


# High-Linear Digital Hydraulic Valve Control by an Equal Coded Valve System and Novel Switching Schemes

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## Abstract

This study proposes a novel digital hydraulic valve system using multiple equal size on/off valves and a circulating switching control, with an aim to increase the resolution and the linearity of the digital hydraulic valve systems. The solution is founded on the equal coded valve system concept which represents a recent development in the digital hydraulic valve technology. The circulating switching control algorithm is used to overcome non-linearities occurring in the typical non-circulating switching control and to decrease the operating frequency of single on/off valves. As a result, a substantial improvement in tracking control performance is demonstrated with 8+8 parallel connected valves. The results verify that compact, robust and high-performance valve control can be realized.

## Keywords

pulse frequency modulation, pulse width modulation, pulse number modulation, digital hydraulics.

## Nomenclature

$\omega_c$	Closed loop natural frequency [rad/s]	$Q_{ss}$	Steady state flow rate [m <sup>3</sup> /s]
$\omega_h$	Hydraulic natural frequency [rad/s]	$T$	Control period [s]
$\tau$	Time constant of P-controller filter [s]	$t_c$	Time correction parameter [s]
$\xi_c$	Closed loop damping factor [-]	$t_i$	Time to close one valve [s]
$\xi_h$	Damping factor [-]	$u$	Floating point control value [-]
$A_{cyl}$	Piston area [m <sup>2</sup> ]	$u_{max}$	Maximum control value of MVPWM [-]
$b$	Cavitation choking parameter [-]	$u_{min}$	Minimum control value of MVPWM [-]
$c$	Exponent of flow rate model [-]	$u_{int}$	Integer control value [-]
$f$	Frequency [Hz]	$v$	Piston velocity [m/s]
$f_c$	Control frequency [Hz]	$v_{ref,c}$	Closed loop velocity reference [m/s]
$i_h$	Head index of circular buffer [-]	$v_{ref}$	Velocity reference [m/s]
$i_t$	Tail index of circular buffer [-]	$w_f$	Pulse width of PFM control [s]
$K_p$	Proportional controller gain [1/s]	$w_p$	Pause width [s]
$K_v$	Flow coefficient [ $\frac{m^3}{s\sqrt{Pa}}$ ]	$w_v$	Virtual pulse width [s]
$N$	Number of valves in DFCU [-]	$w_{p,min}$	Minimum allowed pause width [s]
$n_s$	Number of open valves at beginning of $T$ [-]	$w_{v,min}$	Minimum allowed virtual pulse width [s]
$p_A$	A-chamber pressure [MPa]	$x$	Piston position [m]
$p_P$	Supply pressure [MPa]	$x_{ref}$	Position reference [m]
$p_{in}$	Input pressure of flow rate model [Pa]		
$p_{out}$	Output pressure of flow rate model [Pa]		
$Q$	Flow rate [m <sup>3</sup> /s]		

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## Introduction

Hydraulic actuation is widely used in the industrial and mobile applications where high power to weight ratio is required. High-performance valve control is an essential part of most hydraulically actuated servo systems. Servo valves, which are regarded as high-end valve technology from the performance point of view, are often criticized for their sensitivity against impurity and high cost. Digital hydraulic valve systems have been proposed to offer a reliable, cost-effective and energy-efficient alternative to the traditional analogue servo valves.<sup>1-4</sup> The advantages of this technology arise from the usage of robust on/off valves to realize distributed control edges.<sup>5</sup> One of the best advantages is a feasibility for model based control. Furthermore, the flexibility of such a valve system has promoted interest in this technology.

The Digital Flow Control Unit (DFCU) is a key component in these systems. It comprises several parallel connected on/off valves that may differ in size. A method using valves with equal flow capacities (i.e. equal coding) is theoretically superior, since an equal coded valve system is smaller, faster, consumes less electric power, and has the best controlling possibilities when compared to other coding methods.<sup>6</sup> The challenge, however, is that a large number of valves are required to obtain a sufficient flow resolution. The dream of using hundreds of microvalves seems to be currently unattainable. So the question is, how many valves do we really need?

