IMPROVED DIGITAL HYDRAULIC TRACKING CONTROL OF WATER HYDRAULIC CYLINDER DRIVE

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Abstract

A position tracking control system is implemented by utilizing parallel-connected on/off valve series. The pulse code modulation method is used to achieve stepwise flow control and four valve series, each having four two-way solenoid valves, are used. A cost function based controller is used to control simultaneously and independently flow paths from supply to cylinder chambers and from chambers to tank. It is shown that controllability can be improved especially at low velocities by allowing three or four valve series to be open simultaneously instead of using classical inflow-outflow control.

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 et al (1999), Mäkinen and Virvalo (2001), Sanada (2002), Cho et al (2002), Sairiala et al (2003)). The main obstacle to 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, and some papers dealing with water hydraulic tracking control systems have been published in recent last years. Mäkinen et al (1999) studied the effect of reference trajectory on the dynamic behaviour of a water hydraulic servo system. The aim was not to achieve very good tracking performance but to obtain general information about the tracking behaviour of water hydraulic servos with different reference trajectories and velocities. Therefore, a simple proportional position controller was used which yielded moderate tracking performance. An important contribution of this paper was that very slow motions were also studied. A smooth tracking control was demonstrated at velocities only a few percent from the maximum velocity. Later, Mäkinen and Virvalo (2001) introduced a combination of position and velocity controllers which yielded much improved tracking performance. The state controller approach with position, velocity and acceleration feedback was used. 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 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. Sairiala et al (2003) used a proportional controller together with velocity feedforward and compensation for nonlinear characteristics of the valve. The system was the same as in Cho et al (2002), but the load mass was 200 kg instead of 100 kg. Results showed 2.8 mm tracking error with 310 mm/s peak velocity. This is a noteworthy result because of the simple controller structure and the use of position feedback only.

On/off control is a promising alternative for proportional control because of the low-cost and robust valves. The biggest challenge is how to achieve good controllability with rather the slow response valves available. One of the most promising approaches is

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Pulse Code Modulation (PCM), which utilizes parallel-connected valves. If valve sizes are selected according to binary series (1:2:4:8…), it is possible to achieve 2^n discrete flow rates with n parallel-connected valves analogously to a DA-converter. Such a valve series is called here a Digital Flow Control Unit (DFCU). PCM is an old invention (Bower, 1961) and it has been utilised in pneumatic (Miyata et al (1991)), oil hydraulic (Virvalo (1978)) and water hydraulic systems (Laamanen et al (2002)). Recently, Linjama et al (2003a) presented a water hydraulic trajectory control system based on PCM and demonstrated that good tracking control is possible even with simple on/off valves. Both the inflow and outflow were controlled independently by five parallel-connected two-way solenoid valves, as shown in Fig. 1 (a). The achieved tracking performance was at the same level as that of Mäkinen and Virvalo (2001) or Sairiala et al (2003). A cost function based controller was used, which allowed simultaneous control of velocity and pressure.

The control system of Linjama et al (2003a) was further analyzed in Linjama et al (2003b), where it was concluded that it has potential for low-cost, reliable and high performance fluid power technology. Essential features of this *Digital Hydraulics* technology are:

- Separate meter-in and separate meter-out control using parallel-connected on/off valve series
- No need for continuous or high-frequency switching of valves
- Intelligent controller, e.g.
  - Adaptation for different actuators and loads
  - Online pressure control and minimization of power losses
- Redundancy because of parallel-connected valves: The system can work even if some valves are out of order

Linjama and Vilenius (2004) applied the hydraulic circuit of Fig. 1 (a) and the cost function based control approach to an oil hydraulic mobile machine mockup. Six valves were used in both DFCUs and the simultaneous controllability of velocity and pressure was analysed. They concluded that controllability is good at high velocities but poor at low velocities. The theoretical ratio of maximum and minimum velocity is 2^n-1 for n parallel-connected valves but the situation is much worse if simultaneous pressure and velocity control are required. This means that a large number of valves are needed. Another problem with the hydraulic circuit of Fig. 1 (a) is that a large four-way valve is needed for changing direction of movement.

This paper concentrates on the small velocity control problem. Four digital flow control units are used to control independently flow paths P→A, A→T, P→B and B→T, as shown in Fig. 1 (b). This eliminates the need for a separate four-way valve and improves redundancy because the failure of a single valve can never jam the system. The control improvement is attempted by allowing three or four DFCUs to open simultaneously when necessary. This is somehow analogous to an underlapped valve but is more general because all four flow paths can be controlled independently. The approach can also be seen as a generalization of the controller of Linjama et al (2002), where simultaneous opening of three flow paths was studied in the context of a simple bang-bang system. A related work on the pneumatic side has been presented by Zachrison and Sethson (2003). They controlled a pneumatic cylinder using four two-way on/off valves. All 16 opening combinations of valves were simulated in real-time and the best combination was selected by minimizing the given cost function. However, their system was a classical on/off system without any parallel-connected valves.

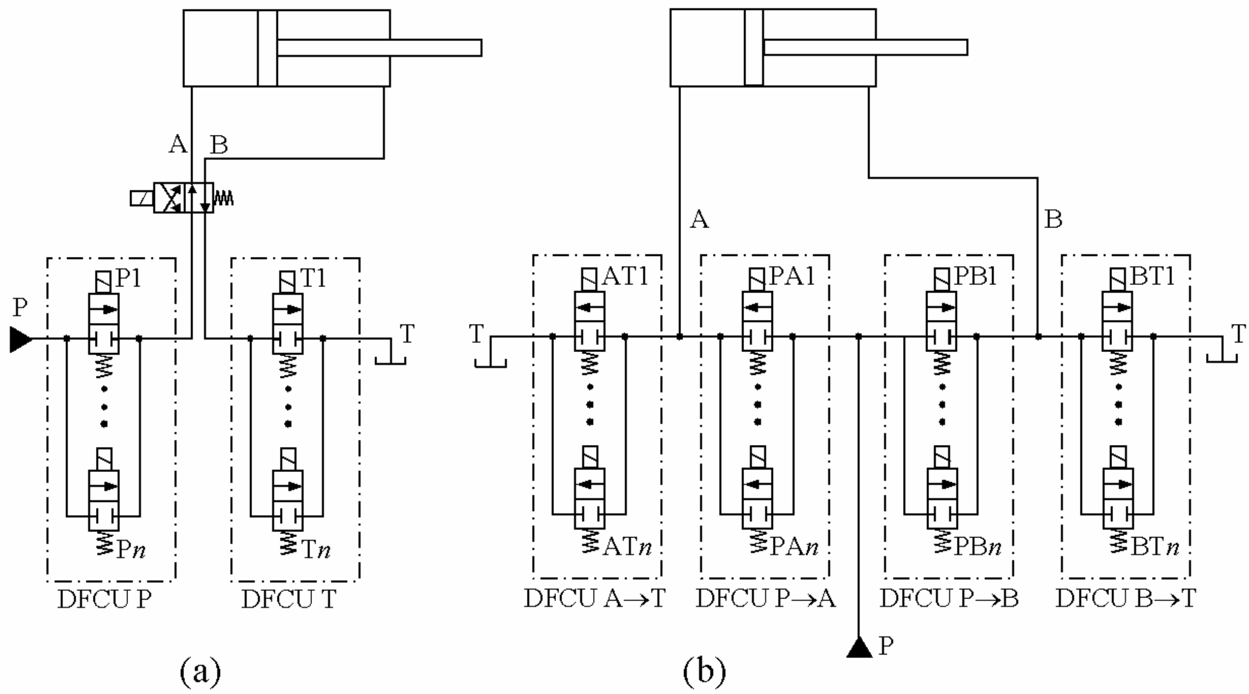

Fig. 1: *PCM control of cylinder with two (a) and four (b) digital flow control units*

