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ACCURATE TRAJECTORY TRACKING CONTROL OF WATER HYDRAULIC CYLINDER WITH NON-IDEAL ON/OFF VALVES

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Abstract

The aim of the work is to develop an on/off valve based trajectory tracking control solution without fast and/or continuous switching of valves. The pulse code modulation method is used to realise stepwise control of inflow and outflow of the actuator. Both inflow and outflow paths have five parallel-connected two-way solenoid valves, each having different flow capacity according to binary series, and a four-way on/off valve is used for changing direction of movement. Cost function based open-loop and closed-loop control solutions are developed and it is demonstrated how the cost function weights can be used to find a reasonable trade-off between tracking performance and pressure surges. Closed-loop results show accurate and reasonably smooth position tracking and simultaneous pressure level control. Achieved control performance is close to that of water hydraulic servo systems.

Keywords: Pulse Code Modulation, On/Off Control, Tracking Control

1 Introduction

The controllability of modern water hydraulic systems is rapidly reaching that of oil hydraulics. Good servo and proportional valves exist (Koskinen et al, 1996; Hyvönen et al, 1997; Takahashi et al, 1999) and good control results can be achieved with these valves (Mäkinen and Virvalo, 2001; Sanada, 2002; Cho et al, 2002). The main obstacle for wider use of water hydraulic servo systems is the high price of valves. This is partly caused by the small number of valves produced, but also by the special requirements due to water (wear and corrosion resistance, leakage, etc.). High price level has restricted the use of water hydraulic servo systems to special applications in which oil hydraulics cannot be used.

Position trajectory tracking control is one basic application of hydraulic servo systems, but only few papers can be found in the literature dealing with water hydraulic tracking control systems. Mäkinen and Virvalo (2001) used a combination of position and velocity controllers for position tracking control of a water hydraulic cylinder. The state controller approach with position, velocity and acceleration feedback was used and the system was a heavily loaded cylinder driven by a servo valve. The natural frequency of the system was 90 rad/s and results showed 2 mm tracking error with 200 mm/s peak velocity. Cho et al (2002) utilised the adaptive sliding mode tracking control for the position control of a low-pressure water hydraulic cylinder. The natural frequency of the system was not given, but the cylinder and load mass were the same as in this paper. A water hydraulic proportional valve was used and the achieved tracking performance was 3 mm tracking error with 100 mm/s peak velocity.

On/Off control is an interesting alternative for proportional control because there are low-cost and reliable water hydraulic on/off valves on the market especially for lower pressure levels (Linjama et al, 2000a). Pulse Width Modulation (PWM) and its variants are the most popular methods for implementing proportional control with on/off valves. The inherent drawback of PWM control is high frequency and continuous switching of valves, which causes noise and rapid wear of valves. In addition, the low bandwidth of water hydraulic on/off valves prevents the use of PWM-like control methods in most water hydraulic applications. An alternative way to achieve almost proportional control with on/off valves is Pulse Code Modulation (PCM), in which several on/off valves of different sizes are connected in parallel to form a *digital flow control unit*. In particular, if flow capacities of valves are in ratios of 1:2:4:8 etc., in total 2^n different flow rates are achieved with *n* valves. This is an old idea (Bower, 1961), but it is quite seldom applied in hydraulic systems. Miyata et

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al (1991) used the PCM method to control position and pressure of a pneumatic cylinder drive. A special feature of this research was that four digital flow control units were used to independently control inflow and outflow of cylinder chambers. Virvalo (1978) used the PCM method to control the velocity of an oil hydraulic cylinder and Liu et al (2001) applied the method in the position control of an oil hydraulic spraying robot. Recently, Laamanen et al (2002) utilised the PCM method in velocity control of a water hydraulic motor and Linjama et al (2002) applied it in position control of a water hydraulic cylinder.

This paper further develops the initial control results of Linjama et al (2002). The hydraulic system is similar and consists of two digital flow control units for controlling independently inflow and outflow of the cylinder. Earlier research studied the special case of equal opening at both digital flow control units and only step responses were presented. This paper develops a general tracking control solution which allows simultaneous control of velocity and pressure level and is not limited to equal openings. Experimental results show that accurate and rather smooth position tracking control is possible with an inexpensive on/off control system. Achieved control performance is of the same level as that presented earlier in the literature (Mäkinen and Virvalo, 2001; Cho et al, 2002). The rest of the paper is organised as follows. Section 2 introduces the concept of cost function based open-loop control, discusses different cost functions and proves that the concept can be used also in closed-loop control. Section 3 introduces the test system, Section 4 explains the controller implementation, Section 5 presents open- and closed-loop results and Section 6 presents conclusions.

2 Cost Function Based Control of PCM System

2.1 Definitions and Assumptions

The hydraulic circuit of the suggested PCM system is shown in Fig. 1. A digital flow control unit is installed in both the inflow and outflow path of the system, allowing separate meter-in meter-out flow control. This additional degree of freedom is used to control steady-state pressure levels in cylinder chambers. The four-way valve is used to select the direction of the movement. Another possibility would be to use four digital flow control units as in Miyata et al (1991). The controller is developed under the following assumptions:

2.a) Load force is known and slowly varying.
2.b) Supply pressure is known and slowly varying.
2.c) Valves are infinitely fast.
2.d) Pressure losses of pipes, couplings and the four-way valve are negligible.
2.e) System dynamics is fast compared to the sampling rate of the controller.

Slowly varying load force and supply pressure means that they are essentially constant between sampling instants. The assumptions are used to simplify controller development and later it is shown experimentally that the controller can be used even if these assumptions are not fully satisfied.

