ENERGY EFFICIENCY OF THREE-CHAMBER CYLINDER WITH DIGITAL VALVE SYSTEM

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ABSTRACT

Commonly used hydraulic cylinders have a piston and a piston rod. The piston divides the inside of the cylinder in two chambers and pressures which affect how the piston generates the linear motion. Use of distributed valve system enables several control modes in a system of this type because different control edges can be controlled independently. These control modes can be used for decreasing energy consumption and improving controllability. The traditional hydraulic cylinder has only a limited number of control modes, but by utilizing a multi-chamber cylinder the number of control modes can be increased. In this paper, a three-chamber cylinder is studied using measurements and simulations. The control of the cylinder is presented and measurements are done in a 1-DOF boom mock-up to show the operation of the system in practice. A simulation model is built to investigate further the energy saving capability of the system. The studies show that losses can be significantly reduced by replacing traditional cylinder drives with multi-chamber cylinders.

Keywords: digital cylinder, multi-chamber cylinder, energy efficiency

1 Introduction

A traditional proportional valve is typically capable of driving a cylinder only in inflow-outflow connection or in differential connection. By replacing a traditional, single-spool valve with a distributed valve system, more versatile control options are achieved. As each flow edge is controlled independently, both, the inflow-outflow control mode as well as differential connection become possible and reduction of energy consumption can be achieved with correct selection of control modes (Jansson and Palmberg**,** 1990; Pfaff, 2005).

This paper studies further the three-chamber digital hydraulic cylinder, which is originally presented by Huova and Laamanen (2009). The studied system consists of the three-chamber cylinder and a digital distributed valve system, which drives the cylinder as shown in Fig. 1. Digital valve system has been previously used with traditional cylinders and in the beginning of the research, only inflow-outflow modes were used (Linjama and Vilenius, 2004). Later on the research has been focusing on reduction of energy losses, which has lead into using more control modes (Linjama et al., 2007a; Linjama and Vilenius, 2007b). The number of different exploitable control modes is limited with the conventional cylinder but they can be increased by utilizing multi-chamber digital cylinder. Digital cylinder has three or more chambers and they can be organized in several ways.

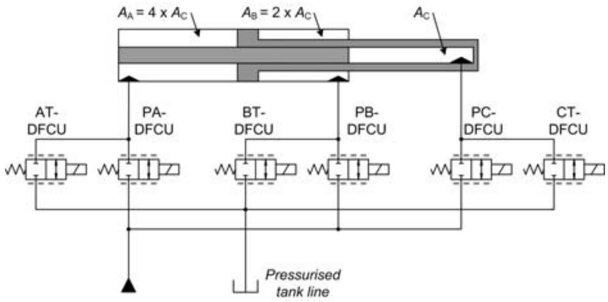

Fig. 1: *A resistance-controlled three-chamber cylinder*

Multi-chamber cylinder is not a new invention but they have been used in several applications for decades. Advantage of multi-chamber cylinder is for example symmetry without extra rod. This enables equal force and displacement to both directions which ease the control of the system (Sampson et al., 2005; Habibi and Goldenberg, 2000). Extra chambers have been used also for balancing the load (Ding, 2004), synchronizing the movement (Mouton, 1985) and generating different forces (Robinson, 1986).

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While Linjama et al. (2009) studied the use of force controlled multi-chamber cylinder, this study concentrates on resistance control of a three-chamber cylinder. To be able to efficiently drive a multi-chamber cylinder, a pressurized tank line along with a constant pressure supply line is used as shown in Fig. 1. In this study, each chamber is controlled with two digital flow control units (DFCU). One DFCU is used to control flow rate between chamber and supply line and the other to control the flow rate between chamber and the tank line. This gives a possibility to connect each chamber to supply line or tank line during motion to enable a number of different control modes.

In traditional proportional valve control, throttling induces significant losses, when the supply pressure is not close to the cylinder load pressures. As a three-chamber cylinder has different control modes with different force ranges, the matching is relatively good regardless of the load force. Principle idea of resistance control of a three chamber cylinder is to drive the cylinder in control mode that gives a maximum force that is only slightly bigger than the load force and to control the velocity of the cylinder by throttling with the DFCUs.

2 Control Strategy

While the traditional two chamber cylinder gives four different control modes to choose from, the three-chamber cylinder enables the use of eight different control modes to both directions. Possible control modes are listed in Table 1, in which letters P and T are used to represent the connection of a chamber to either pump line or tank line. In this study the cylinder chamber areas are sized according to the binary series. The chamber A is twice as big as the chamber B and four times as big as the chamber C. The cylinder is constructed in such way that chambers A and C generate force into extending direction, and the chamber B into retracting direction.

