Efficiency Study of an Electro-Hydraulic Excavator

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A Matlab/Simulation model is utilized to study the total energy consumption and power distribution of the micro excavator. The model consists of the hydraulic and mechanical systems related to actuation of front hoe, i.e. boom, arm, and bucket. The excavator is equipped with pressure and position sensors, from which the measurement data is collected for parameterization and verification of the simulation model. Two different duty cycles, named *digging and loading cycle* and *leveling cycle*, based on the JCMAS standard, are utilized in this study. The results indicate a formidable room for improvement, concerning the hydraulic system, since a power loss of as much as 60% is localized in the directional valve group.

Keywords: Excavator, efficiency, hydraulics, losses
Target audience: Mobile Hydraulics

1 Introduction

Off-highway working machines cover a wide range of applications, including agriculture, earth-moving, and mining machinery. These machines are often utilized in challenging conditions, and their duty cycles consist of quick and high power peaks. This makes off-highway working machines a demanding target for research and development. Their requirement for high maximum power, along with full mobility is a reason why these machines are predominantly powered by a diesel engine. Due to the fluctuating fuel price and tightening regulation for emissions, there is a constant need to improve the efficiency and reduce the engine size and emissions of the machines. Numerous studies have been published about improving the efficiency of off-road machinery.

Excavators are ordinarily equipped with conventional, centralized hydraulic system, which consists of one or two main pumps that supply volumetric flow for the whole system. This flow is directed from the pump to actuators through control valves, and the returning flow is directed into the tank. Conventional hydraulic system has numerous disadvantages. Many supportive functions are required, including pressure control and load-sensing functions. The power demand of the system changes, which prevents the engine from running at its optimal speed. Even in idle mode, flow losses are present due to continuous circulation of fluid through valves. In addition, the distance between pumps and actuators may be long, which causes pressure loss and an additional weight of long hoses filled with fluid.

Load-sensing (LS) system is common an example of the conventional hydraulics. In a load-sensing (LS) system, a load-sensing circuit monitors the load pressures on all actuators, and adjusts the system pressure to match the highest load. If several actuators operate at the same time, which often is the case, the excess pressure is decreased by throttling. According to Zimmerman *et al.* /1/, these throttle losses may be responsible for as much as 35% of total energy losses during a typical digging cycle. Knowing the energy distribution of the machine is vital in order to steer the research towards the most relevant targets.

This study concentrates on improving the energy efficiency and performance of a JCB micro excavator. In the beginning of project, the 14 kW diesel engine of the excavator was replaced with a 10 kW electric motor and a lithium-titanate battery. This electric system showed performance of same level as the previous diesel powered version. In addition, the noise level decreased, and the emissions of diesel engine were eliminated, both of which are influential factors, especially when working indoors. However, due to low battery capacity and long charging time, the operation time was dramatically lower than of a diesel powered version. /2/. The excavator was further improved with a start-stop system that was able to save energy as much as 32%, but demand for further improvements still exists /3/. The original hydraulic system of the excavator controlled by manually operated directional valves was replaced with the electro-hydraulic proportional valves, which were installed parallel to the original valves /4/. The great amount of power is wasted in throttling losses in the hydraulic circuit, and novel solutions for the issue have been developed. Direct-driven or valveless hydraulics, in particular, have demonstrated encouraging potential to eliminate a large share of hydraulic losses in off-highway mobile machinery. /5/, /6/. Therefore, in this project, the efficiency of the hydraulic system of the excavator is under detailed investigation. Boom, arm and bucket actuation are included in the scope of this work.

The objective of this study is to resolve the actual energy consumption and power distribution of the front hoe of the micro excavator (Figure 1), including boom, arm and bucket actuators. This leads to creation of a simulation model in Matlab/Simulink environment of excavator and experimental validation of model. Excavator is fitted with pressure and position sensors, and the simulation model was verified with laboratory measurements. The verified model was utilized to calculate the power consumption of the excavator during a digging and loading and a levelling cycle.

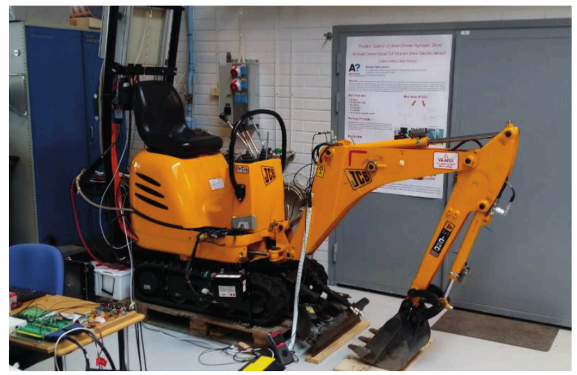

Figure 1: JCB Micro excavator

This structure of this paper is as follows. The excavator and the simulation model is presented in chapter 2. In chapter 3, the simulation model is utilized to study the power consumption of the excavator during two standardized working cycles. The final conclusions, together with suggestions concerning the upcoming research, are presented in chapter 4.