Linjama et al. introduced a sum flow controller to improve the controllability of the digital hydraulic cylinder drive, especially at low velocities.<sup>7</sup> The main principle is to allow pressure and tank side DFCUs to open simultaneously. This affects cross flow directly from the pressure line to the tank line. The cylinder is controlled with the residual net-flow. This technique may increase the effective resolution since there are new flow rate levels for the cylinder. However, this method increases energy consumption and sensitivity to load force variations.

Pulse-Width Modulated (PWM) fast-switching valves are another possibility to implement digital flow control. Plenty of research has been carried out on this topic both in hydraulics and pneumatics. Lühmann B. and Muto et. al. have presented pioneering studies on PWM control in hydraulics.<sup>8,9</sup> An advantage of the PWM control system is its potential to achieve reasonably high resolution with one switching valve. However, there are conflicts between different characteristics. The high PWM carrier frequency decreases the pressure pulsation and improves dynamics but increases the nonlinear behavior of the valve. Dead bands, as the main nonlinearity, result from the valve's inability to change its state if the duration of the control pulse is shorter than the valve delay time. This occurs at the extreme regions of the duty cycle. Varseveld and Bone developed PWM schemes to linearize the behavior in a pneumatic system, but the solution consumes energy even at zero velocity.<sup>10</sup> Muto et al. presented a differential PWM method to remove the dead band, but the solution requires the use of symmetric cylinder and also causes it to lose the ability for a separate Meter In Meter Out control.<sup>11</sup> Belforte et al. used two PWM valves connected in parallel in a pneumatic system

with different PWM frequencies to improve the dynamic performance of the system.<sup>12</sup>

Schepers et al. developed an optimal control method that utilized ballistic modes to realize flow rates that are unattainable with the typical PWM control.<sup>13</sup> In the ballistic mode, the valve opens but the duration of the control pulse is shorter than the valve needs to open fully. There is also inverse ballistic mode when the control pause is shorter than the valve needs to close fully. The ballistic modes are highly non-linear and require a pressure balanced valve.

A promising solution to robustly manage the disadvantages of the PWM control and the low resolution of the DFCU is to combine DFCU and PWM control. Huova & Plöckinger added PWM valves into DFCU and reported significant improvement in the resolution.<sup>14</sup> Ferraresi studied different alternatives to using a pneumatic valve system, including nine similar valves connected in parallel and found the best configuration to be set as follows: All valves are in the original condition having equal flow capacities. One of the valves is driven with PWM. Eight valves are used to bias the flow rate level, and the PWM valve is used to fine-tune it.<sup>15</sup> In this way, the width of the dead band is rather narrow but it occurs in multiple sections. Another flaw of that solution is that a fixed PWM valve causes the uneven wearing of the valves. Paloniitty et al. combined an equal coded DFCU and switching control in a way that switching duty distributes for all valves.<sup>16</sup> However, they used only Pulse Frequency Modulation (PFM), which can improve the flow resolution only at small control values.

One application of PWM controlled valves, namely switching converters, share the challenge of limited PWM frequency. To solve that, Kogler and Scheidl proposed to replace a single switching converter with multiple parallel connected switching converters and to use them in phase shifted operation in order to increase the effective output frequency.<sup>17</sup> With their method, the output frequency multiplied by the number of parallel connected converters; however, they controlled the system with the initial frequency of a single converter. In other words, the PWM duty cycle of each converter was equal for one switching period although the PWM periods of each converter began in different timing.

As can be seen from the previous paragraphs, all benefits provided by the integration of DFCU and switching control are not yet examined. Thus, this study proposes a novel architecture to efficiently combine the principles of DFCU and switching control, aiming to obtain new state-of-the-art control performance in the category of digital hydraulic valves. The main objective of this paper is to implement a high-resolution and high-linear digital metering edge with a moderate number of equal on/off valves.