Table 1: Control modes of a three-chamber cylinder

Chamber A	P	P	P	P	T	T	T	T
Chamber B	P	P	T	T	P	P	T	T
Chamber C	P	T	P	T	P	T	P	T

Mode choosing logic uses feasibility information of each mode to select the optimal control mode. Feasibility of each mode is calculated using the force equation of the three chamber cylinder and measured supply and tank line pressures as presented in (Huova and Laamanen, 2009). In the selection of control modes, less energy consuming modes are favored. The selection of control mode uses the feasibility information and calculated energy consumption, when choosing the optimal control mode. When motion is initiated, the logic chooses a control mode that consumes the least energy and stays feasible even if the load force were to increase by a user set hysteresis value. It is common that the load force or pressure of pump line or tank line change significantly during motion and the current mode in use becomes unfeasible. When the mode is not feasible for the duration of a user set time interval the control mode logic switches to another mode, which is feasible, and minimizes the energy consumption.

For example during deceleration, the load force usually decreases allowing the use of lower energy consuming modes, which have smaller force ranges. The less lower energy consuming control mode is enabled only if the mode is feasible for a user set time interval, and the load force is still decreasing at the time of the switching. The described method avoids repeated switching between modes by using hysteresis and time intervals to exclude temporary changes of the system state.

Figure 2 represents the usable pressure range for each chamber during extending motion. Every possible control mode is presented on the horizontal axis so that the least force producing mode (TPT) is at the left side of the diagram and the maximum force generating mode (PTP) is at the right side of the diagram. Vertical axes represent the pressure level of each chamber.

The grey area in the diagram represents the usable pressure range for each mode. For example in mode TPT, the chambers A and C are connected to the tank line, and the usable pressure range in extending direction is between p_T - Δp and p_{min}. Chamber B is connected to the pump line, and the resulting pressure range is between $p_P + \Delta p$ and p_{max}. Black lines represent the chamber pressure references, which are calculated by the mode choosing logic. The pressure reference is chosen so that pressure level of a cylinder chamber stays constant during mode switching whenever it is possible.

Changing between modes TPP-TTT, TTP-PPT and PPP-PTT is more difficult, because the pressure references of the current and the target mode are not equal. Therefore a special transition mode is used during mode switching. During transition mode, the pressure reference of the target mode is used and the chambers are connected to either pump or tank line allowing the pressure reference of both, the target mode and the current mode, to be in the possible pressure range. For example, switching from mode TTP to PPT requires the connection of chamber A and C to supply line and chamber B to tank line during transition mode. This is very energy consuming and the duration of transition mode should be thus minimized. Switching between transition modes is allowed only if the target mode of the current transition mode becomes unfeasible.

Valve control algorithm is a further developed version of a distributed digital valve controller for two-chamber cylinder (Linjama and Vilenius, 2007b). Three main parts of the controller are the determination of search space, calculation of steady-state values and cost function. The basic idea of the controller is to find a good compromise between velocity and chamber pressure error, activity of valves and energy consumption as presented in (Huova and Laamanen, 2009). The upper-level controller is a P-type position controller together with a velocity reference feed forward.