2. Excavator test case

JCB excavator is one of study cases in EL-Zon project. JCB excavator utilizes LS system. It is noteworthy, that the term 'LS system' is commonly used to describe a system with a variable-displacement pump. The hydraulic system of this excavator, however, has a fixed-displacement pump with constant rotational speed. It senses the load pressure, and adjusts the system pressure accordingly, by directing the excess volume into the tank, via the pressure adjustment valve.

For clarity in this paper, the conventional hydraulic system refers to the current setup, which is powered by an electric motor, and controlled with electrical valves (for modification in excavator refer to /2-4/. In contrast, the factory-made system, with diesel engine, and manually controlled directional valves, is referred to as original system. The current hydraulic system of the excavator is illustrated in Figure 2.

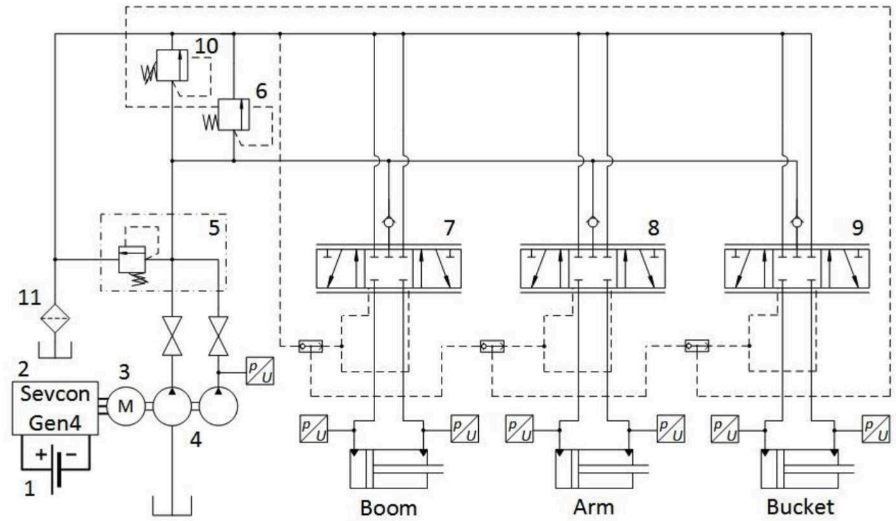

Figure 2: Simplified hydraulic schematic of the hydraulic system of the excavator

Number	Component	Details
1	Battery pack	
2	Motor controller	
3	Electric motor	
4	Pump	Parker PGP511 gear pumps, with a fixed volume of 2 x 6 CC /7/
5	Junction block with PRV	
6	Pressure adjustment valve	
7-9	Directional valves	
10	Pressure relief valve	
11	Return oil filter	

Table 1: Main components of the hydraulic system

Excavator is fitted with additional sensors and data acquisition system. As Figure 2 demonstrates the excavator is fitted with pressure sensors in all cylinder ports and in the pump outlet port, and position sensors at the cylinder rods. The measurement data, combined with dependable, measure-based dynamic model, gives access to otherwise hard-to-find parameters, such as cylinder friction forces and volumetric losses of the system. The measurement signals are collected and recorded at a target-pc, which is operated by Matlab/Simulink Real-time. A simple position feedback controller is established to control the front hoe.

2 Simulation model

A simulation model of micro excavator is created in Matlab/Simulink environment. Developed model is utilized to study the energy consumption of the micro excavator. This work is focused in the hydro-mechanical system of the excavator front hoe, and excluding powertrain. The developed hydraulic model contains the hydraulic pump, directional valve group, auxiliary valves, hydraulic cylinders, the mechanical model of the front hoe, and the connecting hoses. The electric motor and the motor controller are assumed to be ideal, with 100% efficiency and constant rotational speed. In developed multibody model of excavator is created with Simscape Multibody toolbox. Following sub-sections will introduce in short realizations for the modelled components. For detail explanation please refer to /16 /.

2.1 Hydraulic pump model

Two hydraulic pumps in the excavator are Parker PGP511 gear pumps, with a fixed volume of 2 x 6 CC. According to manufacturer data /7/, the PGP511 pump has a 12-tooth gear profile and optimized flow metering to provide reduced pulsation and quiet operation. Therefore, the output flow is assumed “flat” and modelled with a simple lookup table, producing the flow according to the curve provided by manufacturer /7/.

2.2 Hose model

The pressure drops of the hoses are estimated by mathematical formulas. A hose model consists of the pipe friction, orifice, and volume sub-models. In the transition phase between laminar and turbulent flow, the friction factor is modelled with a simple continuous function, which is also continuously differentiable. The accordant friction factor is selected based on the Reynold’s number. In addition to pipe friction, there are flow losses in the system, which are related to change of speed or direction of the flow. These losses are present in joints and bends, for example /8/. Values listed in Kauranne *et al.* /8/ have utilized for friction factor ζ.