2 Digital Hydraulic Control Approach

2.1 Definitions

The hydraulic circuit of the suggested system is shown in Fig. 1 (b). Four DFCUs are used to control flow paths P→A, A→T, P→B and B→T. The individual valves are denoted by subscripts PA*i*, AT*i*, PB*i* and BT*i*, respectively, where *i* is the ordinal number of the valve. In order to simplify the notation, it is assumed that each DFCU has the same number of valves, denoted by *n*. Let us consider DFCU P→A in more detail. The flow rate of an ideal valve is assumed to follow the equation of turbulent flow, i.e.

$$Q_{\mathrm{PAi}} = Q_{\mathrm{N,PAi}} u_{\mathrm{PAi}} \sqrt[*]{p_{\mathrm{S}} - p_{\mathrm{A}}} \tag{1}$$

where $\sqrt[*]{\bullet}$ is a shortcut for signed square root $\mathrm{sgn}(\bullet)\sqrt{|\bullet|}$, $Q_{\mathrm{N,PAi}}$ is flow coefficient of the *i*-th valve of DFCU P→A and u_{PAi} is the valve control signal (0 or 1). Let us define $n \times 2^n$ binary matrix **B**:

$$\mathbf{B} = \begin{bmatrix} 0 & 1 & 0 & 1 & 0 & 1 & 0 & 1 & \cdots & 1 \\ 0 & 0 & 1 & 1 & 0 & 0 & 1 & 1 & \cdots & 1 \\ 0 & 0 & 0 & 0 & 1 & 1 & 1 & 1 & \cdots & 1 \\ \vdots & \vdots & \vdots & \vdots & \vdots & \vdots & \vdots & \vdots & & \vdots \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \cdots & 1 \end{bmatrix} \tag{2}$$

$$\textit{State } u_{\mathrm{PA}}: \; 0 \quad 1 \quad 2 \quad 3 \quad 4 \quad 5 \quad 6 \quad 7 \quad \cdots \quad 2^n - 1$$

Each column of the matrix defines one possible *state* of the DFCU and the *i*-th column of **B** is equal to state *i*–1, represented as *n*-bit binary number. The flow coefficient of the whole DFCU is the sum of flow coefficients of opened valves and can be expressed as a function of state u_{PA} as follows:

$$Q_{\mathrm{N,PA}}\left(u_{\mathrm{PA}}\right) = \boldsymbol{b}^T_{u_{\mathrm{PA}}+1} \begin{bmatrix} Q_{\mathrm{N,PA1}} \\ Q_{\mathrm{N,PA2}} \\ \vdots \\ Q_{\mathrm{N,PAn}} \end{bmatrix} \tag{3}$$

where $\boldsymbol{b}_{\mathrm{i}}$ is the *i*-th column of matrix **B**.

2.2 Steady-State Velocity and Pressures

The following development is presented only for extending direction of movement, but retracting direction can be handled similarly. Referring to Fig. 1 (b), the steady-state equations of the system are

$$\begin{aligned} & Q_{\mathrm{N,PA}} \sqrt[*]{p_{\mathrm{S}} - p_{\mathrm{A}}} - Q_{\mathrm{N,AT}} \sqrt[*]{p_{\mathrm{A}} - p_{\mathrm{T}}} = A_{\mathrm{A}} v \\ & Q_{\mathrm{N,PB}} \sqrt[*]{p_{\mathrm{S}} - p_{\mathrm{B}}} - Q_{\mathrm{N,BT}} \sqrt[*]{p_{\mathrm{B}} - p_{\mathrm{T}}} = -A_{\mathrm{B}} v \\ & F = A_{\mathrm{A}} p_{\mathrm{A}} - A_{\mathrm{B}} p_{\mathrm{B}} \end{aligned} \tag{4}$$

The analytical solution of Eq. 4 requires the solution of the fourth-order polynomial and is too complicated for practical use. The special case $u_{\mathrm{AT}}=u_{\mathrm{PB}}=0$, $u_{\mathrm{PA}}>0$, $u_{\mathrm{BT}}>0$, $p_{\mathrm{A}}<p_{\mathrm{S}}$, $p_{\mathrm{B}}>p_{\mathrm{T}}$ can be solved analytically and the result is

$$\begin{aligned} p_{\mathrm{A}} &= \frac{\kappa^2 p_{\mathrm{S}} + \gamma^2 p_{\mathrm{T}} + \gamma^2 F / A_{\mathrm{B}}}{\kappa^2 + \gamma^3} \\ p_{\mathrm{B}} &= \gamma p_A - F / A_B \\ v &= \frac{Q_{\mathrm{N,PA}}}{A_{\mathrm{B}}} \sqrt{\frac{\gamma\, p_{\mathrm{S}} - p_{\mathrm{T}} - F / A_{\mathrm{B}}}{\kappa^2 + \gamma^3}} \\ \gamma &= A_{\mathrm{A}} / A_{\mathrm{B}} \\ \kappa &= Q_{\mathrm{N,PA}} / Q_{\mathrm{N,BT}} \end{aligned} \tag{5}$$

Equation 5 defines steady-state pressures and velocity in extending direction when DFCUs P→A and B→T are used alone. These $(2^n\text{-}1)^2$ different state combinations are called *standard state combinations* hereafter. It is important to note that estimates for supply and tank pressures as well as load force are required in the calculation of steady-state values.