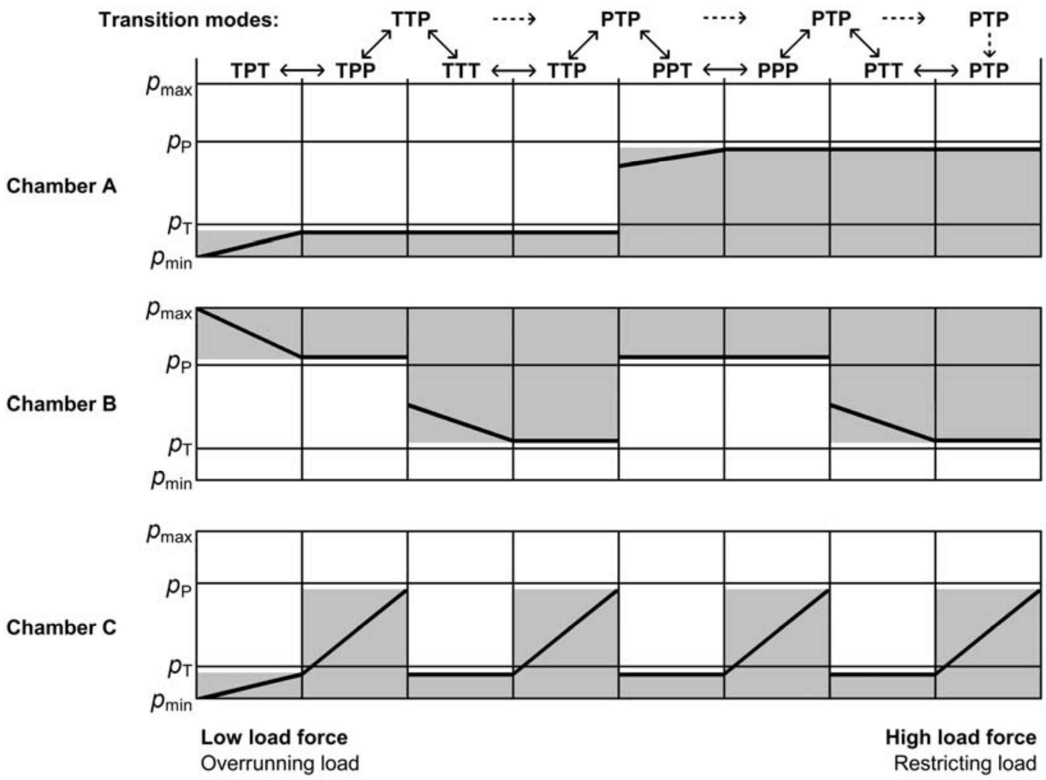

Fig. 2: *Usable pressure range of each cylinder chamber with different control modes, extending movement*

3 Test Application

Figure 3 shows the test application, which is a four meter long boom driven by the three chamber cylinder. Energy consumption of proportional valve controlled two-chamber cylinder was studied previously using the same test system (Linjama and Vilenius, 2007b). As only minimal modifications were done to connect the digital cylinder to the test boom, the energy consumptions can be compared.

Fig. 3: *Test boom*

Table 2: Orifice diameters [mm]

	1	2	3	4	5
PA	1.2	1.5	2.0	3.0	3.0
AT	1.2	1.5	2.0	3.0	3.0
PB	1.2	1.5	2.0	3.0	3.0
BT	1.2	1.5	2.0	3.0	3.0
PC	1.0	1.2	1.5	2.0	
CT	1.0	1.2	1.5	2.0	

The hydraulic system, presented in Fig. 4, consists of supply unit, the digital flow control units and the three- chamber cylinder. The hydraulic energy is generated by a gear pump, which is connected to the hydraulic accumulator. Electrically actuated by-pass valve is used to produce on/off –type pressure control for the supply line. Another hydraulic accumulator along with pressure relief valve forms the pressurized tank line.

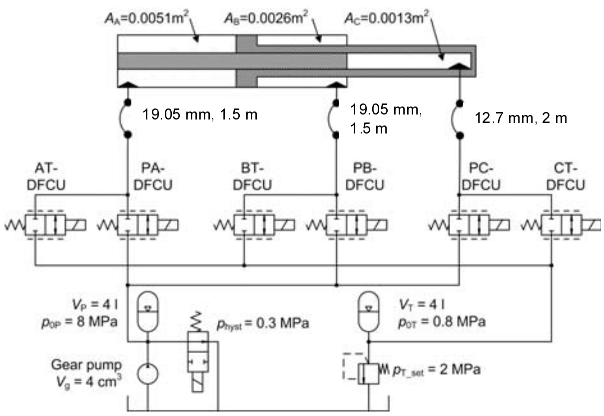

Fig. 4: *Hydraulic diagram of the test system*

The six DFCUs consist of Hydac WS08W-01 on/off-valves. Four on/off-valves form the units PC and CT, while the rest of the units have five valves. Orifices, listed in Table 2, are connected in series with the on/off-valves to maximize the number of unique flow rates through the DFCUs.

Table 3: Tested loading situations

	m_1 [kg]	m_2 [kg]	m_3 [kg]	m_4 [kg]
Load A	150	0	50	200
Load B	0	0	200	200
Load C	200	100	0	100

Measurements are done using three different loading situations which are shown in Table 3. Loading A is an almost balanced loading. Loading B is a restricting loading and C an overrunning loading. Measurements are done with number of different supply pressure settings. All three loadings are measured using supply pressure settings 16, 18 and 20 MPa and tank line pressure setting 2 MPa. The three-chamber cylinder is capable of producing bigger force into positive direction than into negative direction. Therefore the loadings A and B are measured also with lower supply pressures 10, 12 and 14 MPa with slightly modified controller parameters. Controller parameters are presented in (Huova and Laamanen, 2009). All measurements are done three times and average energy consumption is calculated.

4 Experimental Results

Figure 5 shows a response with almost balanced loading, when supply pressure is set to 12 MPa. The figure shows measured position and velocity along with the reference (thin line). Control mode is presented and the sign of the mode information represents the direction of the movement and the absolute value represents the energy efficiency. The smaller the absolute value, the bigger the generated force and the energy consumption are. Thin line represents the use of transition mode. Input and output power (thin line) is shown along with the input and output energy information. The control signals for the two DFCUs of each chamber are shown in separate diagrams (thick line represents the pump line DFCU) as well as the chamber pressures. The pump line pressure (thick line) and tank line pressure levels are shown. The load force (thick line) is calculated from chamber pressures and the estimated load force is shown in the same diagram.