The paper is structured as follows. In Section II, a novel switching scheme to provide improved linearity is introduced, and the model based control of the system is described. In Section III, the limitations of the proposed switching scheme are analyzed, and the linearity of the output is compared with other solutions. A prototype valve system, a cylinder drive assembly and a reference method as well as the tuning of controllers are introduced in Section IV. In Section V, verification of valve dynamics and experimental results, for the reference method and for

the novel switching method, are presented. Finally, the conclusion is outlined in Section VI.

### Improving linearity of equal coded DFCU

The principle of increasing the linearity with the proposed method is quite similar with the phase shifting method proposed by Kogler and Scheidl.<sup>17</sup> However, there are some reasons why their method needs to be improved when utilized for DFCU control. They used the individual converters with a given frequency and controlled the system at that frequency. By adding more converters parallel, they increased the effective output frequency rather than linearity. We want to utilize a given output frequency and decrease the frequency of single valves in order to increase the linearity. That can be realized with the phase shifting method, but, as the drawback, the controlling frequency decreases as well. That is not acceptable since the frequency reduces remarkably when using multiple valves in parallel.

The proposed method can realize the above described frequency reduction in the valves, and, nonetheless, the system can be controlled with the non-decreased output frequency.

### Multi-Valve Pulse-Width Modulation

The input to the proposed algorithm is a floating point control value  $u$  which can have value from zero to  $N$ , where  $N$  is the number of the valves in the DFCU. The algorithm then aims to realize  $u$  as an effective opening over a control period. Since, the DFCU can realize only integer valued openings by nature, partial openings are conducted by closing one of the valves within the control period at the right timing. Thus,  $u$  should be converted to  $u_{int}$  which is the integer valued control signal at higher sampling rate than  $u$ . The conversion is made by the Multi-Valve Pulse-Width Modulation (MVPWM). First the *virtual pulse width*  $w_v$  is calculated. The expression for the  $w_v$  can be written as:

$$w_v = \frac{u}{f_c} = uT, \quad (1)$$

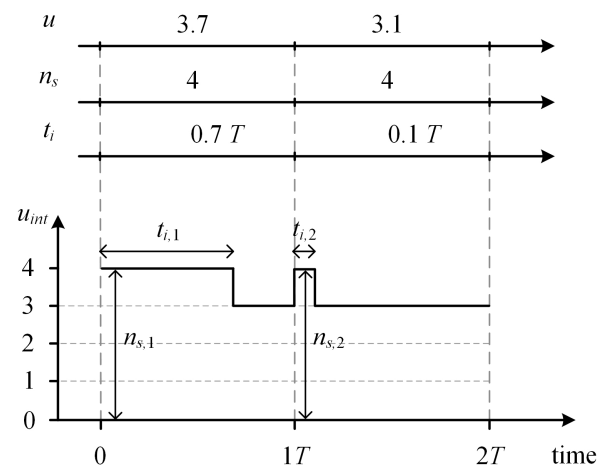
where  $f_c$  is the control frequency of the DFCU and  $T$  the corresponding control period. The *virtual pulse width* can be understood as a sum of the opening times over the control period when all the valves are considered separately.

Secondly, the *virtual pulse width* is converted to the parameters  $n_s$  and  $t_i$ . The parameter  $n_s$  determines the number of valves that should be open at the beginning of the control period and  $t_i$  determines the time when one of the valves should be closed. The expressions for them can be written as:

$$n_s = \text{floor}(u) + 1 \quad (2)$$

$$t_i = w_v \bmod T. \quad (3)$$

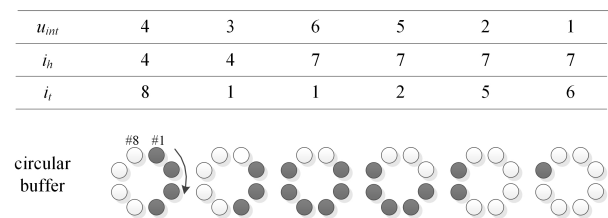
The principle of the modulation is shown in the Fig. 1. The upper part of the diagram shows the values of the parameters  $u$ ,  $n_s$  and  $t_i$  for two control periods. The lower part shows the opening scheme of the DFCU. Note that the diagram is not describing which valves in the DFCU are open. The MVPWM determines only the number of valves to be open.