If three or four DFCUs are open simultaneously, Eq. 4 can be solved numerically by using e.g. Newton-Raphson iterations. These state combinations are called *general state combinations*. The Newton-Raphson solution process is

$$\begin{aligned} & \boldsymbol{x} = \boldsymbol{x}_0 \\ & \text{for } k \text{ from } 1 \text{ to } n_{\mathrm{iter}} \\ & \quad \boldsymbol{x} = -\mathbf{J}^{-1}\boldsymbol{F} + \boldsymbol{x} \\ & \text{end} \end{aligned} \tag{6a}$$

where

$$\begin{aligned} \boldsymbol{x} &= \begin{bmatrix} v & p_{\mathrm{A}} & p_{\mathrm{B}} \end{bmatrix}^T \\ \boldsymbol{F} &= \begin{bmatrix} Q_{\mathrm{N,PA}} \sqrt[*]{p_{\mathrm{S}} - p_{\mathrm{A}}} - Q_{\mathrm{N,AT}} \sqrt[*]{p_{\mathrm{A}} - p_{\mathrm{T}}} - A_{\mathrm{A}} v \\ Q_{\mathrm{N,PB}} \sqrt[*]{p_{\mathrm{S}} - p_{\mathrm{B}}} - Q_{\mathrm{N,BT}} \sqrt[*]{p_{\mathrm{B}} - p_{\mathrm{T}}} + A_{\mathrm{B}} v \\ A_{\mathrm{A}} p_{\mathrm{A}} - A_{\mathrm{B}} p_{\mathrm{B}} - F \end{bmatrix} \\ \mathbf{J} &= \begin{bmatrix} -A_{\mathrm{A}} & \mathbf{J}_{1,2} & 0 \\ A_{\mathrm{B}} & 0 & \mathbf{J}_{2,3} \\ 0 & A_{\mathrm{A}} & -A_{\mathrm{B}} \end{bmatrix} \\ \mathbf{J}_{1,2} &= -\frac{Q_{\mathrm{N,PA}}}{2\sqrt{|p_{\mathrm{S}} - p_{\mathrm{A}}|}} - \frac{Q_{\mathrm{N,AT}}}{2\sqrt{|p_{\mathrm{A}} - p_{\mathrm{T}}|}} \\ \mathbf{J}_{2,3} &= -\frac{Q_{\mathrm{N,PB}}}{2\sqrt{|p_{\mathrm{S}} - p_{\mathrm{B}}|}} - \frac{Q_{\mathrm{N,BT}}}{2\sqrt{|p_{\mathrm{B}} - p_{\mathrm{T}}|}} \end{aligned} \tag{6b}$$

Comparing Eq. 5 and 6, it can be concluded that the solution of a general state combination is computationally much more demanding than the solution of a standard state combination. The iteration of Eq. 6 can also fail e.g. because of unbounded or singular Jacobian matrix **J**.

2.3 Controllability of velocity and pressure

Consider a cylinder drive of Fig. 1 (b) with the following parameters: cylinder diameter 32 mm, rod diameter 16 mm, p_{S} = 3 MPa and F = 0 N. Identical DFCUs with four valves are assumed and flow coefficients of valves are $[1\ 2\ 4\ 8] \times 10^{-8}$ $\mathrm{m^3/(s\sqrt{Pa})}$.

The standard state combinations are analysed first. DFCUs P→A and B→T have fifteen nonzero state combinations and the total number of combinations is

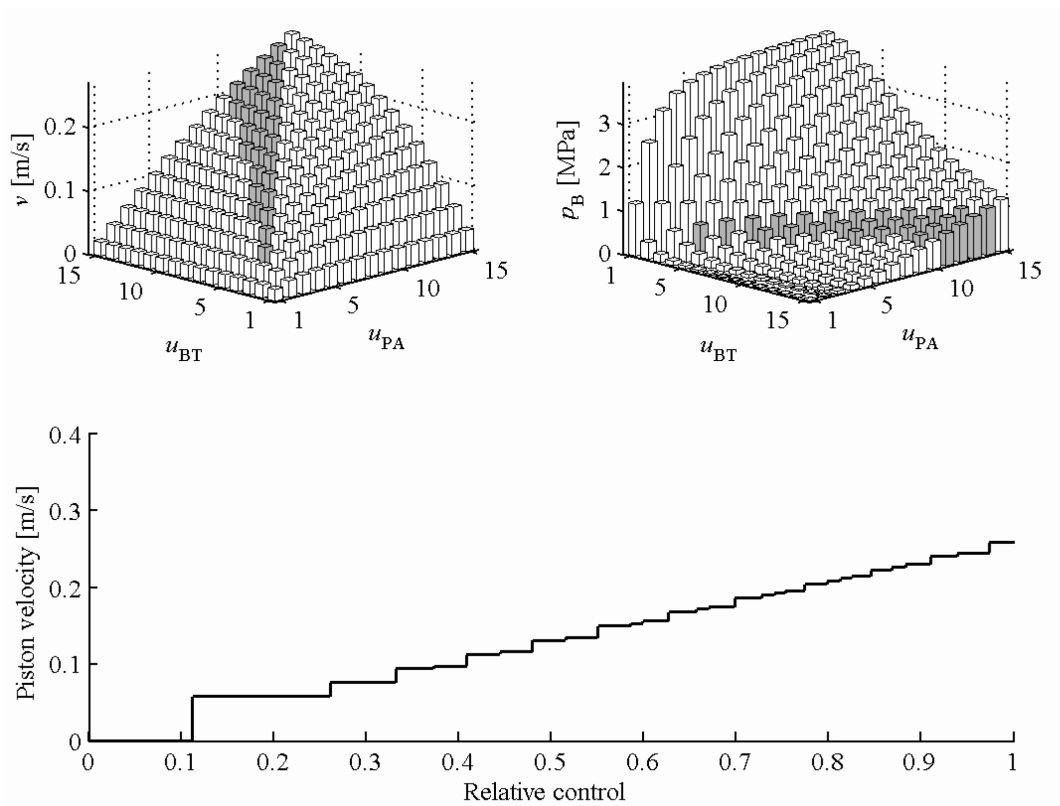

Fig. 2: *Controllability of velocity if steady-state pressure of B-chamber is required to lie between 0.7 and 1.1 MPa. Only standard state combinations are considered and allowed combinations are shown in grey*

$15^2 = 225$. Assume now that the steady-state pressure in B-chamber is required to be near 0.9 MPa, say 0.9±0.2 MPa. In total, 28 state combinations can satisfy this requirement and Fig. 2 depicts controllability of velocity for this situation. The first plot shows steady-state velocity as a function of states and the second plot shows steady-state pressure at B-chamber. The combinations, which can satisfy the pressure requirement, are shown in grey. It is seen in the third plot that controllability of velocity is poor at small velocities and improves with increasing velocity.