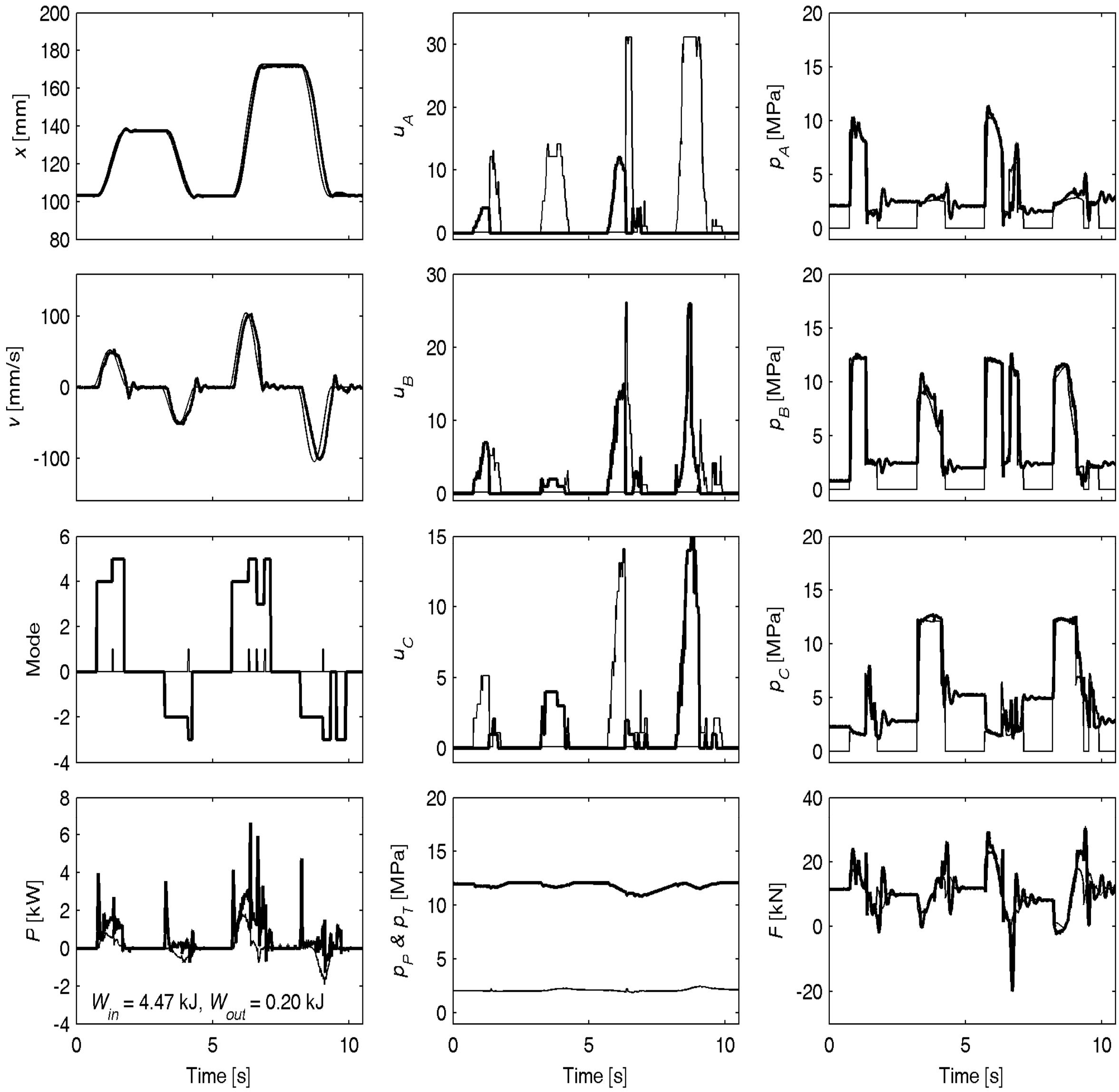

Fig. 5: *Measured response with 12 MPa supply pressure setting and almost balanced loading A*

Figure 6 shows a summary of measured energy losses on different system configurations. Proportional valve is capable of driving all loadings already with 12 MPa supply pressure. A simulation study is made in order to investigate the possibility of using a digital cylinder to drive an overrunning load on lower supply pressure.