Referring to Fig. 1, the pump side digital flow control unit has n_P valves connected in parallel. The flow capacities of valves with one Pa pressure differential are denoted by $Q_{N,P1}$, $Q_{N,P2}$ etc. and the control signals of valves are denoted by u_{P1}, u_{P2} etc. Usually – but not

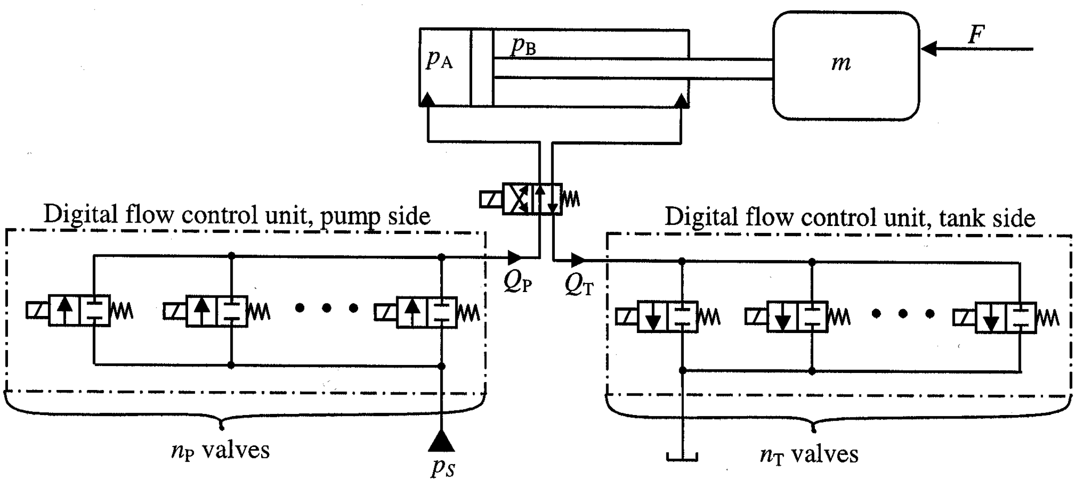

Fig. 1: *The hydraulic circuit of the suggested PCM system*

Table 1: The relation between the state of the digital flow control unit and the control signals of individual valves

u_P	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	…	31
u_{P1}	0	1	0	1	0	1	0	1	0	1	0	1	0	1	0	1	0	1	…	1
u_{P2}	0	0	1	1	0	0	1	1	0	0	1	1	0	0	1	1	0	0	…	1
u_{P3}	0	0	0	0	1	1	1	1	0	0	0	0	1	1	1	1	0	0	…	1
u_{P4}	0	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	0	0	…	1
u_{P5}	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	1	…	1

necessarily – the flow capacities of valves are according to binary series such that the next valve is two times bigger than the previous one. In any case, the flow control unit has in total 2^{n_P} states denoted by an integer number u_P between 0 and $2^{n_P}-1$. The relation between the state u_P and valve control signals u_{Pi} is binary u_{P1} corresponding to the least significant bit. As an example, Table 1 shows this relation for $n_P = 5$.

The effective flow capacity of the flow control unit is given by

$$Q_{N,P} = \sum_{i=1}^{n_P} u_{Pi} Q_{N,Pi} \quad (1)$$

The vector of all nonzero flow capacities, $Q_{N,P}$, is formed such that its i-th element is calculated according to Eq. 1 using valve control signals corresponding to the state $u_P = i$. The above discussion applies to the tank side digital flow control unit by changing subscript 'P' to 'T'.

2.2 Steady-State Velocity and Pressures

Referring to Fig. 1 and assuming extending direction of movement, the steady-state equations of the system are

$$\begin{aligned} Q_P &= \mathbf{Q_{N,P}}(u_P)\sqrt{p_S - p_A} = A_A v \\ Q_T &= \mathbf{Q_{N,T}}(u_T)\sqrt{p_B} = A_B v \\ F &= A_A p_A - A_B p_B \end{aligned} \quad (2)$$

where notations $Q_{N,P}(u_P)$ and $Q_{N,T}(u_T)$ are used to denote u_P-th and u_T-th elements of vectors $Q_{N,P}$ and $Q_{N,T}$, respectively. Solving for steady-state velocity and pressures gives

$$\begin{aligned} p_A &= \frac{z^2 p_S + \gamma^2 F / A_B}{z^2 + \gamma^3} \\ p_B &= \frac{\gamma z^2 p_S - z^2 F / A_B}{z^2 + \gamma^3} \\ v &= \frac{\mathbf{Q_{N,P}}(u_P)}{A_B}\sqrt{\frac{\gamma p_S - F / A_B}{z^2 + \gamma^3}} \\ \gamma &= A_A / A_B \\ z &= [\mathbf{Q_{N,P}}(u_P)] / [\mathbf{Q_{N,T}}(u_T)] \end{aligned} \quad (3)$$

Steady-state velocity and pressures can be solved similarly for the retracting direction and the result is

$$\begin{aligned} p_A &= \frac{\gamma^2 p_S + \gamma^2 F / A_B}{z^{-2} + \gamma^3} \\ p_B &= \frac{\gamma^3 p_S - z^{-2} F / A_B}{z^{-2} + \gamma^3} \\ v &= -\frac{\mathbf{Q_{N,T}}(u_T)}{A_B}\sqrt{\frac{p_S + F / A_B}{z^{-2} + \gamma^3}} \end{aligned} \quad (4)$$

Equation 3 and 4 are valid only if load force is between $-A_B\, p_S$ and $A_A\, p_S$ and if calculated pressures are positive. It is seen that steady-state velocity and pressures depend on the states of digital flow control units, load force, supply pressure and direction of movement. The direction of movement is defined by the control signal of the four-way valve, u_{dir}, which is +1 for extending direction and –1 for retracting direction.

2.3 Example of Steady-State Velocity and Pressure with Binary Coded Flow Control Units

An important special case of digital flow control unit is the case in which valve flow capacities are according to binary series, i.e., exactly in ratios of 1:2:4:8 etc. In this case, the elements of vectors $Q_{N,P}$ and $Q_{N,T}$ are equal to $u_P Q_{N,P1}$ and $u_T Q_{N,T1}$, respectively, and the relation between the state and opening of the digital flow control unit is 'linear' as shown in Fig. 2.