2.3 Proportional valve model

Danfoss PVG 32 proportional valve group consists of three main module types: pump side module (PVP), basic modules (PVB), and actuation modules (PVED-CC). The PVP connects to the pump and tank ports, and it has different functions depending on the application. In this valve group, the PVP is an open-centre version, which is to be used with fixed displacement pumps. The manufacturer part number is 157B5110, and the operation is explained in detail in /10/.

The pressure adjustment spool is modelled with a lookup table, result of which corresponds the graph given by the manufacturer, with some adjustments made to make the simulation model match the measured data. A rate limit block is added to limit the transition speed of the spool, and the flow is saturated to minimum of 0 l/min and maximum of 140 l/min. The pressure relief valve is modelled based on the manufacturer data. The nominal set point for the valve block PRV is 18 MPa. However, the system pressure is, based on the measurements, limited to 12-13 MPa. Thus, the pressure limit of the valve block is never reached, and only the junction block PRV activates when the pressure rises up to the limit. The PVB module has a manufacturer part number 157B6100 for the PVB module and 7005 for the spool. The model is controlled with a spool position command u, and it outputs the flow for each valve port (P, A, B, T). It also compares the load pressure of the active port against the loads on other spools, and passes forward the highest pressure. The spool dynamics, consisting of a transfer function and a saturation block, based on the work of Bak & Hansen, which studied the dynamic behaviour of a PVG 32 valves /11/. It must be noted, that this valve has different spool size and different components, so the transfer function parameters serve only as an estimation of the actual spool dynamics. The spool position is converted into the relative openings of the control edges of the spool, which are modelled with four lookup tables, based on the measurement data. Oil flow through the ports was calculated with four separate orifice blocks. Orifice model was utilized to calculate a volumetric flow, caused by a pressure difference over a flow path, and it takes the flow type (laminar, turbulent or transient) into consideration. Ellman & Piche equation for a polynomial laminar flow formula is utilized for the pressure difference is below the transition pressure /12/.

The piecewise equation for volumetric flow q [m^3/s] can be written as:

$$\boldsymbol{q(P)} = \begin{cases} \boldsymbol{K_V\, sgn(p_1 - p_2)\sqrt{|p_1 - p_2|}} & \boldsymbol{(|p_1 - p_2| > p_{tr})} \\ \boldsymbol{\dfrac{K_V(p_1 - p_2)}{2\sqrt{p_{tr}}}\left(3 - \dfrac{|p_1 - p_2|}{p_{tr}}\right)} & \boldsymbol{(|p_1 - p_2| \le p_{tr})} \end{cases} \qquad (1)$$

where p_1 and p_2 are the pressures before and after the orifice [Pa], p_{tr} is the transition pressure [Pa]. The flow coefficient K_v is calculated for each valve port from equation:

$$K_V = \frac{q_{nom}}{\sqrt{p_{nom}}}, \tag{2}$$

where q_{nom} is the nominal flow [m³/s], and p_{nom} is the nominal pressure differential [Pa]. The values are determined for each spool separately, based on the measurement data. Used values are collected in Table 2 at the pressure difference of 1 MPa.

Spool	Nominal flow (P-AB) [l/min]	Nominal flow (AB-T) [l/min]
Boom	5.6	6.1
Arm	5.3	6.2
Bucket	5.4	6.1

Table 2: Nominal flow rates of the control spools

2.4 Cylinder model

The cylinder model transforms the introduced volumetric flow into chamber pressures and output force. Output force is a product of chamber pressure and the piston area on that side. Cylinder friction consists of forces between surfaces of the cylinder and the seals, which includes seals of piston and piston rod. Jarf /13/ exploited a dynamic friction model, proposed by Canudas de Wit *et al.* /14/. The utilized model takes into account most of the dynamical friction behaviour, including Stribeck effect, hysteresis, stick, and varying break-away force. In developed model, also known as the LuGre model, the contact is thought as bristles moving against each other. When applied to the cylinder friction, the bristles can be seen as representatives of cylinder seals. The average deflection of the bristles is marked z and its time derivative is modelled by:

$$\frac{dz}{dt} = v - \frac{\sigma_0|v|}{g(v)}z \ , \tag{3}$$

where

$$g(v) = F_c + (F_s - F_c)e^{-\left(\frac{v}{v_s}\right)^2}, \tag{4}$$

$$F_\mu = \sigma_0 z + \sigma_1 \dot{z} + \sigma_2 v \ , \tag{5}$$

where F_μ is the friction force [N], z is the average deflection of bristles [m], and v is the relative velocity between sliding surfaces [m/s]. The friction parameters, demonstrated in Table 3, are based on the literature, and adjusted according to the performed measurements.