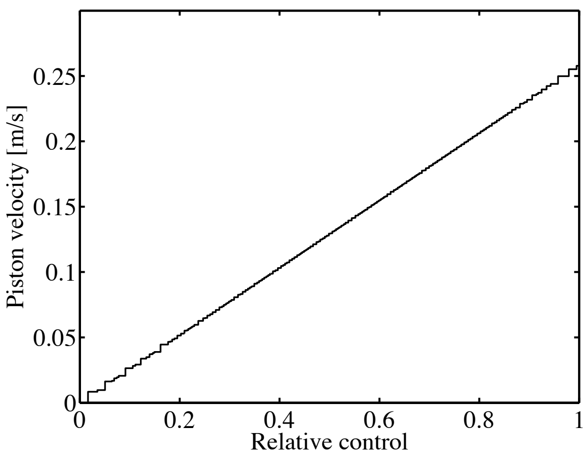

Fig. 3: *Controllability of velocity if steady-state pressure of B-chamber is required to lie between 0.7 and 1.1 MPa. Some general state combinations are considered such that the u_{PA} & u_{BT} are between 1 and 15, and u_{AT} & u_{PB} are between 0 and 2*

Controllability can be improved by utilizing general state combinations. The system has $15^4+4\times15^3 = 64125$ general state combinations (three or four DFCUs are open simultaneously) and about half of them give extending movement. The solution of general state combinations is computationally expensive and it is not feasible to consider all of them. Figure 3 depicts the special case in which states 1 to 15 are considered for DFCUs P→A & B→T and states 0, 1 and 2 are considered for DFCUs A→T & P→B. The total number of analysed state combinations is 2025, and 276 combinations can satisfy the pressure requirement. Figure 3 demonstrates clearly improved controllability. Therefore, the utilization of general state combinations has potential for improving controllability of digital hydraulic servos especially at low velocities.

2.4 Cost Function Based Control

The control approach is similar to that in earlier publications (Linjama et al (2003a), Linjama et al (2004)). A cost function is defined and the states of DFCUs are selected at each sampling instant such that the cost function is minimized. The cost function is based on calculated steady-state velocity and pressures (Eq. 5 or 6) and a modified version of the cost function of Linjama et al (2003a) is used:

$$J = (v_r - v)^2 + K_{pd}(p_{dr} - p_d)^2 + K_{\Delta u} n_{sw} \tag{7}$$

where v is calculated steady-state velocity, v_r velocity reference, p_d calculated downstream pressure (equal to p_B for extending movement and p_A for retracting movement), p_{dr} downstream pressure reference and n_{sw}

the number of valve switchings required for transition from the current state to the new state. The controller tries to minimise velocity error, downstream pressure error and the number of valve switchings simultaneously and the tuning parameters K_{pd} and $K_{\Delta u}$ are used to find a compromise between velocity and pressure tracking as well as activity of valves. The calculation of steady-state velocity and pressures requires estimates for load force as well as supply and tank pressure. The pressure measurements are assumed and measured supply and tank pressures can be fed to the controller after low-pass filtering. The load force estimate is achieved by low-pass filtering the calculated force $p_A A_A$-$p_B A_B$. The control approach is open-loop – states of DFCUs are selected such that the error between the target and *calculated steady-state* velocity/pressure is minimised. Dynamic effects and disturbances cause errors and therefore a closed-loop position controller is used to correct velocity reference according to measured position error. The block diagram of the control system is shown in Fig. 4.

2.5 Finding Minimum of Cost Function

The most difficult task in the controller of Fig. 4 is the finding of the minimum of the cost function. Earlier studies (Linjama et al (2003a), Linjama et al (2004)) have considered standard state combinations only, and the minimum has been found by calculating cost function values for all combinations at each sampling event. This approach cannot be used here because general state combinations are included. The search space increases from $(2^n\text{-}1)^2$ to more than $(2^n\text{-}1)^4$ and the solution of general state combinations is computationally demanding. Therefore, the real-time calculation of all combinations is too demanding for existing control hardware. Standard optimisation techniques are difficult to apply because the cost function has many local minima. Genetic algorithms might be one feasible solution but a heuristic search algorithm is used in this paper. All standard state combinations are calculated first and the found suboptimum is used as a starting point for seeking a better solution among the general state combinations. The search algorithm can be described for the extending direction of movement as follows:

1. Read inputs (measured supply and tank pressure, filtered load force estimate $\hat{F}$, closed-loop velocity reference v_{rC} and downstream pressure reference p_{dr})
2. Calculate cost function J for $u_{PA} = 1 \ldots 2^n-1$, $u_{AT} = 0$, $u_{PB} = 0$, $u_{BT} = 1 \ldots 2^n-1$ (i.e. all standard state combinations)
3. Find $\min(J) \hat{=} M_1$ and corresponding $u_{PA} \hat{=} u_{PA,subopt}$ and $u_{BT} \hat{=} u_{BT,subopt}$
4. Calculate cost function J for $u_{PA} = u_{PA,subopt} \ldots u_{PA,subopt}+n_1$, $u_{AT} = 0 \ldots n_2$, $u_{PB} = 0 \ldots n_2$, $u_{BT} = u_{BT,subopt} \ldots u_{BT,subopt}+n_1$, excluding standard state combinations
5. Find $\min(J) \hat{=} M_2$ and corresponding $u_{PA} \hat{=} u_{PA,opt}$, $u_{AT} \hat{=} u_{AT,opt}$, $u_{PB} \hat{=} u_{PB,opt}$, $u_{BT} \hat{=} u_{BT,opt}$
6. If $M_1 \leq M_2$, set $u_{PA} = u_{PA,subopt}$, $u_{AT} = 0$, $u_{PB} = 0$, $u_{BT} = u_{BT,subopt}$ else set $u_{PA} = u_{PA,opt}$, $u_{AT} = u_{AT,opt}$, $u_{PB} = u_{PB,opt}$, $u_{BT} = u_{BT,opt}$

The steady-state velocity and pressures, which are needed in the calculation of cost function, are calculated using Eq. 5 in step 2 and Eq. 6 in step 4. Parameters n_1 and n_2 determine the search space in the neighbourhood of the best standard state combination. The above procedure does not guarantee that the global minimum is found. However, the selected search space in step 4 can be motivated by the fact that velocity decreases if DFCU A→T and/or DFCU P→B are opened. Thus, it is unlikely that any improvement is achieved by decreasing u_{PA} or u_{BT} in step 4. Also, from the energy consumption point of view, it is desirable to open DFCUs A→T and P→B as little as possible.