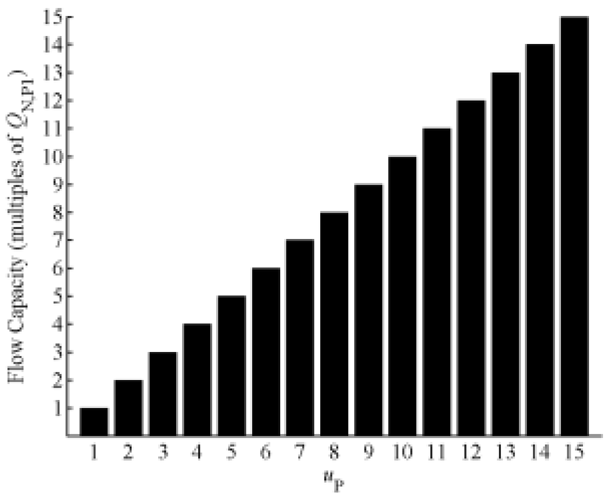

Fig. 2: *Relation between the state and flow capacity of binary coded digital flow control unit, n_P=4*

As an example of a system with binary coded digital flow control units, Fig. 3 and 4 show calculated steady-state velocity and A-side pressure for extending movement for a system with the following parameters: cylinder diameter 32 mm, rod diameter 16 mm, p_S = 30 bar, F = 0 N, $n_P = n_T = 4$, $Q_{N,P1} = Q_{N,T1} = 1\times10^{-8}$ $m^3/(s\sqrt{Pa})$. Figure 3 shows also a constant velocity contour for 0.15 m/s velocity. It is seen that for essentially the same velocity there are several u_P–u_T combinations with clearly different pressure levels. This feature makes it possible to control almost independently velocity and pressure levels.

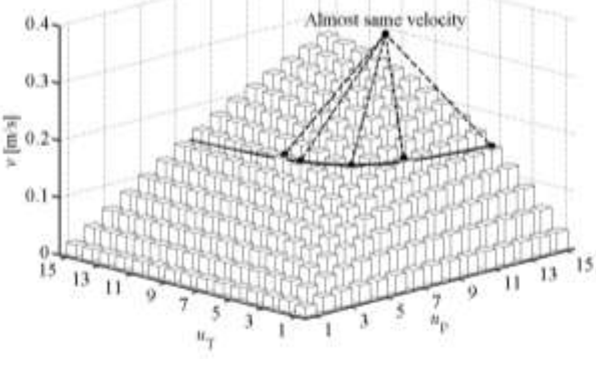

Fig. 3: *Steady-state velocity as a function of states of digital flow control units*

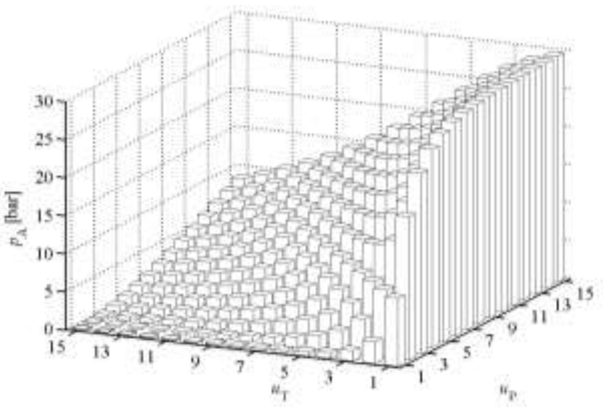

Fig. 4: *Steady-state pressure in A-chamber as a function of states of digital flow control units*

2.4 Cost Function Based Open-Loop Control

The operation principle of the suggested control method is to select at each sampling instant the states of digital flow control units (u_P and u_T) such that a given cost function is minimised. The cost function is based on calculated steady-state values of Eq. 3 and 4, and the assumption 2.e is essential so that a steady-state situation is well achieved before the next sampling instant. Some possible cost functions are

$$J(k) = \left[v_r(k) - \hat{v}(k)\right]^2 \tag{5}$$

$$\begin{aligned} J(k) &= \left[v_r(k) - \hat{v}(k)\right]^2 + K_1\left[\hat{p}_A(k-1) - \hat{p}_A(k)\right]^2 \\ &+ K_2\left[\hat{p}_B(k-1) - \hat{p}_B(k)\right]^2 \end{aligned} \tag{6}$$

$$\begin{aligned} J(k) &= \left[v_r(k) - \hat{v}(k)\right]^2 + K_3\left[p_{Ar}(k) - \hat{p}_A(k)\right]^2 \\ &+ K_4\left[p_{Br}(k) - \hat{p}_B(k)\right]^2 \end{aligned} \tag{7}$$

Where the hat "^" notation is used to denote the fact that velocity and pressures are not measured but calculated using Eq. 3 and 4. The first version considers only the steady-state velocity error. Although it gives the smallest velocity error, it may result in high pressure surges. The second version weights also the change in calculated chamber pressures and therefore reduces pressure variations at the cost of velocity error. The third version sets weight on the difference between calculated and target pressures of cylinder chambers. This allows the simultaneous control of pressure level and velocity. The coefficients of cost function can depend e.g. on piston position, velocity or direction of movement. The operation principle of the open-loop controller is as follows:

- Read current reference velocity and reference pressure values.
- Calculate steady-state velocity and pressures with all state combinations of digital flow control units.
- Calculate the value of cost function with all calculated steady-state values.
- Select the state combination (u_P, u_T) that minimises the value of cost function.
- Wait for the next sampling instant.

This is a 'brute force' solution for finding the optimal state combination as all combinations are calculated at each sampling instant. It is clear that the optimum could be found much more effectively using discrete optimisation, but this is not the topic of this paper. The block diagram of the open-loop controller is shown in Fig. 5(a).

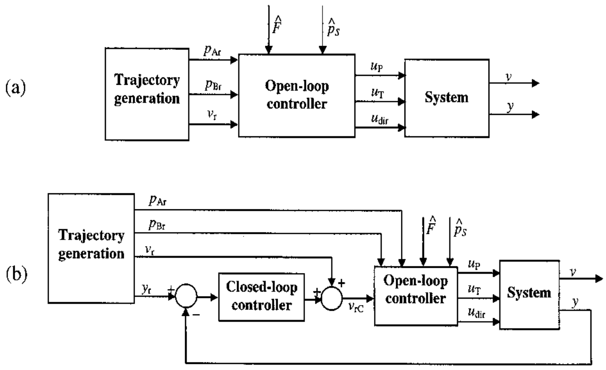

Fig. 5: *Block diagrams of open-loop (a) and closed-loop controller (b)*

2.5 Closed-Loop Control

A drawback of the open-loop control is its sensitivity to disturbances. Accurate open-loop control requires accurate knowledge of system parameters and good estimates for the load force and supply pressure. Especially the load force is difficult to know accurately because it includes the friction force of the piston. In addition, open-loop strategy is based on the assumption that the system achieves its steady-state velocity after a negligibly short transient period, which is seldom true in real systems. These errors accumulate in position control and therefore pure open-loop control is suitable only for manually controlled systems or velocity control of hydraulic motors. However, the open-loop control strategy can be used as a feedforward term to improve the closed-loop position tracking. The studied closed-loop control structure is shown in Fig. 5(b). Its core is the open-loop controller and the closed-loop position controller corrects the velocity reference according to measured position error.