**Figure 1.** Principle of MVPWM. Control scheme for two different control values are presented.

### Circular buffer

Circular buffering is a method by which the valve openings are equally distributed to the valves at the control edge.<sup>6</sup> The operation of the circular buffering is quite simple. The only input to the circular buffer is  $u_{int}$ . In addition, there are two inner variables: a head index  $i_h$  and a tail index  $i_t$ . To open valves, the head index is moved forward and to close valves, the tail index is moved forward. All the valves between the indices are kept open in a way that the head index determines the valve that is last opened and the tail index determines the valve that is last closed. The operation is visualized in the Fig. 2.



**Figure 2.** Principle of the circular buffer with eight valves. Gray color indicates opened valves. An imaginary control sequence is demonstrated.

Circular buffered multi-valve PWM is a system that uses the floating point control value  $u$  as an input and gives individual valve commands as an output. It uses multi-valve pulse width modulation to determine how many valves should be open and the circular buffer to determine which valves they are.

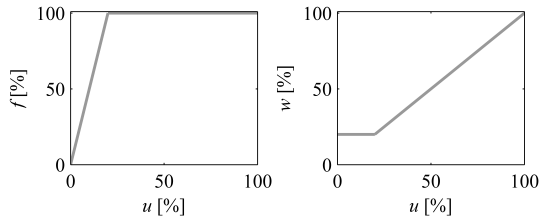
### Improving the linearity with Pulse Frequency Modulation

Despite the circular buffered multi-valve PWM, there is a dead band at small control values when the virtual pulse width becomes too short. In order to further increase the linearity of the digital valve control, PFM is used at very small control values. PFM control utilizes constant pulse width that can be selected long enough to enable the valve to response properly. Small control values are then realized

by decreasing the frequency (Fig. 3) rather than decreasing the pulse width and therefore the dead band is avoided. The frequency of the PFM control can be expressed as:

$$f(u) = \frac{u}{w_f}, \quad (4)$$

where  $w_f$  is the constant pulse width of the PFM scheme. PFM pulses are also transferred to the DFCU through the circular buffer. It means that opening pulses are conducted with different valves by turns. This enables overlapping pulses because the next valve could be opened before the previous valve is closed. However, in this study the modulation method is changed to PWM before the overlapping pulses occur.

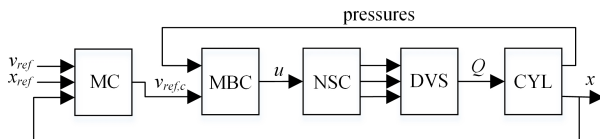


**Figure 3.** The pulse frequency modulation is used up to the control value when the frequency of the PWM control is achieved. After that the frequency is kept constant and PWM control is used

The drawback of this method is that the low frequency may lead to stop-go movements of an actuator, although correct mean velocity is achieved. Nevertheless, the PFM is effective to fine-tune the cylinder position.