3 Test System

The hydraulic circuit of the test system is depicted in Fig. 5. It consists of a multistage centrifugal pump unit, four digital flow control units and an asymmetric cylinder with inertial load. Each flow control unit has

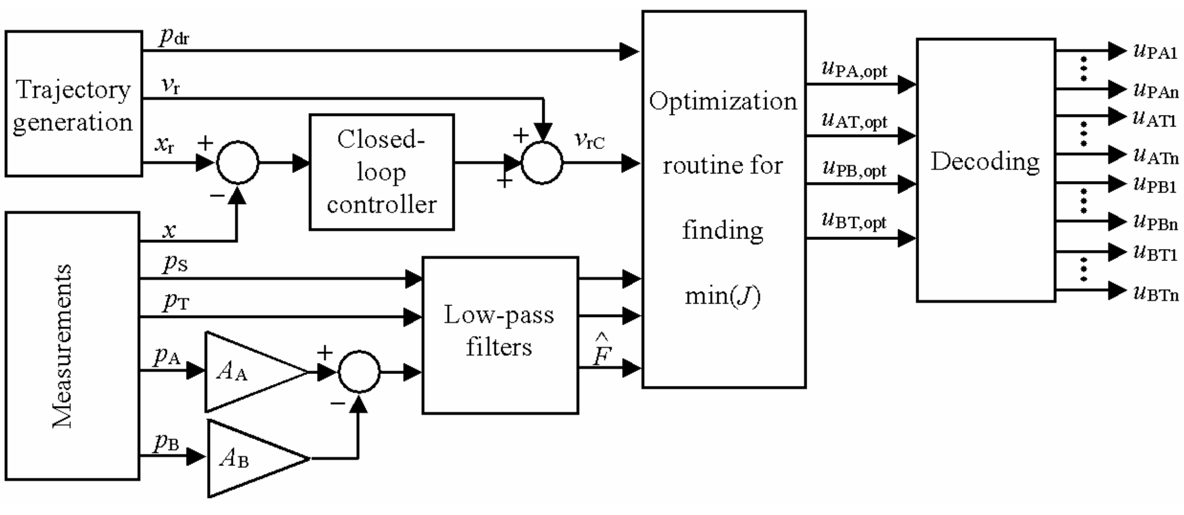

Fig. 4: *Block diagram of the controller used*

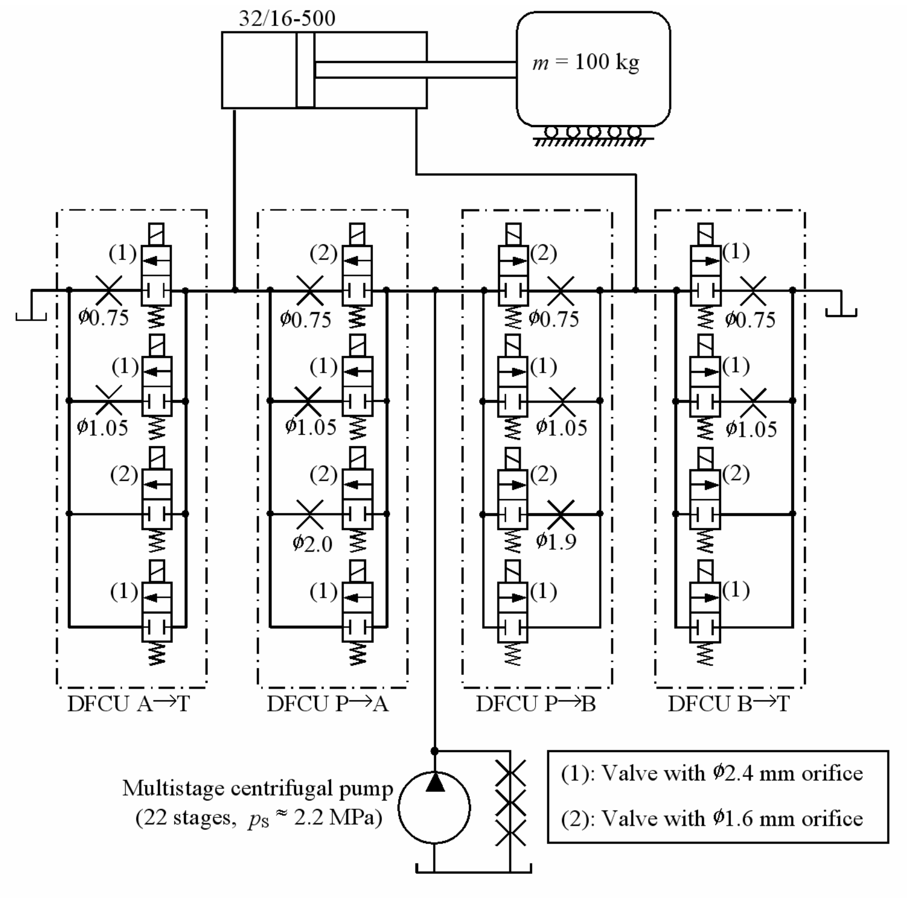

Fig. 5: *The hydraulic circuit diagram of the test system*

four directly operated solenoid valves. Two types of valves are used: the larger type has 2.4 mm internal orifice and the smaller type has 1.6 mm orifice. Flow capacities are adjusted with external fixed orifices, and orifice sizes as well as valve types are shown in Fig. 5. These valves have been studied by Linjama et al (2000) and the response time is 5–30 ms with AC current. The valve package with a homemade manifold is presented in Fig. 6. The manifold is designed for 4×5 valves and the fifth positions are plugged.

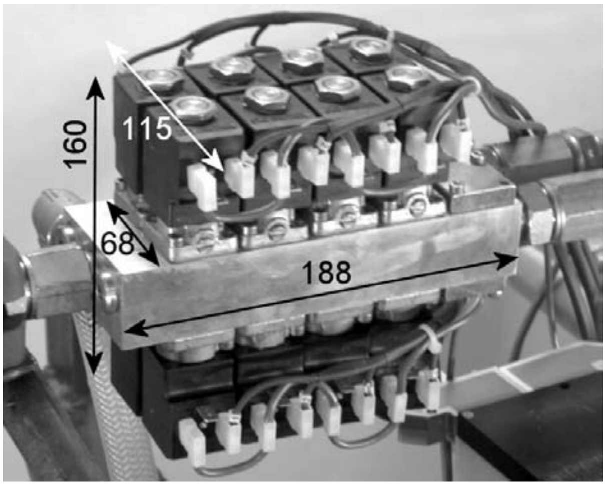

Fig. 6: *Valve package used. The manifold is designed for 4×5 valves*

The control hardware consists of a dSPACE DS1006 controller board with 2.2 GHz 64-bit AMD Opteron main processor. The valve control electronics has been implemented by a microcontroller (Micro-Chip PIC 18F458) and low-side MOSFETs (2SK2201). Transient suppressor diodes (1.5KE68CA, 68 V breakdown voltage) are connected in parallel with MOSFETs to protect the system from voltage peaks. Although the valves have 24 V AC-coils, a 15 V DC is used to reduce variation in delays. A rotary encoder with belt transmission measures the piston position, giving the theoretical position resolution of 266060 pulses/m.

4 Implementation of Control System

4.1 Valve Flow Capacities

Flow capacities of valves are measured by opening one valve on the pump and tank side DFCU and recording steady-state velocity and pressures. Measured flow capacities are given in Table 1. Values are presented as standard K_v-values, i.e. flow rate in l/min at 0.1 MPa pressure differential. The flow capacities do not follow exactly the binary series and the flow capacities of the tank side digital flow control units are smaller than those on the pump side because of cavitation choking. These facts do not decrease the control performance, because the true flow capacities are coded into the controller.