The use of this closed-loop control structure can be justified by the fact that the 'gain' of the open-loop controller block does not change its sign even if the load force estimate is incorrect. In other words, increase in the velocity demand v_{rC} yields increase in the piston velocity and vice versa. This can be seen in the third expression of Eq. 3 or 4, where the influence of load force is a common multiplication factor independent of combinations of u_P and u_T.

3 Test System

The hydraulic circuit of the test system is depicted in Fig. 6. It consists of a water hydraulic pump unit, two digital flow control units, a four-way valve and asymmetric cylinder with inertial load. Each flow control unit consists of five directly operated solenoid valves. The largest valve is without external orifice and other valves have an orifice such that the flow ratios follow approximately the binary series. These valves

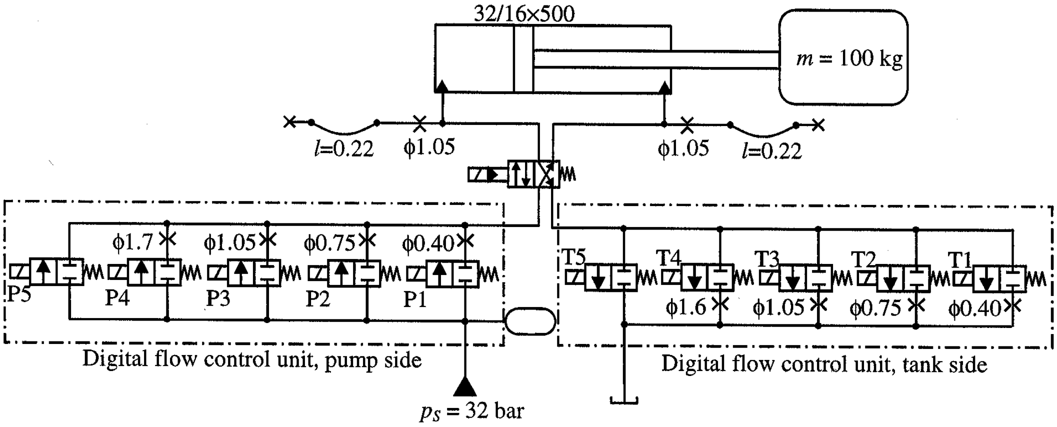

Fig. 6: *The hydraulic circuit of the test system*

have been studied by Linjama et al (2000a) and the response time is 5–30 ms. The four-way valve is a commercial spool-type pneumatic valve that is suitable also for short-term low-pressure water hydraulic use. Its weak points are leakage and the rather slow response time of 60 ms, which hamper some experiments. Nevertheless, this valve is used because of the poor availability of water hydraulic four-way valves. The system is equipped also with 'hose accumulators' to reduce pressure surges. These small pieces of hose do not reduce the stiffness of the system too much but are effective in reduction of rapid pressure spikes, see Linjama et al (2000b). The supply line has a small accumulator for reducing supply pressure pulsation.

The control hardware consists of a dSPACE DS1104 PowerPC based controller board and the valve power stage is implemented with NAIS AQZ205 PhotoMOS relays. The power circuit is equipped with an RC-filter (R = 100 Ω, C = 1 μF) and can be considered an ideal relay. A rotary encoder with belt transmission measures the piston position, giving the theoretical position resolution of 266060 pulses/m. The supply pressure and pressures in cylinder chambers are measured by 0–100 bar pressure transducers (Trafag).

4 Implementation of Control System

4.1 Valve Flow Capacities

Flow capacities of two-way valves are adjusted with external fixed orifices. Orifices are available with 0.05 mm increments and therefore the exact binary series is impossible to achieve. Flow capacities are measured by opening one valve on the pump and tank side and recording steady-state velocity and pressures. The measurement is repeated in both directions and the mean value is considered the true flow capacity. Figure 7 presents the measured flow capacities of digital flow control units as a function of states u_P and u_T. The flow capacities do not follow exactly the binary series, yielding unequal steps in flow and velocity curves. The flow capacity of the tank side digital flow control unit is smaller than that on pump side because of cavitation choking.

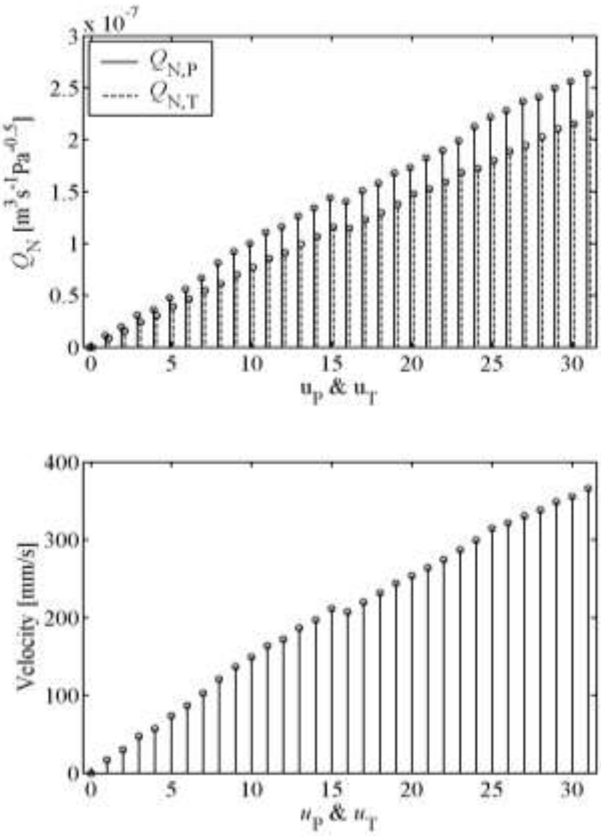

Fig. 7: *Measured flow capacities of digital flow control units and measured steady-state velocity for extending direction when equal openings are used in both digital flow control units*

4.2 Valve Delays and Sampling Time

The opening delay of two-way valves varies between 20 and 30 ms and the closing delay between 5 and 15 ms. This asymmetry is compensated by delaying the closing of two-way valves by 12 ms. By default, the 60 ms delay of the four-way valve is not considered at all, which sometimes causes increased tracking error when the motion starts.

