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# ENERGY-EFFICIENT MOTION CONTROL OF A DIGITAL HYDRAULIC JOINT ACTUATOR

Matti LINJAMA and Matti VILENIUS

Institute of Hydraulics and Automation
Tampere University of Technology
P.O.Box 589, FIN-33101 Tampere, Finland
(E-mail: matti.linjama@tut.fi)

**ABSTRACT**

This paper studies the energy efficiency of a heavily loaded cylinder drive. Pulse code modulation is used to control all four flow path independently with parallel connected on/off valves. A valve manifold with 4×5 two-way screw-in cartridge valves is used and the dimensions of the test system are such that actuator size, load force, effective inertia and natural frequency are at the same level as in joint actuators of typical medium size mobile machines. A cost function based control solution is used for online minimization of power losses. Power losses are compared with traditional proportional load-sensing system.

**KEY WORDS**

Digital hydraulics, pulse code modulation, energy efficiency

## INTRODUCTION

The main reason for the use of hydraulic actuators is high power to weight ratio. This is why typical applications are in high-power systems, e.g. mobile machines. Unfortunately, the total efficiency of typical hydraulically actuated machines is very low, even below 5 percent [1]. This yields of excess fuel consumption, emissions and economical losses.

The reduction of power losses has been under active research for decades. The most commonly applied approach is Load Sensing, in which the pump flow is matched according to flow demand of all actuators and the pump pressure is matched according to the highest load pressure. This is only a partial solution because high losses still occur when pressure demands of actuators are not at the same level. More sophisticated solutions are the use of hydraulic transformers [2] or pump-controlled actuators [3]. These are relatively expensive solutions and are not widely used yet. The separate meter-in and separate meter-out (SMISMO) control approach has also been studied quite a lot [4–6]. It gives freedom to control independently velocity and pressure level of the actuator, which can be utilized in energy saving by minimizing pressure losses. Mattila [4] used hydraulic circuit of Fig. 1 (a) to implement a SMISMO controlled cylinder. Energy saving up to 50 percent was demonstrated with overrunning loads. Jansson and Palmberg [5] used four two-way proportional valves as shown in Fig. 1 (b).

Different control strategies were discussed including differential circuit. Liu and Yao [6] added one extra valve to control cross port flow as shown in Fig. 1 (c). This valve was used to minimize pump flow with overrunning loads. Basic problems of SMISMO approach are the need for several high-quality proportional/servo valves as well as complexity of control systems. Typically, a model based controller with closed-loop pressure and flow/velocity controllers are used. Successful pressure control requires high-bandwidth valves and good dynamic model of the system.

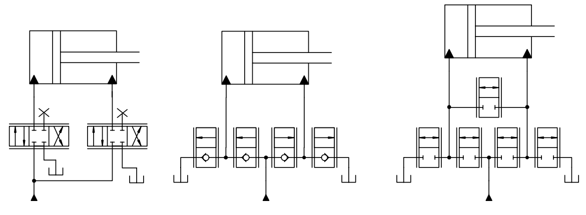

Figure 1. Different SMISMO systems found in literature.

This paper studies so-called digital hydraulic SMISMO system [7]. The basic idea is that each flow path is controlled by a parallel-connected on/off valve series – so called digital flow control unit (DFCU), see Fig. 2. The flow capacities of on/off valves are set according to binary sequence, which is known as pulse code modulation (PCM) approach [8]. The approach gives $2^N$ discrete opening values for each DFCU where $N$ is the number of parallel connected valves. Some benefits of digital hydraulic control approach are that low-cost on/off valves can be used and that the system is fault-tolerant [7, 9].

Although PCM control is an old idea, it has seldom been applied in hydraulic systems. Virvalo [10] used the PCM method to control the velocity of an oil hydraulic cylinder. One DFCU with four valves was installed in the outflow path of the cylinder in order to synchronize piston velocity with trolley conveyor. Similar meter-out control with one DFCU was used by Liu et al. [11], who controlled the position of the joint actuator of a spraying robot. Tanaka [12] implemented a high-speed DFCU for small flow rates. He addressed also the inherent pressure surge problem of PCM control.

The author's research group has studied SMISMO PCM control systems, such as shown in Fig. 2 [7, 9, 13-15]. The aim is to combine good properties of SMISMO control and robust on/off technology. The hydraulic circuit of Fig. 2 is flexible and allows simultaneous control of velocity and pressure level, for example. One difficulty is that the system has $2^{4N}$ different states and the control system must somehow select the best opening combination of valves for each situation. This problem was addressed in [13, 14] where a cost function based control approach was developed. The controller utilized the steady-state model of the system, and did not hence require any dynamic model of the system. The aim of this paper is to show that power losses can also be included in this general framework and that high-quality and energy efficient motion control is possible with slow-response on/off valves without any active pressure control.