Table 1: Flow rates (in l/min) of valves at 0.1 MPa pressure differential

	DFCU P→A	DFCU A→T	DFCU P→B	DFCU B→T
Valve 1	0.33	0.30	0.33	0.29
Valve 2	0.65	0.57	0.66	0.54
Valve 3	1.36	1.16	1.35	1.09
Valve 4	2.40	2.06	2.32	2.18

4.2 Valve Delays and Sampling Time

The valve delays vary between 8 and 14 ms with the control electronics used. When compared to the earlier study (Linjama et al (2003a)), the delays are considerably smaller due to DC voltage and improved control electronics. The variation in delays is so small that there is no need for delay compensation as in previous studies. The sampling time of the controller is selected to be 20 ms, and a faster 2 ms sampling time is used for data acquisition.

4.3 Test Trajectories

The nominal load mass is selected as 100 kg and a fifth-order polynomial is used as a position reference. The movement time is 1.5 s and the initial piston position is 150 mm. Three strokes are studied, i.e. 200 mm, 50 mm and 15 mm. The 200 mm movement is the same as in the earlier study of Linjama et al (2003a), so that results can be compared. The downstream pressure reference p_{dr} is selected rather arbitrarily to be a constant 6 bar.

4.4 Implementation of Controllers

Two controllers are studied: Controller 1 uses only standard state combinations, while Controller 2 considers also general state combinations, as described in Section 2.5. A proportional controller is used as a closed-loop controller and the closed-loop velocity reference is given by (see Fig. 4)

$$v_{rC} = v_r + K_P (x_r - x) \quad (8)$$

The measured supply pressure and calculated load force are filtered with a discrete-time second-order Butterworth filter and fed to the controller. The tank pressure is assumed to be zero.

The developed controller cannot stop the motion because the state combination $u_{PA}= u_{AT}= u_{PB}= u_{BT}=0$ is not included in the search space. Therefore, the controller is commanded to close all valves if v_{rC} is smaller than a certain treshold value v_{tr}. The threshold should be as small as possible, but too small value yields limit cycles. Maximum steady-state position error is v_{tr}/K_P, provided that the system is stable.

The procedure for calculating min(J) for standard state combinations is described in detail by Linjama et al (2003a) and is not repeated here. The output is the best standard state combination, which is also the output of Controller 1. The best standard state combination is used as an input to Controller 2 to determine the search space for general state combinations, as explained in Section 2.5. The steady-state values are calculated by three Newton-Raphson iterations of Eq. 6 with the following rules

- Initial value for $\boldsymbol{x}$ is $\boldsymbol{x}_0= [0, p_S/2, p_S/2]^T$
- Velocity is limited between –1 and 1 m/s during iterations
- Pressures are limited between $0.01*p_S$ and $3*p_S$ during iterations
- Solution is rejected and a big value is assigned to corresponding J if
  - Solution is not converged (over 10 N error in force or over 0.01 l/min error in flow),
  - Steady-state velocity is smaller than v_{tr} for extending movement or bigger than $-v_{tr}$ for retracting movement OR
  - Calculated steady-state pressure is smaller than 0.1 MPa

where p_S = 2.2 MPa.

5 Experimental Results

5.1 Tuning of Controllers

Controller parameters used are shown in Table 2. The controller gain K_P is tuned to be about one half of the critical gain of the system. The tuning parameter K_{pd} is tuned such that good velocity tracking and reasonable pressure tracking is achieved. The tuning parameter $K_{\Delta u}$ is tuned such that the number of valve switchings is reduced as much as possible without any significant decrease in control performance. The search space for general state combinations (n_1 and n_2) is found by using computer simulations. Tuning is done by increasing n_1 and n_2 until no significant improvement in control performance is seen. The stopping treshold v_{tr} is tuned as small as possible so that no limit cycles occur. The break frequency of low-pass filters of supply pressure and load force is selected such that 10 dB attenuation is achieved at the natural frequency of the system (≈100 rad/s).

Table 2: Numerical values of controller parameters

Controller gain K_P [1/s]	12
Downstream pressure error weight K_{pd} [$m^2/(s^2Pa^2)$]	3×10^{-16}
Weight for the number of valve switchings $K_{\Delta u}$ [m^2/s^2]	3×10^{-6}
Search space parameter n_1	5
Search space parameter n_2	3
Stopping treshold for Controller 1 [mm/s]	4
Stopping treshold for Controller 2 [mm/s]	2
Break frequency of low-pass filters [rad/s]	60

The parameters of Table 2 imply that maximum steady-state position error is 0.33 mm for Controller 1 and 0.17 mm for Controller 2. The number of analyzed general state combinations is 540 and the turnaround time of the whole control code is 1.8 ms on DS1006.

5.2 Measured Responses

Figure 7 depicts the measured 200 mm response for Controller 1 and Fig. 8 the same response for Controller 2. The position and velocity tracking behaviour is quite similar for both controllers and the maximum tracking error is about 1 mm. The pressure tracking is better for Controller 2 and force oscillations are smaller. There is a small overshoot with both controllers and the overshoot prolongs the settling time especially with Controller 2. This is caused by the high static friction of the cylinder and by the fact that the general state combinations are more sensitive to errors in the load force estimate.