Notation	Explanation	Boom cylinder	Arm cylinder	Bucket cylinder	Unit
v_s	Stribeck velocity	0.001	0.001	0.001	m/s
F_c	Coulomb friction	200	200	200	N
F_s	static friction	800	400	400	N
σ_0	stiffness of bristles	$1.6 \cdot 10^6$	$1.6 \cdot 10^6$	$1.6 \cdot 10^6$	N/m
σ_1	damping coefficient	$5 \cdot 10^3$	$5 \cdot 10^3$	$5 \cdot 10^3$	Ns/m
σ_2	viscous friction coefficient	$5 \cdot 10^3$	$5 \cdot 10^3$	$2 \cdot 10^3$	Ns/m

Table 3: Friction coefficients of the cylinders

2.5 Simscape Multibody model

In order to simulate the physical response of the mechanical system, a multibody model is created with Simscape Multibody toolbox. Table 4 shows the utilized weights of the mechanical system of the micro excavator. In this work, it is assumed that all joints are ideal, and provide only certain degree of freedom. No friction acts on the joints accept friction in cylinder model. Bodies are considered to rigid, no deflections or vibrations applied. In addition further simplification have been implemented. The attachment point of the front hoe, the kingpost, is modeled with welded connection to the world frame, whereas the actual excavator provides some degree of movement.

Component	Weight of component [kg]
Boom	59.50
Arm	28.00
Bucket	30.00
Linkages	10.00
Boom cylinder	16.67
Arm cylinder	11.57
Bucket cylinder	9.41
Hoses	1.70
Pins 1, 3, 4, 6	1.08
Pin 2	0.63
Pins 5, 7	0.66
Pins 8, 10	0.83
Pins 9, 11	0.74

Table 4: Weights of the components of the front hoe

2.6 Model verification

The simulated cylinder pressures verified against the measured ones. In order to acquire the necessary measurement data, the excavator is controlled to move the arm cylinder from fully retracted position to fully extracted position and back on full speed. The valve flow signal (AVEF) is recorded and utilized as an input for the spool position in the simulation model. Measured cylinder position, pressures in chambers A and B, and the system pressure at pump outlet port are plotted together with simulated values, and are shown in Figure 3.

Neglectable fluctuation is visible in transition states, namely at 17.5 s, is due to the properties of the electric motor and the controller. In the simulation model, the motor speed is assumed constant, which results in different pressure curves. However, after the transition phase, the pressures are settling on the correct level. The chamber pressures correlate with the cylinder movement appropriately. Despite a minor difference in the trajectories on the position curve, the simulated average speed of the actuator corresponds to the measured speed.

Other actuators (boom and bucket) were driven in predetermined positions, to ensure corresponding inertial properties, but were not subject to control, to prevent any unwanted disturbance for the system pressure. The bucket and boom actuations are evaluated in a similar manner. The simulated system pressure matches the measured values, the chamber pressures are on adequate level, and the cylinder position curves are nearly identical.