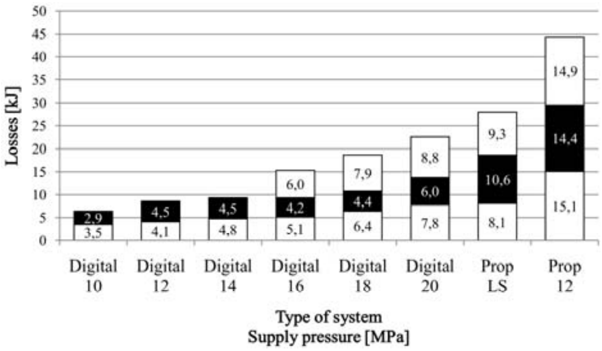

Fig. 6: *Summary of measured energy losses: Loading C on top, Loading B in the middle, Loading A at bottom*

5 Simulation Study

5.1 The Simulation Model

A simulation model is built based on the measurements done with the experimental setup. The simulation model is built using MATLAB/Simulink tools and lumped parameter approach as shown in Fig. 7. The three-chamber cylinder consists of three hydraulic volume models, which are based on the pressure build-up equation. Load mechanism is built also as a Simulink model. Load force of the piston consists of the Simulink based mechanism model and piston friction model. Supply line is modelled as a hydraulic volume, with a capacity of 4 L and bulk modulus of 35 MPa and an on/off-type flow source as in real system. The tank line is modelled also as a hydraulic volume of 4 L and bulk modulus of 7 MPa. Bulk modulus of the volume is tuned to match the behaviour of the measured system. The pressure relief valve is modelled as an orifice, which has an opening proportional to overshoot of the tank line pressure and no internal dynamics.

As the three chamber cylinder is not capable of producing big forces into negative direction with low supply pressures, a system with 18 MPa supply pressure is taken as a starting point for the simulation study. Figure 8 shows a comparison of simulated (black line) and measured results (grey line). Simulated energy losses are 6.3, 4.9 and 8.4 kJ for the loadings A, B and C.

5.2 Reduction of Capacitance

The goal of the simulation study is to further investigate the energy saving potential of the cylinder with optimized parameters. In order to minimize losses, which occur during mode switching, hydraulic capacitances of the system should be minimized. As the pressure of the cylinder chamber is often lowered from supply pressure to tank line pressure by directing flow from the chamber to the tank line, losses occur (Linjama et al., 2009). Also the pressurization of the chamber from supply line induces losses. Capacitance of the hoses (see Fig. 5) is removed in the simulation model in order to investigate the energy consumption of the optimized system. The bulk modulus of the hose was set to 600 MPa in the original simulations.

Simulated energy losses are 5.8, 4.3 and 7.7 kJ for the three loadings. When compared to original simulated results, the losses are reduced by 9 %. The reduction depends on how frequent the mode changes are and what is the difference of supply and tank line pressures. As the supply pressure is increased, the energy loss during mode change is increased as well.