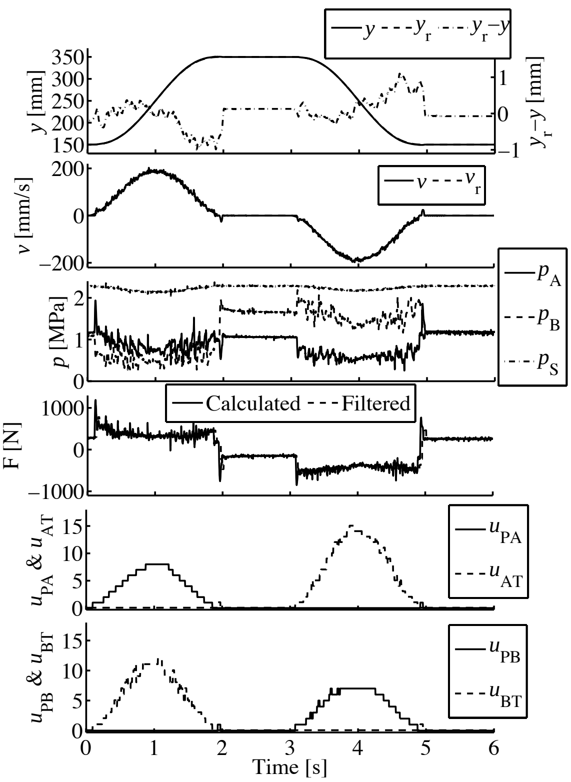

Fig. 7: *Measured 200 mm response with Controller 1*

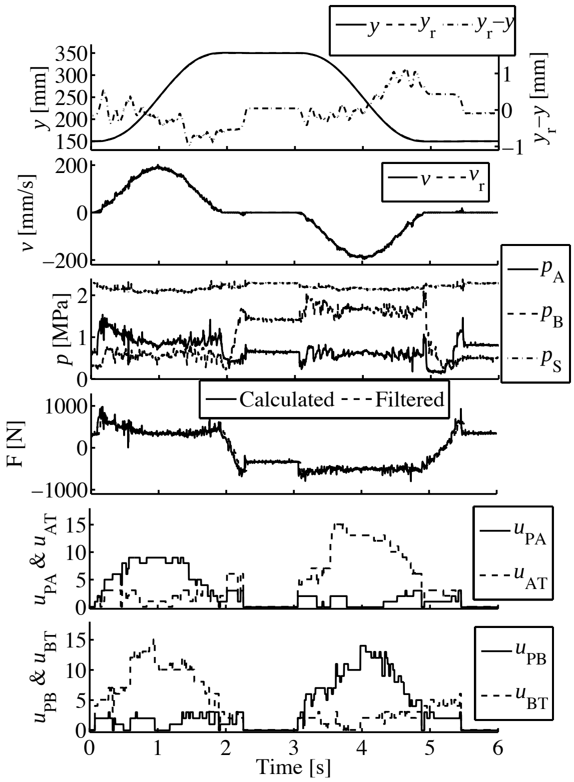

Fig. 8: *Measured 200 mm response with Controller 2*

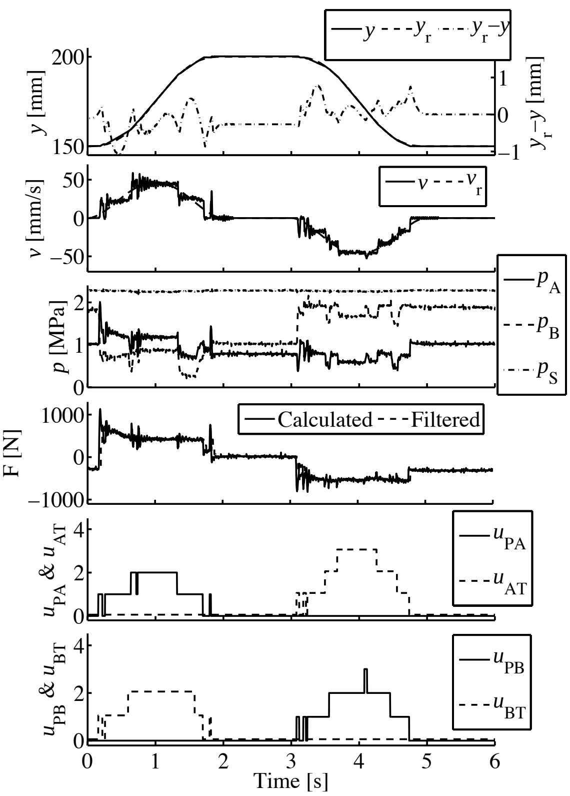

Fig. 9: *Measured 50 mm response with Controller 1*

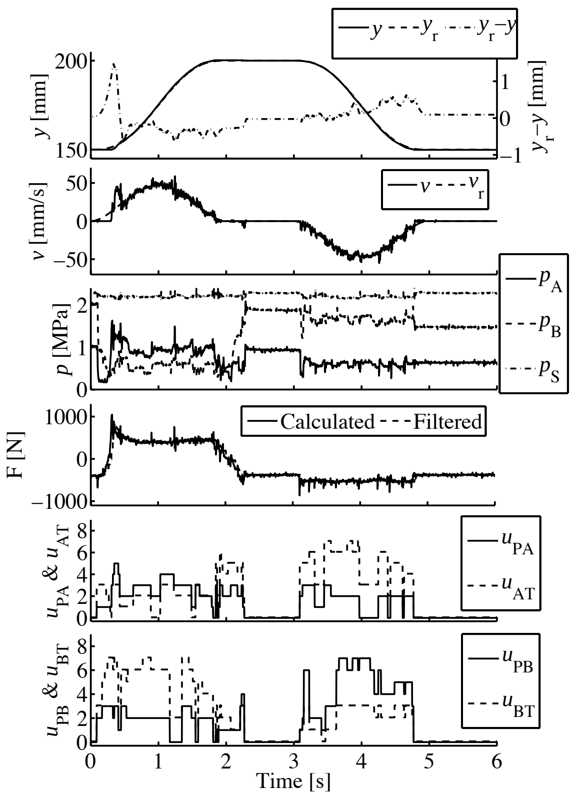

Fig. 10: *Measured 50 mm response with Controller 2*

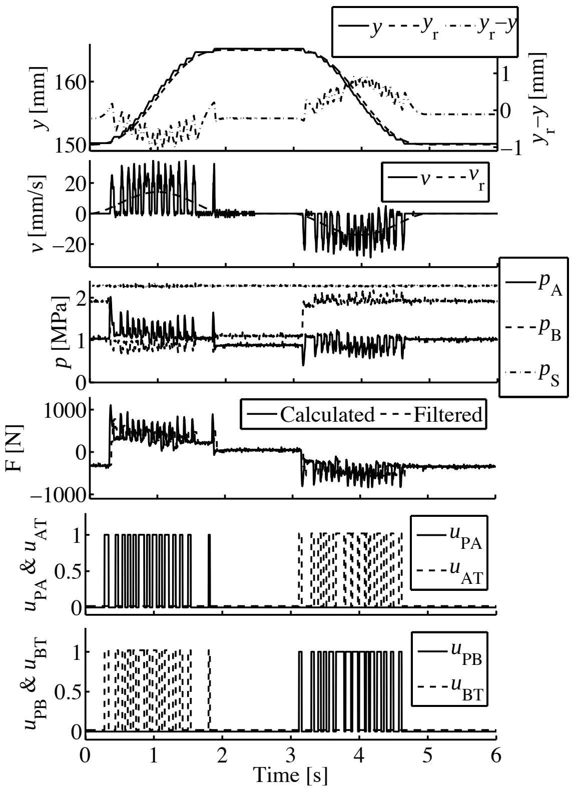

Fig. 11: *Measured 15 mm response with Controller 1*

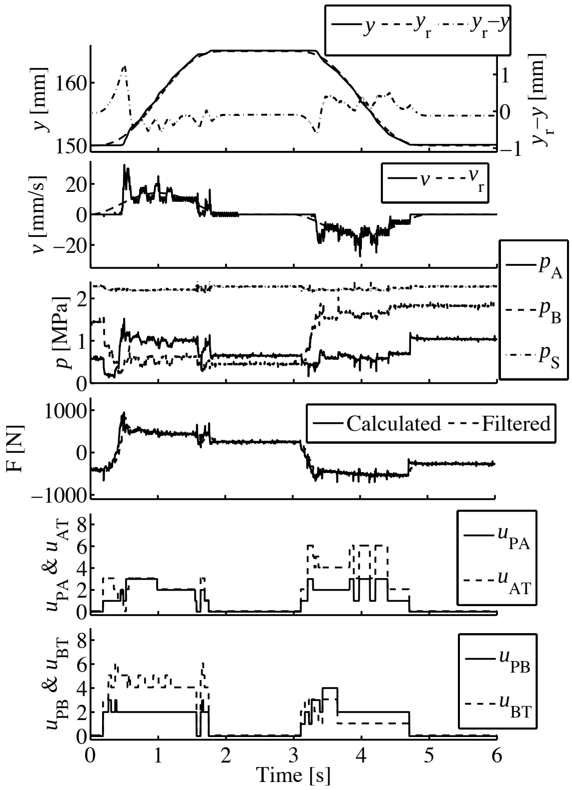

Fig. 12: *Measured 15 mm response with Controller 2*

Figures 9 and 10 present measured 50 mm responses for both controllers. Controller 1 gives rather jerky motion and poor velocity and pressure tracking. This is caused by the fact that there are so few suitable standard state combinations at low velocities. Controller 2 gives smooth motion and good pressure tracking, but large position tracking error exists at the beginning of the motion. This is caused by the static friction force and error is small during motion.

Figures 11 and 12 depict measured 15 mm responses. Controller 1 cannot track the velocity at all because the velocity reference is smaller than the minimum velocity of the system with standard state combinations. This results in jerky bang-bang type motion. Controller 2 can still perform some kind of velocity tracking and simultaneous pressure control. These figures show that the minimum velocity can be reduced about 80 percent and controllability at low velocities can be improved accordingly by including general state combinations.

6 Conclusions

The main contribution of this paper is the introduction and successful utilization of general state combinations, i.e. simultaneous and independent control of three or four flow paths instead of two. Experimental results show that controllability of digital hydraulic systems can be improved by including general state combinations. In the case studied, the minimum velocity is reduced about 80 percent even if only a fraction of general state combinations are considered. This in turn means that the number of valves in digital flow control units can be kept low. The ratio of about 50 between the maximum and minimum velocity can be achieved by four DFCUs each having four valves. The two-DFCU version with the same controllability would require at least six valves per DFCU and a large four-way valve yielding the same or higher costs.

The achieved tracking performance is 1 mm error with 200 mm/s peak velocity. This is at the same level as results of Mäkinen and Virvalo (2001) or Sairiala et al (2003) and it can be concluded that the tracking performance is comparable to traditional servo systems at high velocities. However, the low-velocity tracking, although improved, is not at the same level as in good servo systems. Mäkinen et al (1999) have demonstrated smooth tracking control with a water hydraulic servo valve at velocities around one percent of the maximum velocity. An obvious way to improve low-velocity tracking performance of the presented system is to increase the number of valves. Only minor pressure surges exist in measured responses even if the system has no damping elements as in the previous study (Linjama et al, 2003a). This is due to improved control electronics and the use of DC voltage, which reduce uncertainty in valve delays.