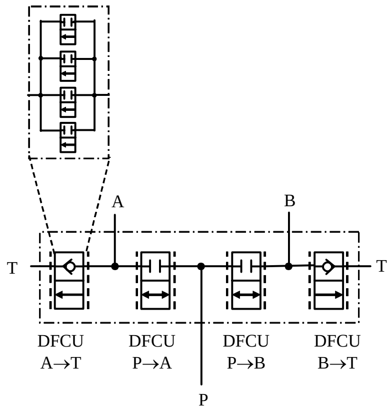

Figure 2. Digital hydraulic SMISMO valve system.

## TEST SYSTEM AND SIMULATION MODEL

The system studied is the same as in [14] and its hydraulic circuit diagram is shown in Figure 3. The system mimics the dynamics of the joint actuator of a typical medium sized mobile machine boom. The design as well as static and dynamic characteristics of the system are presented in [15].

The results of this paper are based on simulations and the simulation model includes rigid body dynamics of the boom, standard models for cylinders and oil volumes together with following assumptions:

- On/off valves are modeled as 30 ms delay plus first order dynamics (time constant 2 ms, maximum rate 200 1/s)
- On/off valves of DFCUs P→A and P→B allow flow in both directions
- Flow capacities of on/off valves are the same as in [14], i.e. the largest valve 19 l/min @ $\Delta p$ = 0.5 MPa and the others approximately according to binary series
- Pump is modeled as variable supply pressure source with first order dynamics (time constant 50 ms)
- Hose dynamics are neglected but the effect of hoses on the effective bulk modulus is included

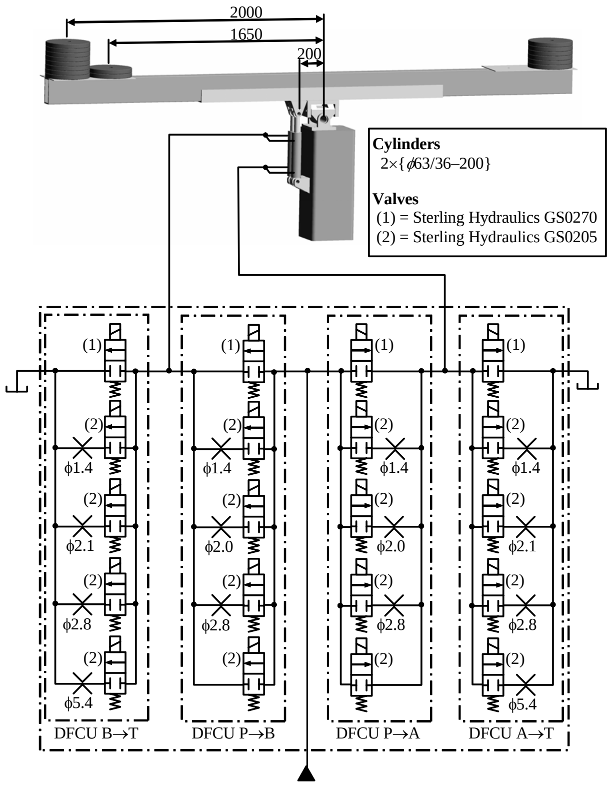

Figure 3. Hydraulic circuit diagram of test system.

Figure 4 compares simulated and measured responses. It is seen that the simulation model predicts accurately the dynamic response of the system and that the system dynamics is dominated by a 4 Hz natural frequency.

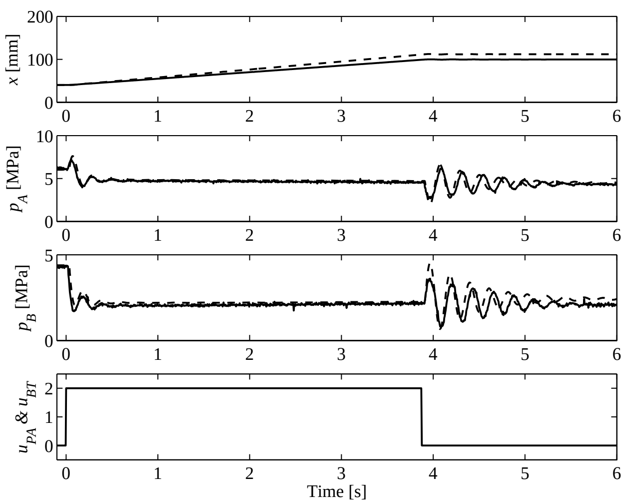

Figure 4. Simulated (---) and measured (—) response when second smallest valves are opened and closed. The load masses are 200+100 kg at piston side and 100+0 kg at the other end.