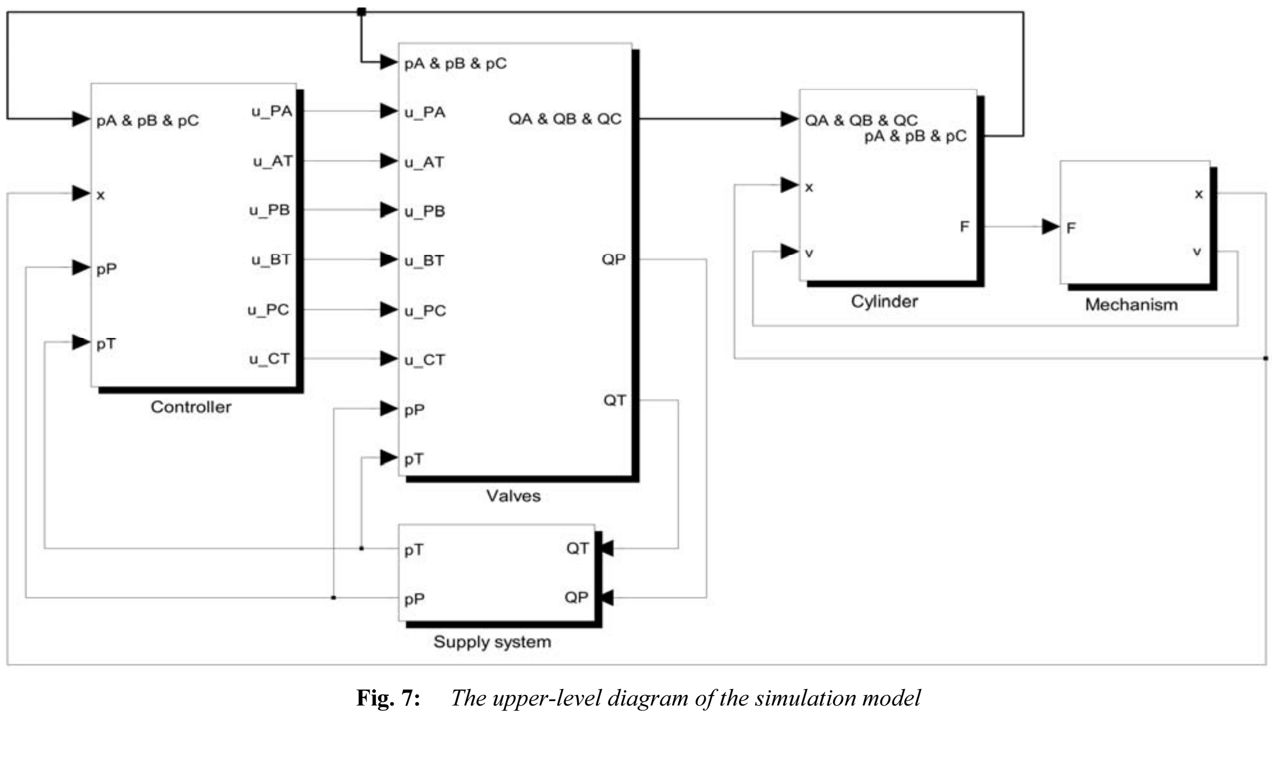

Fig. 7: *The upper-level diagram of the simulation model*

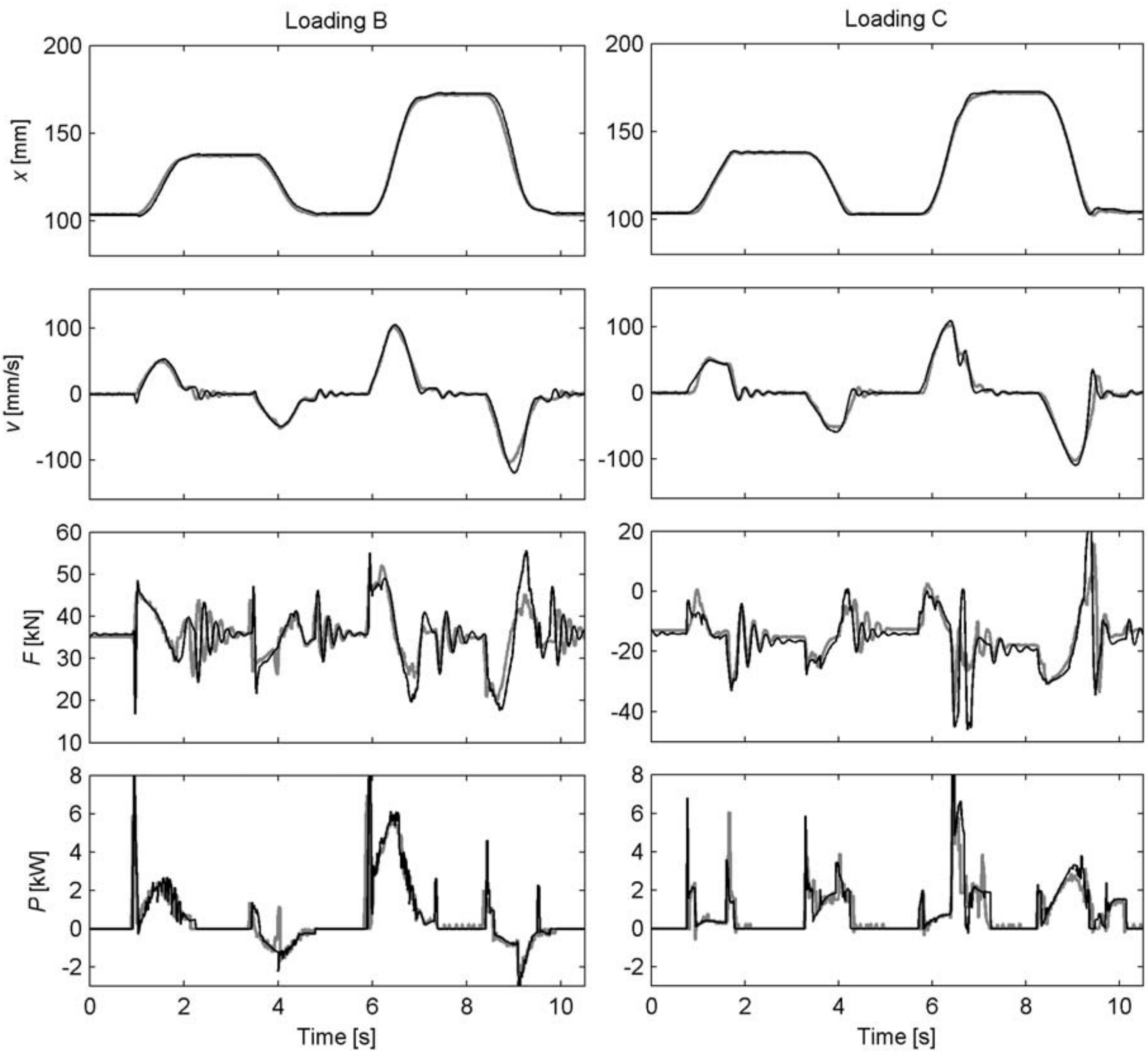

Fig. 8: *Comparison of simulation and measurement data*

5.3 Adjustment of the Chamber Areas

The original three-chamber cylinder is not capable of producing big force into negative direction, making the drive of the overrunning load troublesome. The problem can be overcome by increasing the area of chamber B. Chamber areas in following simulation study are 0.0051, 0.0033 and 0.00075 m^2. The effective area of the chamber A is not changed, while the area of the chamber B is increased one third. In order to maintain same wall thickness for the piston rod, the size of the chamber C is decreased significantly. The series of forces that the cylinder is capable of producing by utilizing different control modes is now changed. Adjustment enables 50 % more force into negative direction, when supply pressure is set to 14 MPa making it possible to drive also the overrunning load.

Parameters of the system are tuned by simulating the three loadings and Fig. 9 shows the response, when the overrunning loading C is driven. One of the adjusted parameters is the weight term of energy consumption in the cost function, which decreases the use of crossflow and thus energy consumption. Simulated energy losses with the optimized system are 5.6, 4.6 and 3.8 kJ with the loadings A, B and C. Summary of the simulation study is shown in Fig. 10. Measured and simulated losses of the original system with 18 MPa supply pressure are shown. The simulated losses of the system with reduced capacitances and 18 MPa supply pressure are also shown as well as the losses of the cylinder with modified chamber areas and supply pressure of 14 MPa.