Several experiments were made to determine a suitable sampling time and it was found that 30 ms sampling time gives the best control performance. A shorter sampling time would lead to malfunction of valves, while longer sampling times would increase the tracking error. The selected sampling time does not satisfy assumption 2.e because the system achieves the steady-state velocity after a 30–80 ms transient period. The controller uses a 6 ms base sampling time for delay compensation and data acquisition.

4.3 Test Trajectories and Controller Parameters

The nominal load mass is selected as 100 kg and the nominal position trajectory is a fifth-order polynomial from 150 mm to 350 mm and back with 1.5 s movement time. This gives the maximum velocity of 0.25 m/s and the maximum acceleration of 0.51 m/s^2. In order to minimise the number of tuning parameters, the following cost function is used:

$$\begin{aligned} J(k) &= \left[v_{\mathrm{r}}(k)-\hat{v}(k)\right]^2 + K_{\mathrm{pd}}\left[p_{\mathrm{dr}}(k)-\hat{p}_{\mathrm{d}}(k)\right]^2 \\ &+K_{\Delta\mathrm{p}}\left\{\left[\hat{p}_{\mathrm{A}}(k-1)-\hat{p}_{\mathrm{A}}(k)\right]^2\right. \\ &\left.+\left[\hat{p}_{\mathrm{B}}(k-1)-\hat{p}_{\mathrm{B}}(k)\right]^2\right\} \end{aligned} \tag{8}$$

where $\hat{p}_{\mathrm{d}}$ is the calculated downstream pressure defined by

$$\hat{p}_{\mathrm{d}} = \begin{cases} \hat{p}_{\mathrm{B}} \text{ for } v_{\mathrm{rC}} > 0 \\ \hat{p}_{\mathrm{A}} \text{ for } v_{\mathrm{rC}} < 0 \end{cases} \tag{9}$$

The downstream pressure reference p_{dr} is 10 bar in most tests. So the open-loop controller tries to minimise velocity error and changes in pressure levels and simultaneously to keep downstream pressure near 10 bar.

The supply pressure estimate is a constant 30 bar although the supply pressure varies in the real system. The real supply pressure is about 32 bar with zero velocity and 28–29 bar with full velocity. The load force estimate considers approximately the Coulomb friction of the cylinder as follows:

$$\hat{F} = \begin{cases} 400 \text{ for } v_{\mathrm{rC}} \geq 0.002 \\ 0 \text{ for } -0.002 < v_{\mathrm{rC}} < 0.002 \\ -400 \text{ for } v_{\mathrm{rC}} \leq -0.002 \end{cases} \tag{10}$$

The controller is commanded to close all valves if required velocity is smaller than calculated velocity with $u_{\mathrm{P}} = u_{\mathrm{T}} = 1$.

4.4 Implementation of Control Algorithm

Steady-state pressures and velocities are calculated using vectorised versions of Eq. 3 and 4. For positive direction, the matrices containing steady-state values are given by

$$\begin{aligned} \hat{\boldsymbol{p}}_{\mathbf{A}} &= \frac{\mathbf{z}^2\hat{p}_{\mathrm{S}} + \gamma^2\hat{F}/A_{\mathrm{B}}}{\mathbf{z}^2+\gamma^3} \\ \hat{\boldsymbol{p}}_{\mathbf{B}} &= \frac{\gamma\mathbf{z}^2\hat{p}_{\mathrm{S}} - \mathbf{z}^2\hat{F}/A_{\mathrm{B}}}{\mathbf{z}^2+\gamma^3} \\ \hat{\boldsymbol{v}} &= \frac{\mathrm{diag}\left(\mathbf{Q}_{\mathbf{N,P}}\right)}{A_{\mathrm{B}}} * \sqrt{\frac{\gamma\hat{p}_{\mathrm{S}} - \hat{F}/A_{\mathrm{B}}}{\mathbf{z}^2+\gamma^3}} \\ \mathbf{z} &= \mathbf{Q}_{\mathbf{N,P}} * \left(1/\mathbf{Q}_{\mathbf{N,T}}^{\mathrm{T}}\right) \end{aligned} \tag{11}$$

where diag($Q_{\mathrm{N,P}}$) is a diagonal matrix with elements of the vector $Q_{\mathrm{N,P}}$ in the main diagonal, the star '∗' denotes matrix product and all other operands operate element-wise. The steady-state values for the negative direction are

$$\begin{aligned} \hat{\boldsymbol{p}}_{\mathbf{A}} &= \frac{\gamma^2\hat{p}_{\mathrm{S}} + \gamma^2\hat{F}/A_{\mathrm{B}}}{\mathbf{z}^{-2}+\gamma^3} \\ \hat{\boldsymbol{p}}_{\mathbf{B}} &= \frac{\gamma^3\hat{p}_{\mathrm{S}} - \mathbf{z}^{-2}\hat{F}/A_{\mathrm{B}}}{\mathbf{z}^{-2}+\gamma^3} \\ \hat{\boldsymbol{v}} &= -\sqrt{\frac{\hat{p}_{\mathrm{S}} + \hat{F}/A_{\mathrm{B}}}{\mathbf{z}^{-2}+\gamma^3}} * \frac{\mathrm{diag}\left(\mathbf{Q}_{\mathbf{N,T}}\right)}{A_{\mathrm{B}}} \end{aligned} \tag{12}$$