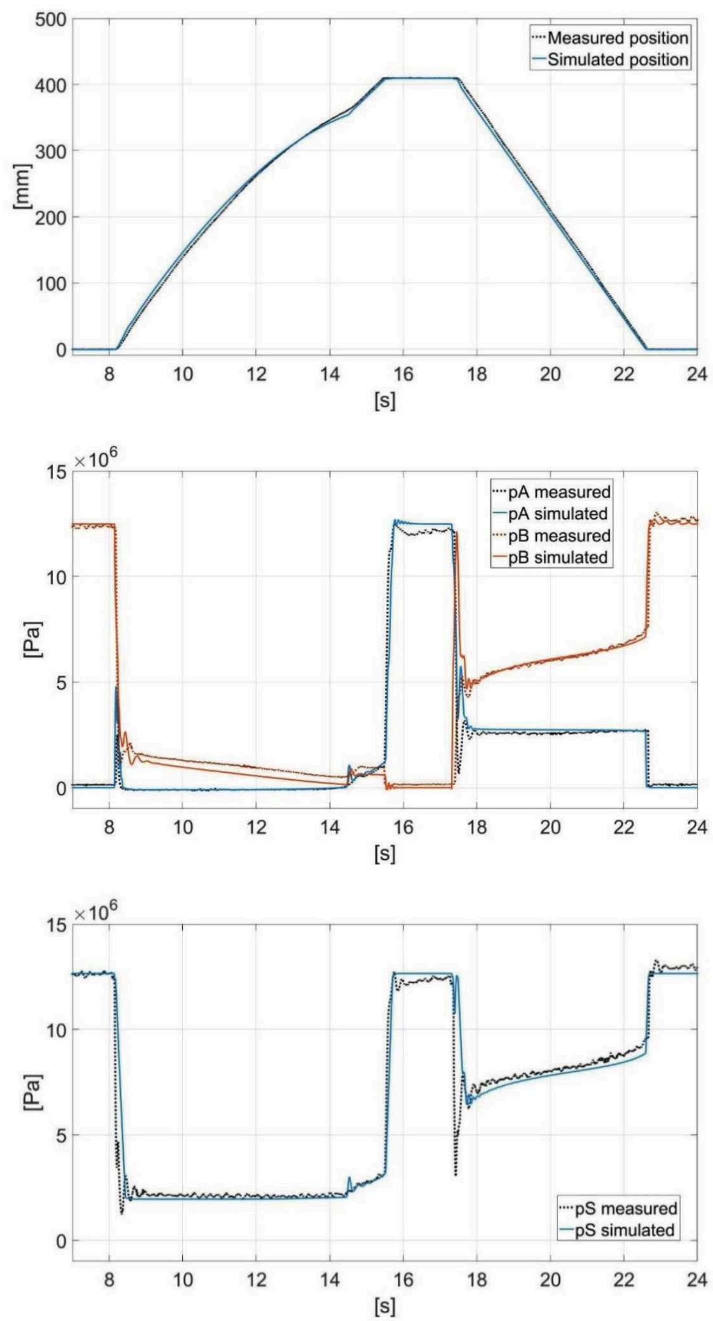

Figure 3: top: Measured and simulated arm position; middle: Measured and simulated arm cylinder chamber pressures; bottom: Measured and simulated system pressures during arm movement

According to the verification tests, all three actuators exhibit a realistic behaviour. The simulated system pressure, during the boom movement, fluctuates more than the measured pressure. This is due to minor difference between the dynamics of simulated and actual pressure adjustment valve. Pressures settle in correct levels and the simulation converts the valve opening signals into precise actuator positions. Therefore, the simulation model is now verified with simple single-actuator manoeuvres.

3 EFFICIENCY ANALYSIS

In this section, the verified simulation model is utilized to study the total energy consumption and power distribution of the micro excavator, and the results are compared and discussed.

In order to evaluate the performance or efficiency of an excavator, it is necessary to determine the actual use of such a machine. Digging movement is common to all duty cycles found in the literature. However, the varying loading conditions make standardizing a difficult task, as the excavators are used in very different conditions. The duty cycles used in the study, are based on The Japan Construction Machinization Association (JCMAS) standard for testing the fuel consumption of hydraulic excavators /15/. Cycles are scaled down, due to the physical limitations of the test excavator. In this study, the loading height is 1.2 m and digging depth 0.75 m. Since the swing motion is not included in the scope of this work, only boom, arm and bucket movements are performed. In the levelling cycle, the processing length is 1.7 m. The reference cylinder lengths for each actuator, compared with visualization of the cycle, are illustrated in Figure 4 for the digging and loading cycle.

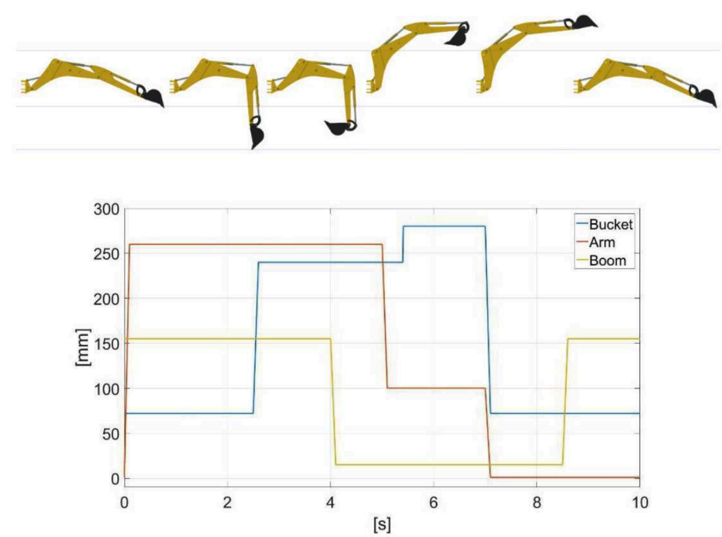

Figure 4: The digging and loading cycle

A similar presentation is displayed in Figure 5 for the levelling cycle.

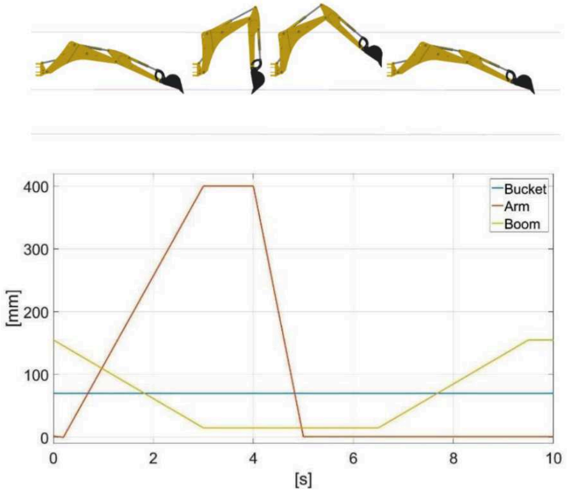

Figure 5: The levelling cycle

First, digging and loading cycle is performed and illustrated in Figure 6. Figure 6 illustrates the power division in seven categories: pump input and output power, pressure adjustment valve (PAV), pressure relief valve (PRV), directional valves, cylinders input and cylinders output.