## CONTROL PRINCIPLE

The control principle is similar to predictive control approach: the future response of the system is calculated via a model and valve controls are selected by minimizing a cost function. The steady-state model of the system is used:

$$\begin{aligned} &Q_{N,PA}(u_{PA})\sqrt[*]{p_P - p_A} - Q_{N,AT}(u_{AT})\sqrt[*]{p_A - p_T} = A_A v \\ &Q_{N,PB}(u_{PB})\sqrt[*]{p_P - p_B} - Q_{N,BT}(u_{BT})\sqrt[*]{p_B - p_T} = -A_B v \\ &F = A_A p_A - A_B p_B \end{aligned} \tag{1}$$

where $\sqrt[*]{\bullet}$ is a shortcut for signed square root $\mathrm{sgn}(\bullet)\sqrt{|\bullet|}$, and $Q_{N,PA}$, $Q_{N,AT}$, $Q_{N,PB}$ and $Q_{N,BT}$ are flow coefficients of DFCUs P→A, A→T, P→B and B→T, respectively (see Fig. 3). Each flow coefficient depends on the state $u$ of the DFCU and has therefore $2^5$ different values. The steady-state velocity and pressures can be solved from Eq. 1, if the load force $F$ as well as supply and tank pressure of the system are known. The system has $2^{20} \approx 10^6$ different states, which makes the real-time calculation of all combinations impossible. Therefore, the reduced search space is defined by analyzing flow balance of both chambers separately (see [14] for details). The steady-state velocity and pressures are then solved from Eq. 1 for all elements of the reduced search space by using Newton-Raphson iterations. Finally, the cost-function values are calculated for each steady-state solution and the best opening combination is selected. This process is repeated at each sampling instant. It is important to note that the approach allows control of all four flow paths simultaneously. The inputs of the controller are target velocity and pressures, measured or estimated supply and tank pressures, and measured or estimated load force.

## MINIMIZATION OF POWER LOSSES

The hydraulic input power of the system is

$$P_{in} = Q_P p_P \tag{2}$$

where $Q_P$ is flow rate into the valve system and $p_P$ is pressure at the P-port. Following rules and assumptions are used in the development of the control strategy:

- The lowest allowed target pressure is 2 MPa in order to avoid cavitation
- Pressure differential over active DFCUs should be at least 2 MPa for sufficient controllability
- Tank pressure $p_T$ is zero

Four cases are studied: large restricting load and large overrunning load at both directions.

**Restricting load force**

The control strategy for restricting load is straightforward. The target pressure for the meter-out side is 2 MPa, and the target pressure of meter in-side is calculated according to load force and meter-out pressure. Let us consider the extending movement with restricting load force. Pressure references can be calculated as follows:

$$\begin{aligned} p_{B,ref} &= 2\ \text{MPa} \\ p_{A,ref} &= \left(p_{B,ref} A_B + F\right) / A_A \\ p_{P,ref} &= p_{A,ref} + 2\ \text{MPa} \end{aligned} \tag{3}$$

where it has been assumed that the load force is so big that $p_{A,ref}$ is bigger than 2 MPa. The power losses can be minimized by minimizing flows P→B and A→T. As the results of [13] show, these 'short-circuiting' flows are needed only at lower velocities for improving controllability. The suggested cost function for extending movement with restricting load is

$$\begin{aligned} J = &\left(v_{ref} - v\right)^2 + K_1\left\{\left(p_{A,ref} - p_A\right)^2 + \left(p_{B,ref} - p_B\right)^2\right\} + \ldots \\ &K_2\left[\left\{Q_{N,PB}\left(p_P - p_B\right)^{3/2}\right\}^2 + \left\{Q_{N,AT}\left(p_A - p_T\right)^{3/2}\right\}^2\right] \end{aligned} \tag{4}$$

The cost function has three quadratic terms, one for velocity error, one for pressure errors and one for power losses in DFCUs P→B and A→T. Tuning parameters $K_1$ and $K_2$ can be used to find good trade-off between velocity/pressure tracking and power losses. Velocity and pressures in Eq. 4 are not measured but calculated from steady state model of Eq. 1. The retracting movement with restricting load can be handled similarly.