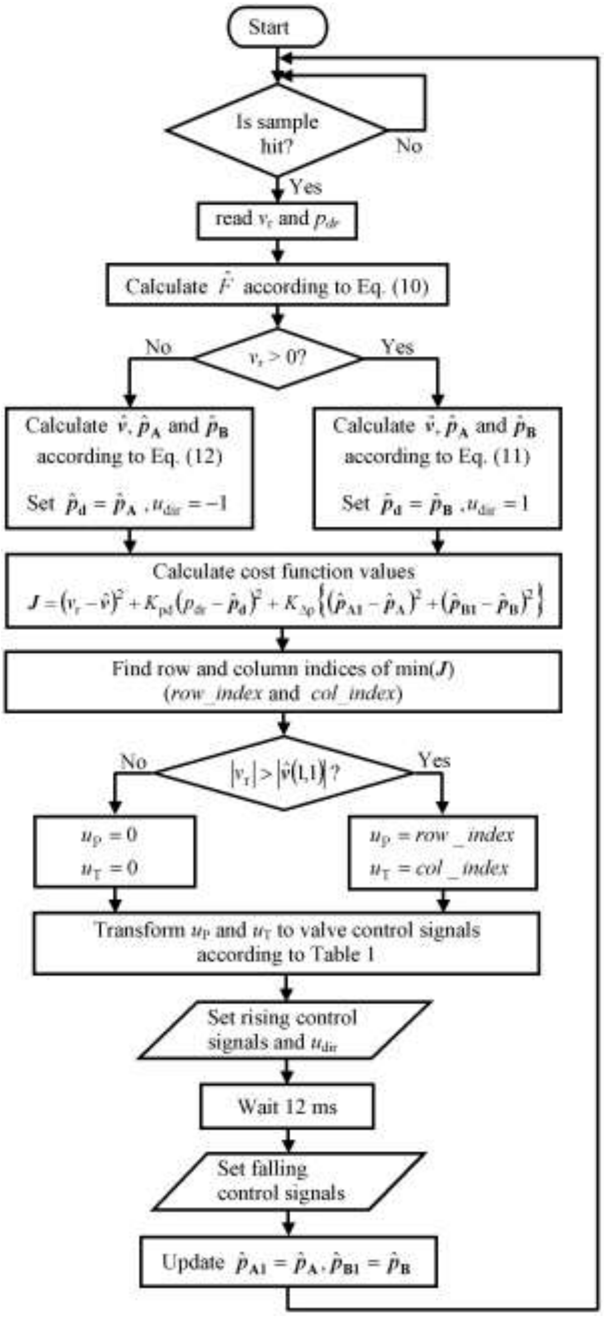

Fig. 8: *The flow chart of the open-loop control algorithm*

The flow chart of the open-loop control algorithm is shown in Fig. 8. The closed-loop controller works similarly but the velocity reference (see Fig. 5) is

$$v_{rC} = v_r + K_P \left(y_r - y \right) \tag{13}$$

i.e., the simplest possible proportional controller is used as a closed-loop controller.

5 Experimental Results

5.1 Open-Loop Responses

Figure 9 shows the measured open-loop response when K_{pd} and $K_{\Delta p}$ are zero. The controller tries to minimise the velocity error and puts no effort into limiting pressure changes. This results in high pressure surges and poor tracking.

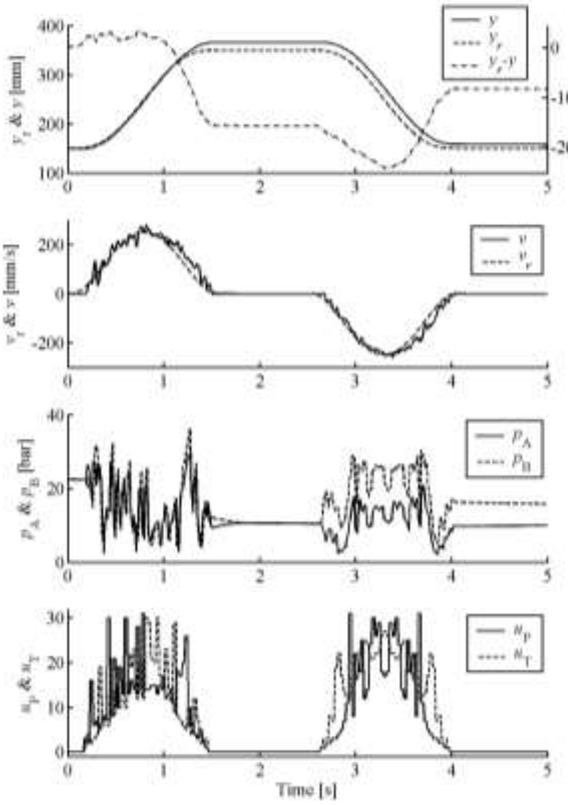

Fig. 9: *Open-loop response with $K_{pd} = K_{\Delta p} = 0$*

Several tests were made with different values of K_{pd} and $K_{\Delta p}$ and it was found that the K_{pd} term is important to keep pressures at suitable level and that the $K_{\Delta p}$ term is not absolutely necessary but helps to reduce pressure surges. Figure 10 shows the open-loop response with $K_{pd} = K_{\Delta p} = 4\times10^{-16}$. Overall tracking behaviour is better and pressure surges are smaller. It can also be seen how the K_{pd} term in the cost function keeps the downstream pressure near 10 bar. The conclusion regarding open-loop tests is that finding a reasonable trade-off between tracking error and pressure behaviour can be done in a straightforward way by adjusting cost function parameters.

5.2 Closed-Loop Responses

The gain of the closed-loop controller is tuned to be about one half of the critical gain of the system and the used gain value is K_P = 15/s. The nominal values for the cost function parameters are selected according to open-loop tests to be $K_{pd} = K_{\Delta p} = 4\times10^{-16}$. Figure 11 shows the nominal closed-loop response. The maximum tracking error is now about 2.5 mm, which can be considered a good value.