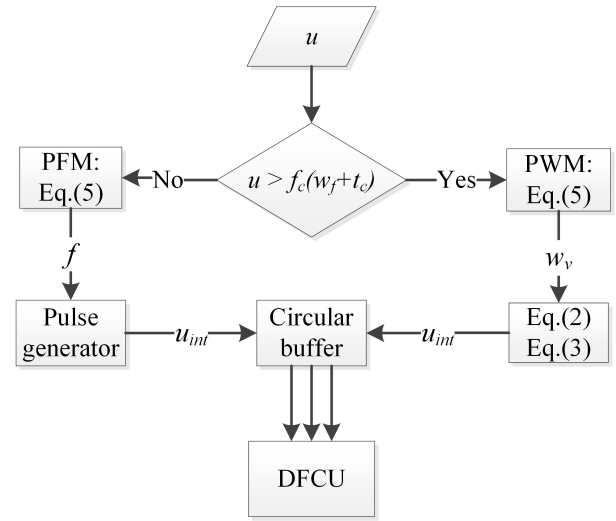
### Using the novel switching scheme

The novel switching scheme can be defined as a combination of PFM control, PWM control and circular buffering. It is aiming to realize a controlled opening fast and accurately. However, in order to control e.g. a cylinder with a velocity input, a model based control should be used. A simplified block diagram of the control system which has been used in this study can be seen in Fig. 4. The motion controller gives the closed loop velocity reference  $v_{ref,c}$  to the model based velocity controller. The model based velocity controller calculates the floating point control value for the novel switching controller. The novel switching controller uses the above described algorithm to control individual on/off valves in the digital valve system.



**Figure 4.** Block diagram of the control system. MC denotes motion controller, MBC denotes model based velocity controller, NSC denotes novel switching controller, DVS denotes the digital valve system and CYL denotes a hydraulic cylinder and a load.

**Novel switching controller** In the ideal case, the equations (1) and (4) can be used to calculate the frequency or the virtual pulse width in order to achieve the desired openings. In the case with asymmetric opening and closing delays, the realized pulse width differs from the controlled pulse



**Figure 5.** Calculation process in the novel switching controller

width. Typically, the closing has a longer delay, which results in a constant addition to the pulse width. Asymmetric delays can be modeled with a time correction parameter  $t_c$ , which describes the difference between the pulses. The parameter is positive when the realized pulse is longer than the controlled and negative in the opposite case. The expression for the novel switching scheme, including the time correction parameter, can be written as:

$$\begin{cases} f(u) = \frac{u}{w_f + t_c}, & w_v = w_f, & u \leq f_c(w_f + t_c) \\ w_v(u) = \frac{u}{f_c} - t_c, & f = f_c, & u > f_c(w_f + t_c) \end{cases} \quad (5)$$

where  $f_c(w_f + t_c)$  is the transition control value where the modulation method is changed. The calculation process is presented in the Fig. 5.

**Model based velocity controller** For sufficient accuracy, the generalized exponent model presented by Linjama et al. is used to model the flow rate of a single valve.<sup>18</sup> The model calculates the steady-state flow rate  $Q_{ss}$  from the port pressures  $p_{in}$  and  $p_{out}$  using three valve parameters: flow coefficient  $K_v$ , exponent  $c$  and cavitation choking parameter  $b$ . The model equation can be written as:

$$Q_{ss}(p_{in}, p_{out}) = \begin{cases} K_v(p_{in} - p_{out})^c, & bp_{in} < p_{out} \leq p_{in} \\ K_v[(1 - b)p_{in}]^c, & p_{out} \leq bp_{in}. \end{cases} \quad (6)$$

To keep the controller as simple as possible, only one steady-state flow model is used per DFCU, as per the concept presented by Paloniitty et al.<sup>19</sup> This approach assumes that all valves in the DFCU are equal. In reality there are variations in valve characteristics due to producing tolerances. Therefore, the model should be tuned for averaged characteristics. If parameters for all valves are measured separately, like in this study, mean values could be used. The control value  $u$  is calculated, based on the flow rate model, as follows:

$$u(v_{ref,c}, p_{in}, p_{out}) = \frac{v_{ref,c} A_{cyl}}{Q_{ss}(p_{in}, p_{out})}, \quad (7)$$

where  $A_{cyl}$  denotes the piston area.

**Motion controller** The motion controller utilizes a closed loop position control and an open loop velocity control. For position control, a discrete time filtered proportional controller is used in order to reduce the gain near the natural frequencies.<sup>20</sup> Thus, the closed loop velocity reference  $v_{ref,c}$  can be written as:

$$v_{ref,c} = v_{ref} + \frac{K_p}{\tau s + 1} (x_{ref} - x) \quad (8)$$

where  $v_{ref}$  denotes the open loop velocity reference,  $K_p$  denotes the proportional controller gain,  $\tau$  denotes the time constant of the P-controller filter,  $x$  and  $x_{ref}$  denote the piston position and position reference and  $s$  is the Laplace variable.