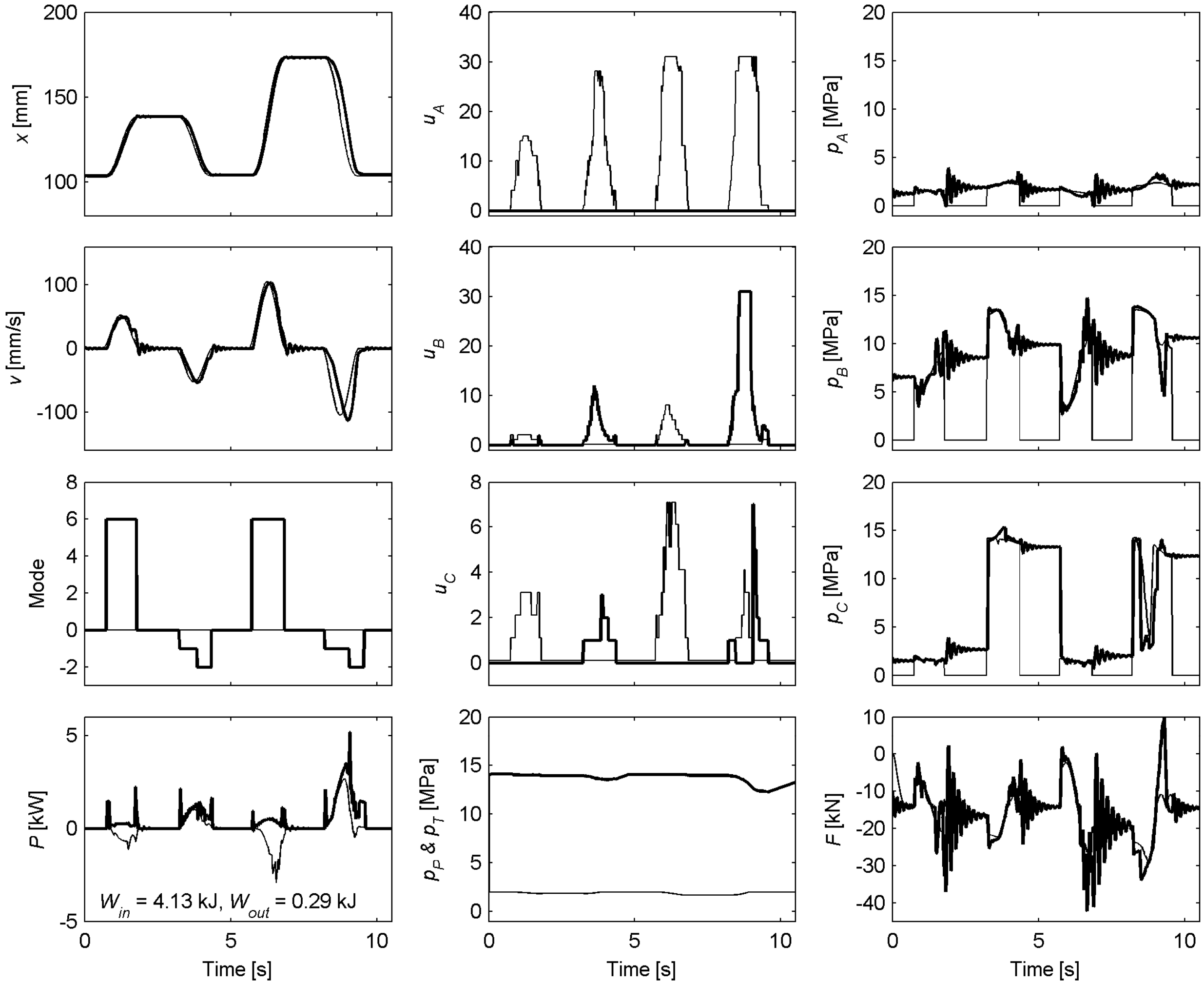

Fig. 9: *Simulated response with 14 MPa supply pressure setting and overrunning loading C*

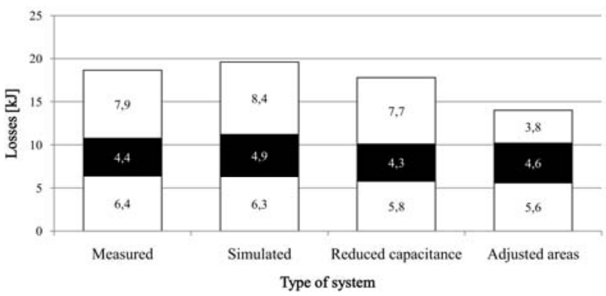

Fig. 10: *Summary of energy losses: Loading C on top, Loading B in the middle, Loading A at bottom*

6 Discussion and Conclusions

Resistance control of three-chamber cylinder is possible by using a model-based control algorithm. Comparison between the proposed system and traditional proportional valve was done. The measured three-chamber cylinder produces significantly higher force to the positive direction resulting from the ratio of the chamber areas. When control of only restricting and balanced loading is needed, the measured energy losses are reduced up to 66 % when compared to traditional load sensing proportional valve system.

Some cylinder applications regularly drive overrunning loads, which require higher supply pressure level if the cylinder measured is implemented. Although the cylinder is capable of driving the negative load with 16 MPa supply pressure, acceptable control performance is achieved only with higher supply pressures. When the three-chamber cylinder with 18 MPa supply pressure is compared to load sensing proportional valve, the average reduction of energy losses is 33 % with the three loadings.

The simulation study was done to investigate further the energy saving potential of the three-chamber cylinder. By changing the effective chamber areas, the cylinder is capable of driving also the overrunning load with 14 MPa supply pressure. In addition to modification of the chamber areas, the capacitance of the system is reduced by excluding the hoses in the simulation model. The simulated system is capable of reducing losses by 50 %, when compared to load sensing proportional valve.

When a load sensing system has two or more active actuators, only the one with the highest load pressure has the energy efficiency of the measured LS-system. Other actuators are driven with unnecessarily high supply pressure and their energy consumption can be estimated from the constant supply pressure measurements of proportional valve. When the simulated three-chamber cylinder with 14 MPa supply pressure is compared to constant supply pressure proportional valve system, the losses are reduced by 68 %.

Nomenclature

A_A, A_B, A_C	Chamber areas of a three chamber cylinder	[m^2]
F	Measured load force	[N]
$m_{1\text{-}4}$	Load masses of the test setup	[kg]
P	Hydraulic power	[W]
p_{0P}, p_{0T}	Pre-pressures of the pump and tank line accumulators	[Pa]
p_{hyst}	Pressure hysteresis of the pump line pressure control	[Pa]
p_{max}	Maximum pressure level of the cylinder	[Pa]
p_{min}	Minimum acceptable pressure level of the cylinder	[Pa]
p_P	Measured pump line pressure	[Pa]
p_{T_set}	Pressure level setting of the pressurized tank line	[Pa]
p_T	Measured tank line pressure	[Pa]
u_A, u_B, u_C	Valve control signals of two DFCUs of each chamber	
V_g	Displacement of the supply pump	[m^3]
V_P, V_T	Accumulator volumes in pump and tank line	[L]
v_{ref}	Velocity reference	[m/s]
x	Piston position	[m]
Δp	Target pressure difference	[Pa]

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