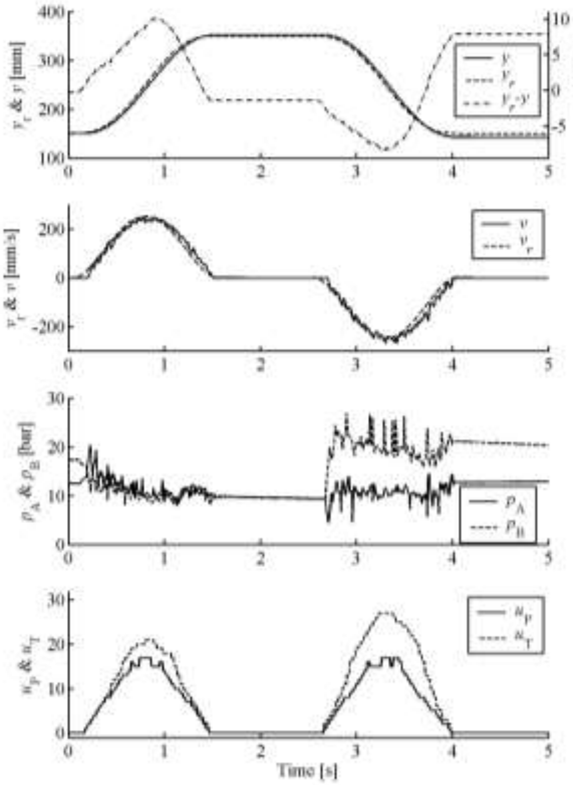

Fig. 10: *Open-loop response with $K_{pd} = K_{\Delta p} = 4\times10^{-16}$*

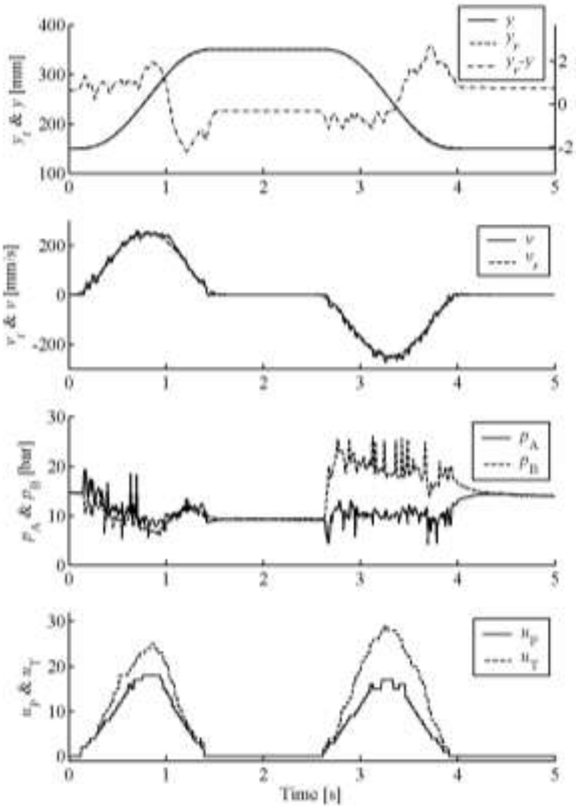

Fig. 11: *Nominal closed-loop response, K_P=15/s, $K_{pd} = K_{\Delta p} = 4\times10^{-16}$*

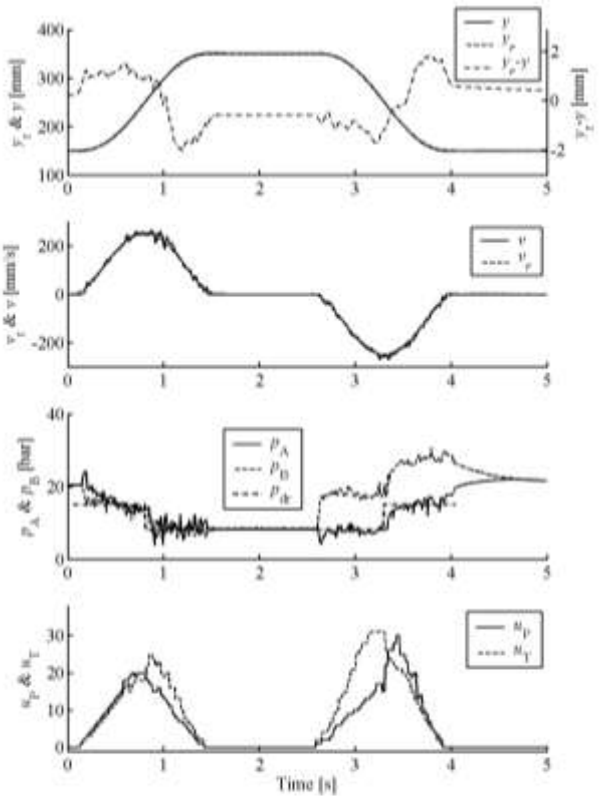

Fig. 12: *Closed-loop response with varying downstream pressure reference, K_P=15/s, K_{pd} = 4×10^{-16}, $K_{\Delta p}$=1×10^{-16}*

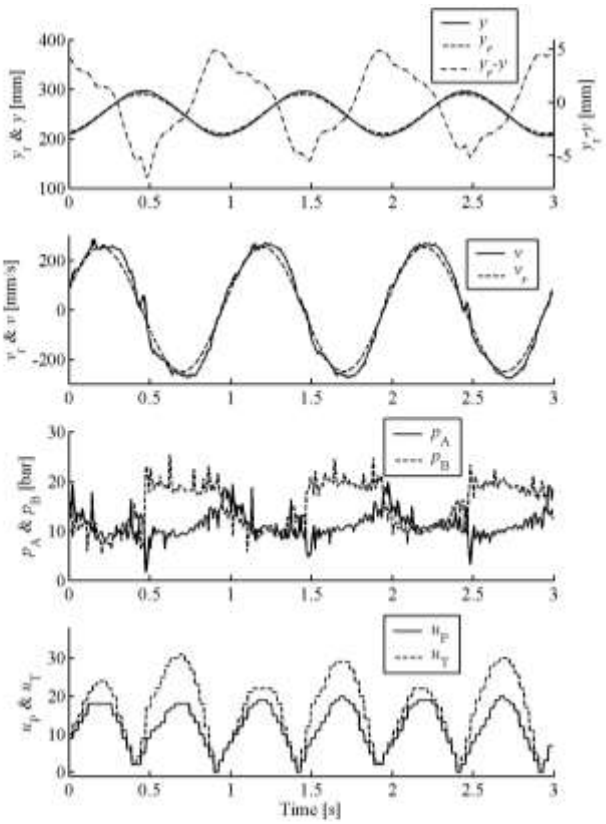

Fig. 13: *1 Hz closed-loop sinusoidal response, K_P=15/s, K_{pd}=$K_{\Delta p}$=4×10^{-16}*

Figure 12 shows the closed-loop response when the downstream pressure reference changes during motion. The downstream pressure reference is 15 bar if y < 0.25 m and 8 bar otherwise. Now the $K_{\Delta p}$ parameter is reduced to 1×10^{-16} in order to achieve faster pressure tracking. It can be seen that almost independent pressure and velocity control is possible with the suggested control scheme. There are some pressure transients when the pressure level changes, but they have only a minor effect on position tracking.