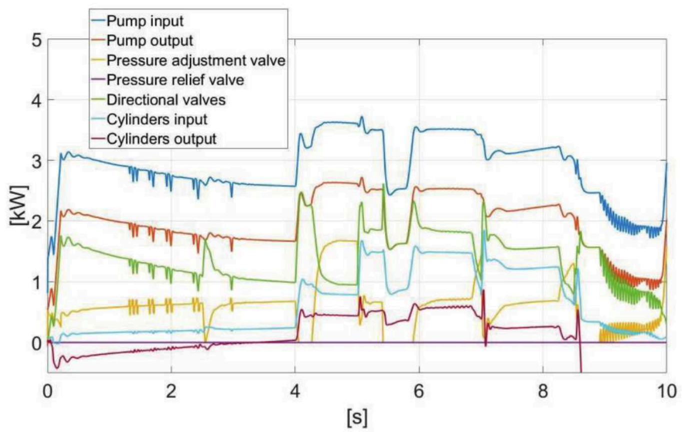

Figure 6: Power distribution during digging and loading cycle

Pump input power, which is the same as the electric motor output, is the mechanical power employed to produce the pressure-dependent torque at the wanted rotational speed. The power model is acquired directly from a data sheet provided by the manufacturer /7/. The pump output power is the hydraulic power leaving the pump, calculated as multiplication of the volumetric flow [m^3/s] and the pressure at the pump [Pa]. The output power was utilized in the directional valves, but a large share of it is lost, mostly in pressure adjustment valve and throttling in directional valves. The power loss of the pump is the difference between input and output power.

The input power of the pressure adjustment valve is the product of pressure at the valve and the flow through the valve. The entire flow returns to tank, so the input power of the PAV is the power loss of the valve. The same applies also for the pressure relief valve, although it is not opening during the test cycles.

Input power of the directional valve group is the product of flow entering the valve and the pressure at the pressure port (P) of the valve. Output power is a combination of outputs of A and B ports, and the power loss is the differential between input and output power.

The hydraulic cylinders transform the hydraulic energy into the actuator work. Losses are caused mainly by the backpressure and mechanical friction. The input power is solely hydraulic power. Output power a combination of mechanical output power and the hydraulic power leaving the cylinder. The volumetric flow out of the cylinder is somewhat problematic. In an ideal system, the leaving flow would jump into the tank in zero pressure. In realistic system, however, the return flow runs through the hoses and valves, which results in a significant pressure rise at the cylinder chamber, the backpressure. It causes a force opposite to working direction, and reduces the net force derived from the actuator. Therefore, using plain net force to calculate the output power gives too disadvantageous picture of the cylinder efficiency. Instead, the leaving flow is count into the output power of the cylinder. The power loss is the differential between input and output power. Power losses are collected in the Table 5.

	Total input	**Pump losses**	**Pressure adjustment valve**	**Directional valves**	**Hoses**	**Actuator losses**	**Actuator work**
Energy, [J]	28 936	9 320	5 852	10 943	2 599	691	10
Share of total loss, [%]	100	32.21	20.22	37.82	8.98	2.39	0.03

Table 5: Energy losses during digging and loading cycle

The actuator work is close to zero as the machine returns to the starting posture in the end of the cycle, where the potential energy is the same as in the beginning.

Second, for levelling cycle Figure 7 demonstrates the power distribution by components.

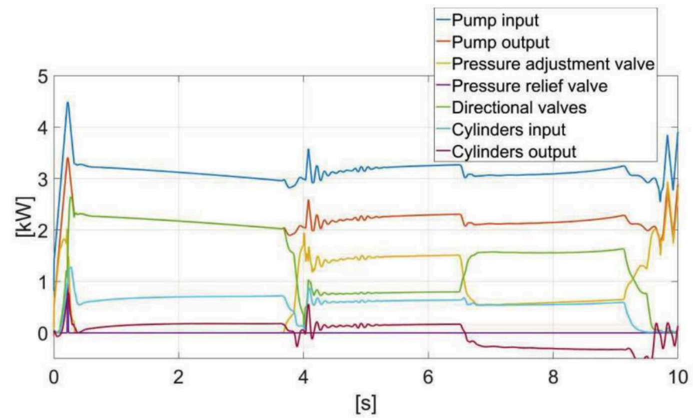

Figure 7: Power distribution during leveling cycle

A similar energy loss calculation is performed for the leveling cycle, and the results are collected in the Table 6