**Overrunning load force**

For overrunning load force, the meter-out pressure must be so big that the minimum pressure requirement is met at the meter-in side. For extending motion, pressure references can be calculated as follows:

$$\begin{aligned} p_{A,ref} &= 2\ \text{MPa} \\ p_{B,ref} &= \left(p_{A,ref} A_A - F\right) / A_B \\ p_{P,ref} &= p_{A,ref} + 2\ \text{MPa} \end{aligned} \tag{5}$$

Again a large load force is assumed, which yields the condition $p_{B,ref} > p_{P,ref}$. In this case, power losses can be minimised by allowing flow B→P and minimizing flows A→T and B→T, i.e. differential circuit. Small flow from pump is needed to compensate for piston rod area. Energy saving is significant especially if high supply pressure must be used because of demands of other actuators. The cost function for this situation is

$$\begin{aligned} J = &\left(v_{ref} - v\right)^2 + K_1\left\{\left(p_{A,ref} - p_A\right)^2 + \left(p_{B,ref} - p_B\right)^2\right\} + \ldots \\ &K_2\left[\left\{Q_{N,BT}\left(p_B - p_T\right)^{3/2}\right\}^2 + \left\{Q_{N,AT}\left(p_A - p_T\right)^{3/2}\right\}^2\right] \end{aligned} \tag{6}$$

The retracting motion can be handled similarly, but the meter-out flow is bigger than meter-in flow. The excess flow can be drained to tank via DFCU A→T or it can be utilized in other actuators. Therefore, the retracting motion with large overrunning load requires no pump flow and power losses are zero.

## SIMULATION RESULTS

**Implementation of Controller**

The controller is implemented according to block diagram of Figure 5. Trajectory generator generates the fifth order polynomial position reference $x_{ref}$ and corresponding velocity reference $v_{ref}$. Position feedback and velocity feedforward are used to generate the closed-loop velocity reference $v_{ref2}$. The load force estimate $F$ is calculated from the chamber pressures by using a second order low-pass filter. Pressure references are calculated by using Eqs 3 and 5, and finally the optimisation procedure is used to find minimum of the cost function (Eqs 4 and 6) and corresponding optimal control. The details of the optimisation procedure are given in [13, 14].

The block diagram shows that required measurements are chamber pressures, supply (and possibly tank) pressure and piston position. However, chamber pressures are used only for estimation of load force and not for active pressure control. The numerical values of main parameters are given in Table 1.

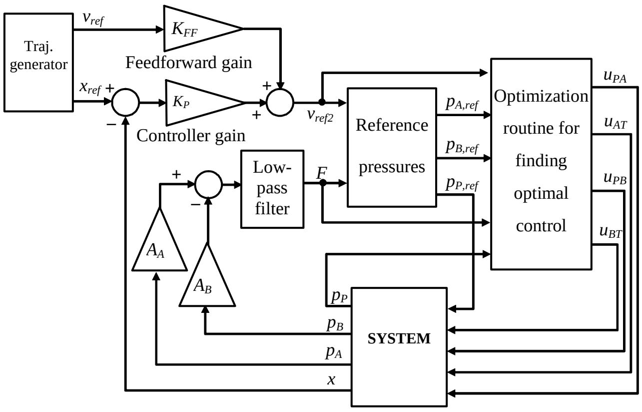


Figure 5. Block diagram of the control system.

Table 1. Main parameters of the control system.

| | |
|---|---|
| Sampling time [ms] | 40 |
| Controller gain $K_P$ [1/s] | 2.5 |
| Feedforward gain $K_{FF}$ [-] | 0.8 |
| Weight for pressure error $K_1$ | $7\times10^{-19}$ |
| Weight for power losses $K_2$ | $1\times10^{-11}$ |
| Break frequency of low-pass filter [rad/s] | 15 |

**Simulated response with digital hydraulic valve**

Figure 6 presents the simulated response when the load masses at boom tips are 200+200 kg at cylinder side and 0+0 kg at the other side. The load force is thus strongly restricting for extending movement and strongly overrunning for retracting movement. Figure shows that the system works as designed: inflow-outflow control is used in extending movement and differential circuit in retracting movement. The input power is even negative in retracting movement. The total work is 4710 J if negative power is considered and 5190 J if negative power is neglected.

It can be seen in Fig. 6 that the position and velocity tracking is good and similar to experimental results in [14]. Slight cavitation exists in large retracting movement and it could be prevented by adopting the anti-cavitation scheme presented in [15].

