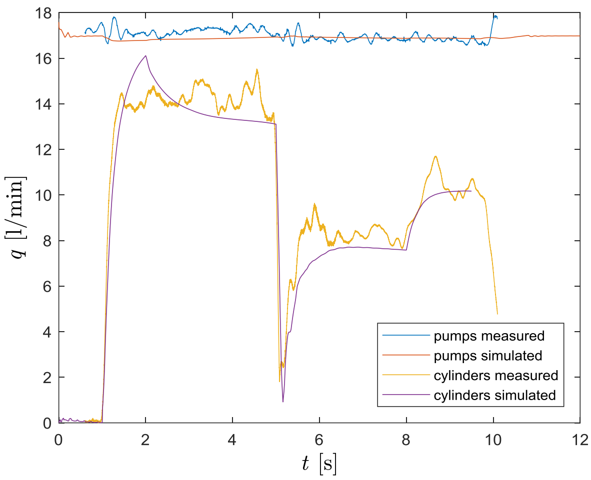

Fig. 12 Measured flow rates of pumps and cylinders for levelling cycle.

In the first section (1–5 s) of the levelling cycle, the flow rate produced by the twin pump is closely matched to the flow need of the actuating cylinders. Hence, only a minor flow rate is directed back to the tank through the pressure adjustment valve or the pressure relief valve, and therefore the energy losses during this section are lower than in the second section, where the difference between produced and needed flows is greater.

VI. DISCUSSION

In this research, a levelling cycle inspired by the JCMAS standard [4] was utilized in analyzing the energy balance of an electric motor powered small sized excavator equipped with conventional hydraulics. Measurement data pertaining to the energy balance was compared with computer simulation data obtained with a Matlab/Simulink model of the system.

The system with interconnected mechanical and hydraulic systems is relatively complex for simulation. However, the attributes related to power use can be modeled with precision which is adequate for designing of power transmission implementations and for assessment of performance of different options.

In Fig. 10 it can be seen that approximately 70% of the energy is consumed in the valves and the actuators. Thanks to the simulation, see Fig. 11, it becomes clear that the proportional directional control valves and the pressure adjustment valve are responsible for most of the energy consumption within the hydraulic system.

The easiest way to reduce the energy consumption in the hydraulic system would be to re-dimension the pipe system (now the simulations showed that the flow velocity could exceed 6 m/s). An increase in the diameter of the pipelines (mainly the hoses) would reduce the flow velocities and thereby significantly reduce the pressure losses.

To obtain major improvements in energy efficiency, larger modifications would be required. The electric motor's speed control strategy should be changed from constant velocity to control where the rotational speed of the motor and thus the pump flow rate correspond to the real flow rate need of the actuators. The control signal could be based on a flow rate estimate which would be calculated by the operator's joystick command signals. Also a realistic fixed value for the pressure difference value in the pressure adjustment valve or online measurements of supply pressure and actuator load pressures would be needed for the flow rate estimate and corresponding rotational speed of electric motor. In the studied system the response of the electric motor control is probably fast enough to respond to the needs in flow rate changes in excavator use.

One well known feature of LS systems is that the pump pressure is determined by the highest load pressure in the actuator system. The pressure provided by the pump is often optimal for one actuator but potentially far from favorable for other actuators. Separate pumps for each actuator could be a functionally acceptable, but a costly remedy for this problem. The use of proportional control valves causes significant losses and applying separate control edge control could diminish these losses.

The levelling cycle used in the tests is basically a zero energy work cycle since the start and stop points are the same. This test includes phases, where moving of the masses requires actuator work but also periods where there would be a possibility to regenerate the changes in the kinetic and potential energy. In the studied system most of this is dissipated in the orifices of the proportional control valves. For enhanced system efficiency an architecture with the possibility for a hydraulic or electric energy recovery could be considered. This would require major changes in the system structure.

The friction in the actuators and other moving parts in the system has only minor effect on the energy balance according to the analysis. The considerable power losses in the hydraulic transmission lines could easily be reduced by reasonable dimensioning of the hoses and fittings.

VII. CONCLUSIONS

This study was carried out by experiments on overall energy balance of electro-hydraulic powertrain and the power losses in individual components in electrified 1.1-tonne JCB Micro excavator. The results indicated that significant source of losses in the studied case are the proportional valves as well as the pressure adjustment valve of the low cost version of LS system. Modification in which the pump flow rate could adapt to the actual flow rate need of the actuators would enhance the energy efficiency remarkably. More radical improvements in energy balance would require for instance changing the proportional valves to independent metering. The actuator flow rates could also be controlled directly with actuator dedicated pumps. Also the possibility for energy recovery should be considered in system architecture selection.

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