## Static analysis

Due to switching control, non-ideal valves and non-ideal power electronics, there are limitations on the control value. Limit values for the static control are derived in the following sections. The dynamic analysis is much more complicated. For example, the extreme control sequence would be a fluctuating signal between the full opening and the zero opening. If this happens at the controller frequency, the pause time of the valves is one controller period. With the power electronics of our prototype, this is too short pause time and therefore the sequence is unfeasible. The feasibility of a certain control sequence depends on the amplitude, the bias and the frequency of the control signal. The feasibility analysis of a dynamic control is not covered in this paper.

### Static limitations of the circular buffered Multi-Valve PWM

Limitations for the digital control come from the valve dynamics and power electronics. The state transition of an on/off valve can be derived into three stages:

1. the delay time,
2. the movement time and
3. the settling time.

The delay time is the time that the plunger stays at the previous position. The movement time is the time when the plunger is moving. The settling time is the time when the plunger is settling towards the mechanical stop. Based on the authors' experience, the closing delay varies if the valve is closed during the settling time. For the minimum variation in the closing delay, the minimum pulse width  $w_{v,min}$  in this study is selected to be 2 ms that is assumed to be long enough for the slowest valve to be fully settled. The settling time of closing event is not in the focus of this study, because the power electronics require 12 ms to store the energy for opening. This is surely longer time than the settling time. Thus, 12 ms is used as a minimum pause time  $w_{p,min}$  for the valves.

To analyze the limit values of  $u$ , the relation between the *virtual pulse width* and the realized pulse width of single

valves should be known. It can be noticed that if the control value is kept constant, they are identical. In this case, the period time for single valves is  $N T$ , which is the absolute maximum pulse width for single valves. The maximum usable pulse width is the period time minus minimum pause time. Thus the static limit values for the  $u$  can be expressed as:

$$u_{min} = f_c w_{v,min} \quad (9)$$

$$u_{max} = f_c \left( \frac{N}{f_c} - w_{p,min} \right) \quad (10)$$

Substituting the minimum pulse and pause widths (2 ms and 12 ms) in the eq.(9) and (10) and assuming  $f_c$  to be 200 Hz and  $N$  to be 8, we obtain the minimum and maximum control values for PWM control to be 0.4 and 5.6 respectively.

### Static limitations of the multi-valve PFM

The maximum control value with the PFM can be derived via the pause time of single valves. The period time for single valves is  $N$  times the period time of the PFM control. Hence, assuming that the control value stays constant, the pause width for a single valve can be written as:

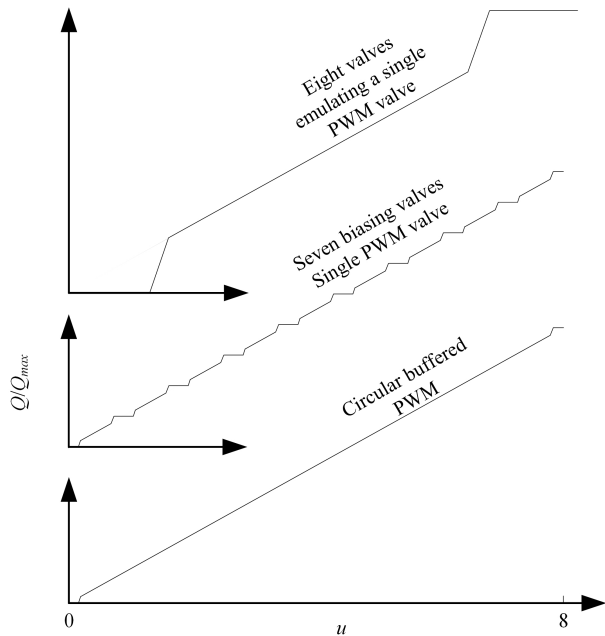
$$w_p = \frac{N w_f}{u} - w_f \quad (11)$$

Solving  $u$  and substituting the minimum pause width 12 ms for the  $w_p$ , we get the maximum control value of 1.15 for the PFM control. However, the PFM is used only up to the minimum control value of the PWM.