	Total input	**Pump losses**	**Pressure adjustment valve**	**Directional valves**	**Hoses**	**Actuator losses**	**Actuator work**
Energy, [J]	31 251	9 483	7 307	12 089	2 201	646	-7
Share of total loss, [%]	100	30.34	23.38	38.68	7.04	2.07	-0.02

Table 6: Energy losses during levelling cycle

In utilized levelling cycle, the actuator work is on the negative side. Compared to the total input energy, the value is still negligible, denoting that the potential energy at the end position is the same as in the beginning of the cycle. Average values of the two duty cycles are collected in Table 7:

Component	Loss, [J]	Share of total loss, [%]
Pump	9401.5	31.3%
Directional valves	11516.0	38.3%
PAV	6579.5	21.8%
Hoses	2400.0	8.0%
Actuator losses	668.5	2.2%

Table 7: Average energy and loss distribution

The results are consistent between the two cycles. The directional valve is the main energy consumer with 38-39% of total energy. Together with the PAV losses, the total energy lost in the valve group is 58-62%. Pump losses, which include the mechanical and volumetric losses, are 30-32%. Rest of the energy is lost in hose friction (7-9%) and cylinders (2%).

4 Discussion

The developed model is constructed in Matlab/Simulink environment and the parametrization and verification of the simulation model was carried out by measurements on the actual micro excavator. Simulated system consists of the electric motor, hydraulic pump, proportional directional valves and the cylinders. Based on previous studies on the topic /1/, the expectation was that the valves would cause nearly one third of all losses. Efficiency of an excavator was calculated by dividing output power (actuator work) by the total input power. Due to complicated work profile, measuring the total efficiency of a machine is problematic, and knowing the nature and distribution of losses calls for a more detailed information on the dynamic behaviour of the system. The developed model was utilized to simulate a realistic digging-loading cycle of the excavator, and significant sources of energy losses were pointed out. Due to the constant volumetric flow and the open-centre valves, remarkable losses occur whenever the excavator is not moving and during phases of slow movement. During the duty cycles, the throttle losses in valves consume up to 60 % of total input energy.

Therefore, the results indicate a great room for improvement concerning the conventional system. The test case, however, is not completely fair. First, the poor efficiency of a LS system during multiple actuator movement is commonly known. The system pressure was adjusted according to the highest load, and the flow into the other actuators is throttled. In addition, the fixed-volume pump of the micro excavator does not adjust the output flow, as in LS systems, but produces a constant flow. A remarkable portion of the flow is wasted through the pressure adjustment valve, to keep the system pressure level low, according to the low loading condition. According to the simulation, more than 20% of the total power is lost in PAV only. This result calls for more research, and an experimental setup is being prepared to test the excavator in realistic contact with earth. The variable ground contact force will be recorded, and added in to the simulation model, as presented in /5/.

Movement speed of the excavator is dependent on the rotational speed of the electric motor. Increasing the speed would, in this test case, result in more volume lost in PAV. With a more realistic loading condition, in which the PAV would stay unopened most of the time, the additional volumetric flow could be utilized in increased movement speed. This would reduce the cycle time and possibly improve the overall efficiency. The electric motor speed is assumed constant in the simulation model. Additional speed and torque sensor will be fitted for more accurate information on the motor dynamics. A current sensor, which will produce data on the total input energy, will also be prepared for upcoming research.

The actuator losses, which are caused by the frictions in cylinders and the joints of the front hoe, account for only 2% of the total losses in the conventional system. This has given little or no incentive for further research. The friction model could be improved by adding a pressure-dependent component and an angular speed-dependent component, since the current model does not take pressure in to account, and the friction is now a function of cylinder speed only.

5 Summary and Conclusions

A Simulation model of the electro-hydraulic excavator was produced. The model consists of the hydraulic and mechanical systems related to actuation of front hoe, i.e. boom, arm, and bucket. Model parameterization was based on measurement data, when available, and the information found in the literature.

The excavator is fitted with pressure and position sensors, from which the information was collected with a data acquisition system. In addition, a CAN-interface is established in order to communicate with aftermarket directional valves the excavator is fitted with. Comprehensive communication system enables the verification of the simulation model, which gives weight on the obtained results.

The verified simulation model is utilized to study the total energy consumption and power distribution of the micro excavator, and the results are compared and discussed. Two different duty cycles, named digging and loading cycle and levelling cycle, based on the JCMAS standard, was utilized to analyze the power consumption of the excavator.

The results indicate a formidable room for improvement concerning the conventional system, since a power loss of as much as 60% was generated in the directional valve group during two different free-space duty cycles.

Suggested next phase of the research would be the application of external load into the duty cycle. The poor efficiency of a LS-system during unloaded multi-actuator movements is a well-known fact, and operating in this area will give a too negative picture of the efficiency of the conventional system. A 'sandbox' for reproducing the ground contact is being planned, and the load-sensing pins will be attached in all cylinders for accurate load measurement.