Finally, Fig. 13 presents a 1 Hz sinusoidal response. To obtain this kind of response is difficult or even impossible without a “trick” because of the too slow four-way valve. The trick used is to command the four-way valve to change its state 60 ms *before* the velocity reference changes its sign. Of course, this kind of predictive control cannot be used in the general case and faster four-way valve is required for real applications. It is seen that the phase shift is small but there are some problems when the velocity changes its sign.

6 Conclusions

Experimental results show that accurate position tracking control of water hydraulic systems is possible with simple, non-ideal and low-cost on/off valves. The achieved position tracking performance is 2.5 mm tracking error with 250 mm/s peak velocity. This is of the same level as results of Mäkinen and Virvalo (2001) and clearly better than results of Cho et al (2002). However, the natural frequency of the test system is about 105 rad/s, being 17 percent higher than that in the system of Mäkinen and Virvalo (2001). Thus, it can be concluded that achieved tracking performance is slightly weaker than in Mäkinen and Virvalo (2001). This is still a significant result because it is achieved with inexpensive and non-ideal on/off valves using pure position feedback.

If the smoothness of movement is considered, the results are not as good as with good servo valves. The smoothness is “reasonable” and sufficient for many applications but it can be improved. The operation principle of the on/off system presented in the paper always generates force pulsation. The effective valve opening changes stepwise and a force pulse is generated to accelerate or decelerate the load to the new velocity level. The magnitude of this force pulse depends on the load mass, system dynamics and the magnitude of velocity change. However, a much more important source of pressure peaks is inexact switching of valves, which is caused by the varying opening and closing delays. For example, the change of the state of the digital flow control unit from 15 to 16 requires simultaneous opening of the fifth valve and closing of all other valves. If the opening of the fifth valve is late, the opening of the digital flow control unit is temporarily zero, yielding a high pressure peak. This kind of peak occurs occasionally and can be seen for example in Fig. 11. Smaller variation in valve delays would improve the smoothness of motion.

An important advantage of the system presented is its ability to control simultaneously pressure level and velocity of the actuator. This feature was demonstrated in this paper but it has not been utilised yet. The pressure level control makes it possible to handle overrunning loads without cavitation or the use of counterbalance valves. The optimisation of pressure level allows the use of lower supply pressure and improves efficiency. This is an obvious and important topic for future research. Another topic for future research is theoretical analysis. The system differs considerably from traditional hydraulic control systems and traditional control design methods are difficult to apply. It is clear from the results presented in the previous sections that the concept works, but it is unclear how to analyse e.g. the stability or robustness of the system.

Nomenclature

A_A	Piston area, piston side	[m^2]
A_B	Piston area, piston rod side	[m^2]
F	Load force	[N]
$\hat{F}$	Load force estimate	[N]
J	Cost function	[–]
K_i	Cost function weight factors, i = 1…4	[–]
K_P	Gain of closed-loop controller	[1/s]
K_{pd}	Weight for downstream pressure error	[–]
$K_{\Delta p}$	Weight for pressure variation	[–]
m	Load mass	[kg]
n_P	Number of pump side valves	[–]
n_T	Number of tank side valves	[–]
$Q_{N,P}$	Nominal flow of pump side digital flow control unit	[m^3/(s√Pa)]
$\mathbf{Q}_{N,P}$	Vector of all nonzero pump side nominal flows	[m^3/(s√Pa)]
$Q_{N,P}(u_P)$	u_P-th element of vector $\mathbf{Q}_{N,P}$	[m^3/(s√Pa)]
$Q_{N,Pi}$	Nominal flow of i-th pump side valve, i = 1…5	[m^3/(s√Pa)]
$Q_{N,T}$	Nominal flow of tank side digital flow control unit	[m^3/(s√Pa)]
$\mathbf{Q}_{N,T}$	Vector of all nonzero tank side nominal flows	[m^3/(s√Pa)]
$Q_{N,T}(u_T)$	u_T-th element of vector $\mathbf{Q}_{N,T}$	[m^3/(s√Pa)]
$Q_{N,Ti}$	Nominal flow of i-th tank side valve, i = 1…5	[m^3/(s√Pa)]
Q_P	Flow of pump side digital flow control unit	[m^3/s]
Q_T	Flow of tank side digital flow control unit	[m^3/s]
p_A	Pressure in A-chamber	[Pa]
$\hat{p}_A$	Calculated pressure in A-chamber	[Pa]
$\hat{\mathbf{p}}_\mathbf{A}$	Matrix of all calculated pressures in A-chamber	[Pa]
p_{Ar}	Reference for A-chamber pressure	[Pa]
p_B	Pressure in B-chamber	[Pa]
$\hat{p}_B$	Calculated pressure in B-chamber	[Pa]
$\hat{\mathbf{p}}_\mathbf{B}$	Matrix of all calculated pressures in B-chamber	[Pa]
p_{Br}	Reference for B-chamber pressure	[Pa]
$\hat{p}_d$	Calculated downstream pressure	[Pa]
$\hat{\mathbf{p}}_\mathbf{d}$	Matrix of all calculated downstream pressures	[Pa]
p_{dr}	Downstream pressure reference	[Pa]
p_S	Supply pressure	[Pa]
$\hat{p}_S$	Supply pressure estimate	[Pa]
u_{dir}	Control signal of 4/2 directional valve	[–]
u_P	State of pump side digital flow control unit	[–]
u_{Pi}	Control signal of i-th pump side valve, i = 1…5	[–]
u_T	State of tank side digital flow control unit	[–]
u_{Ti}	Control signal of i-th tank side valve, i = 1…5	[–]
v	Piston velocity	[m/s]
$\hat{v}$	Calculated piston velocity	[m/s]
$\hat{\mathbf{v}}$	Matrix of all calculated piston velocities	[m/s]
v_r	Piston velocity reference	[m/s]
v_{rC}	Closed-loop velocity reference	[m/s]
y	Piston position	[m]
y_r	Piston position reference	[m]
z	Ratio of pump and tank side nominal flows	[–]
$\mathbf{z}$	Matrix of all ratios of nominal flows	[–]
γ	Ratio of piston areas	[–]

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