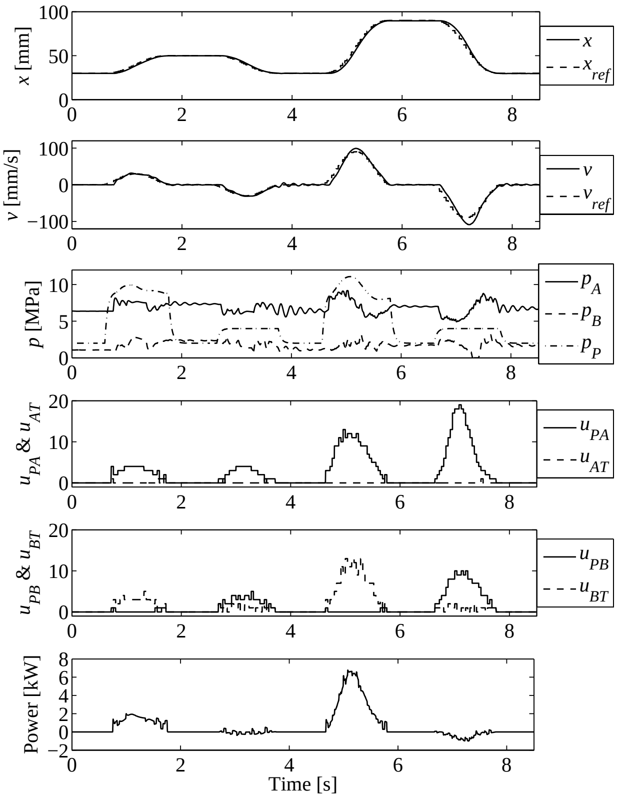


Figure 6. Simulated response with digital hydraulic valve system.

**Simulated response with proportional valve**

In order to compare power losses to traditional valve solution, the simulation of Fig. 6 is repeated with a proportional valve. The valve has matched flow areas, no hysteresis or non-linearity and first order dynamics with 30 ms time constant. The valve does not have pressure compensator. In the extending movement, the pump pressure is adjusted to be 1 MPa higher than $p_A$. In the retracting movement the pump pressure is adjusted 6 MPa higher than $p_B$ in order to avoid cavitation. Similar P-controller plus feedforward is used as a closed-loop controller. Figure 7 shows the simulated response. It is seen that the tracking performance is similar to digital hydraulic system but power losses are higher. The total work is 7340 J, which is 56 percent more than in digital hydraulic solution.

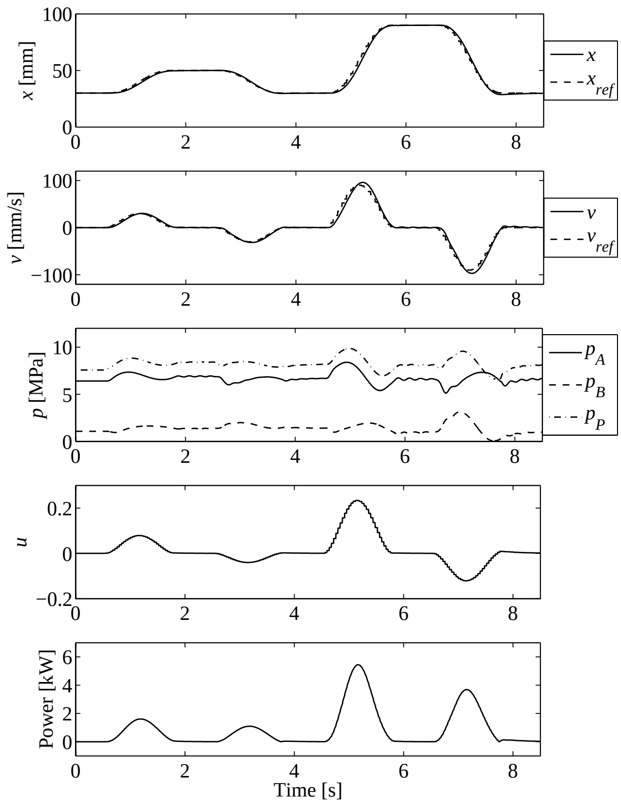


Figure 7. Simulated response with proportional valve.

## SUMMARY

The main result of this paper is that energy efficient motion control is possible with rather slow-response on/off valves. The digital hydraulic valve system allows independent control of all flow paths, which is big advantage when compared to traditional valves where flow paths are mechanically linked together. In this paper, the flexibility of the valve system was utilized by using differential connection when lowering load. This resulted in 36 percent reduction in power losses when compared to traditional valve. The energy saving control was implemented by including power loss terms in the cost function. In this approach, there is no need for active pressure control or accurate dynamic model as in other SMISMO solutions.

The results of this paper are based on simulations and the next step of the research is experimental validation. It can be expected that the real system has more pressure peaks because real valves have variation in delays. The bumpless transfer between different control modes is also an important research topic because the load force may change its sign during the motion in real machines.

## ACKNOWLEDGEMENT

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