The simulation model could be improved with a more accurate model of the electric motor. An additional speed-torque sensor will be installed to acquire information on the motor dynamics, and a current sensor, which will produce data on the total input energy, will be prepared for upcoming research. Finally, the friction model could be improved by adding pressure-dependent and angular-speed-dependent variables.

6 Acknowledgements

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Nomenclature

Variable	*Description*	*Unit*
B_{eff}	Effective bulk modulus	Pa
d	Pipe inner diameter	m
F_c	Coulomb friction	N
F_s	Static friction	N
F_μ	Friction force	N
K_v	Flow coefficient	-

l	Length	m
P	Power	W
p	Pressure	Pa
Δp	Pressure differential	Pa
p_{tr}	Transition pressure	Pa
p_{nom}	Nominal pressure differential	Pa
q	Volumetric flow	m^3/s
q_{nom}	Nominal flow rate	m^3/s
Re	Reynold's number	-
v	Velocity	m/s
V_t	Total volume of the system	m^3
v_s	Stribeck velocity	m/s
z	Average deflection of bristles	m
λ	Friction factor	-
ε	Relative roughness of a pipe	-
ρ	Fluid density	kg/m^3
σ_0	Stiffness of bristles	N/m
σ_1	Damping coefficient	Ns/m
σ_2	Viscous friction coefficient	Ns/m
ζ	Friction factor (single loss)	-

References

/1/ Zimmerman, J.D., Pelosi, M., Williamson, C.A., Ivantysynova, M. Energy Consumption of an LS Excavator Hydraulic System. Proceedings of IMECE2007, 2007 ASME International Mechanical Engineering Congress & Exposition. November 11-15, 2007, Seattle, Washington, USA.

/2/ Maharjan, D., Mahat, C., Liljeström, J., Vauhkonen, N., Kiviluoma, P., Kuosmanen, P. & Sainio, P. Electrification of Excavator. 9th International DAAAM Baltic Conference, Industrial Engineering, 24-26th April 2014, Tallinn, Estonia.

/3/ Hassi, T., Korva, A., Markkula, S., Partanen, T., Sourander, T., Kiviluoma, P., Korhonen, A. & Kuosmanen, P. Improving Energy Efficiency of an Electric Mini Excavator. 11th International DAAAM Baltic Conference, Industrial Engineering, 20-22th April 2016, Tallinn, Estonia.

/4/ Kiviranta, J. Instrumentation of an Automated Excavator. Master's Thesis. Helsinki University of Technology, 2009. Helsinki, Finland.

/5/ Williamson, C., Zimmerman, J., Ivantysynova, M. Efficiency Study of an Excavator Hydraulic System Based on Displacement-Controlled Actuators. Purdue University, Department of Agricultural and Biological Engineering, West Lafayette, Indiana, USA.

/6/ Hippalgaonkar, R., Ivantysynova, M. A Series-Parallel Hydraulic Hybrid Mini-Excavator with Displacement Controlled Actuators. The 13th Scandinavian International Conference on Fluid Power, SICFP2013, June 3-5, 2013, Linköping, Sweden.

/7/ Parker Hannifin Corporation. Gear Pumps/Motors, Series PGP/PGM Fixed Displacement Pumps, Cast-Iron and Aluminium Designs. Catalogue HY30-3300/UK. 2017 Parker Hannifin Corporation.

/8/ Kauranne, H., Kajaste, J., Vilenius, M. Hydraulitekniikka. 1.Painos. WSOY Oppimateriaalit Oy. 2008, Helsinki, Finland.

/9/ Avci, A., Karagoz, I. A Novel Explicit Equation for Friction Factor in Smooth and Rough Pipes. Journal of Fluids Engineering, Vol. 131. 2009, ASME

/10/ Danfoss. Proportional Valve Group PVG 32, Technical Information. 520L0344. Danfoss. March 2016.

/11/ Bak, M.K., Hansen, M.R. Modeling, Performance Testing and Parameter Identification of Pressure Compensated Proportional Directional Control Valves. University of Agder, Norway.

/12/ Ellmann, A., Piché, R. A Modified Orifice Flow Formula for Numerical Simulation of Fluid Power Systems. FPST-Vol. 3, Fluid Power Systems and Tevhnology: Collected Papers. ASME 1996.

/13/ Järf, A. Flow compensation using hydraulic accumulator in direct driven hydraulic differential cylinder application and effects on energy efficiency. Master's thesis. Aalto University, School of Engineering. Espoo, 2016.

/14/ Canudas de Wit, C., Olsson, H., Åström, K. J., Lischinsky, P. A New Model for Control of Systems with Friction. IEEE Transactions on Automatic Control, vol. 40, no. 3, March 1995.

/15/ JCMAS H 020:2007. Japan Construction Mechanization Association. Earth-moving-machinery – Fuel consumption on hydraulic excavator – Test procedures.

/16/ Ville Salomaa, Efficiency study of an electro-hydraulic excavator, Tampere University of Technology, 2017. Helsinki, Finland.