A drawback of the presented control approach is increased sensitivity to force variations. This causes increased tracking error at the beginning of motion when the friction force builds up. It must be noted, however, that the friction force of the system studied is relatively high, i.e. more than 25 percent of the maximum force. Another drawback is that the steady-state equations of general state combinations are hard to solve, which sets strong requirements for the computational power of the control hardware.

The simultaneous and independent control of three or four flow paths is seldom used and the authors know no publications about it. To use analogue valve terminology, the presented system has adjustable pressure gain, adjustable degree of over/underlap and adjustable pressure-flow coefficient. This opens new control principles that cannot be realised by standard valves. One example of this is ‘floating’ or ‘zero stiffness’ control, in which all flow paths are open but velocity is close to zero. Experimental results demonstrate also smooth ramp-type force change at the beginning of motion, which indicates that the system could be used in pressure or force control mode.

Nomenclature

A_A	Piston area, piston side	[m^2]
A_B	Piston area, piston rod side	[m^2]
$\mathbf{B}$	Binary matrix	
$\boldsymbol{b}_i$	i-th row of matrix $\mathbf{B}$	
F	Load force	[N]
$\hat{F}$	Load force estimate	[N]
$\boldsymbol{F}$	Vector of steady-state equations in Newton-Raphson iterations	[m^3/s, m^3/s, N]^T
J	Cost function	[m^2/s^2]
$\mathbf{J}$	Jacobian matrix in Newton-Raphson iterations	
K_P	Gain of closed-loop controller	[1/s]
K_{pd}	Weight for downstream pressure error	[m^2/(s^{2}Pa^2)]
$K_{\Delta u}$	Weight for valve switching	[m^2/s^2]
M_1	Minimum of cost function for standard state combinations	[m^2/s^2]
M_2	Minimum of cost function for general state combinations	[m^2/s^2]
n	Number of valves of DFCUs	[–]
n_{iter}	Number of Newton-Raphson iterations	[–]
n_{sw}	Number of valve switchings	[–]
n_1, n_2	Parameters to define search space of general state combinations	[–]
p_A	Pressure in A-chamber	[Pa]
p_B	Pressure in B-chamber	[Pa]
p_d	Calculated downstream pressure	[Pa]
p_{dr}	Downstream pressure reference	[Pa]
p_S	Supply pressure	[Pa]
p_T	Return pressure	[Pa]
Q	Flow rate	[m^3/s]
Q_N	Flow coefficient of valve or DFCU	[m^3/(s$\sqrt{\text{Pa}}$)]
u	State of DFCU or control signal of valve	[–]
v	Piston velocity	[m/s]
v_r	Piston velocity reference	[m/s]
v_{rC}	Closed-loop velocity reference	[m/s]
v_{tr}	Stopping threshold	[m/s]
$\boldsymbol{x}$	Vector of unknowns in Newton-Raphson iterations	[m/s, Pa, Pa]^T
$\boldsymbol{x}_0$	Initial value for $\boldsymbol{x}$	[m/s, Pa, Pa]^T
x	Piston position	[m]
x_r	Piston position reference	[m]
γ	Ratio of piston areas	[–]
κ	Ratio of flow coefficients of inflow and outflow DFCUs	[–]

Subscripts:

AT	DFCU A→T
ATi	i-th valve of DFCU A→T
BT	DFCU B→T
BTi	i-th valve of DFCU B→T
PA	DFCU P→A
PAi	i-th valve of DFCU P→A
PB	DFCU P→B
PBi	i-th valve of DFCU P→B
opt	Optimal state
subopt	Suboptimal state

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