### Comparison of different PWM schemes

A principle comparison between three different ways to realize PWM control with multiple valves show the advantages of the circular buffered Multi-Valve PWM (see Fig. 6). All cases assume a DFCU with eight equal valves. The first one emulates a single, eight times bigger valve by controlling all valves with the same PWM signal. The second one controls one valve with a PWM signal and seven valves to bias the flow rate level, as presented by Ferraresi.<sup>15</sup> The third alternative is a circular buffered PWM. The main benefits of Ferraresi's solution compared to the single valve PWM are a decreased influence of dead bands and a decreased amplitude of pulsating flow rate. Circular buffered PWM control has following benefits compared to Ferraresi's solution:

1. Switching duty is equally distributed for all valves.
2. High switching frequencies are feasible because the switching frequency of a single valve is the output frequency divided by the number of valves in the DFCU.
3. Dead bands are eliminated except at very small and very high control values.
4. The power electronics of a single valve has more time to store energy for the next switching and is thus able to load the main power source more steadily.



**Figure 6.** Principle comparison of three different modulation methods with eight equal valves taking dead bands and ballistic zones into account.

## Test system

### Valve system

The used valve system consists of 16 prototype switching valves. The valve comprises five relatively simple turned parts as shown in Fig. 7. The valve structure is simple, direct-acting spring returned seat valve. The diameter of the coil is 10 mm. The material of the plunger and the upper part is Vacoflux 50, which represents a high-tech, soft magnetic material with a very high saturation magnetic flux and relatively high mechanical hardness. The seat part is made from AISI 303 stainless steel because of its high tendency to work hardening. The sleeve is made from basic structural steel, contributing to its low price, good machinability and magnetic properties. The coil framework is made from glass fiber-filled polyacetal because of its high stiffness and non-conductivity of electricity. It is also a relatively cheap material, and the part can be made by injection molding.



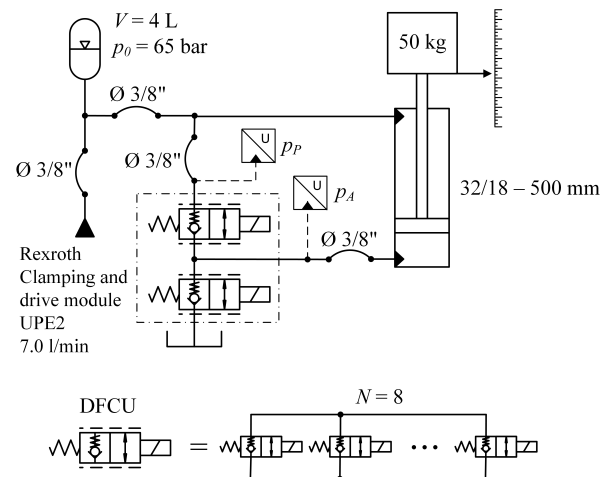
**Figure 7.** Each prototype valve consists of five turned parts, a coil and a spring.

The manifold for the valve system is designed for CETOP 3 interface and produced with the lamination technology presented by Paloniitty et al.<sup>19</sup> The principle is to stack pre-cut metal sheets and braze them together to form a compact manifold. Lamination technology enables the bending of flow channels and decreases the need for machining remarkably.

The valve system was driven with a prototype power electronic circuit, where the switching energy was stored in a capacitor. The capacitor produces a sufficient current peak to support fast opening. As a drawback, this circuit needs 12 ms to charge the capacitor. That significantly limits the maximum control value of the system; therefore, this type of circuit is not recommended to be used in future, unless the charging time is decreased.

### Test setup

Measurements are carried out with a system having a vertically assembled cylinder (32/18-500) with load mass of 50 kg (see Fig. 8 and 9). The chamber B is connected directly to the supply line. The prototype valve system is used as a three-way valve with two DFCUs, each having eight valves. The system contains an accumulator in the supply line. The pump is switched on when pressure in the supply line decreases below 70 bar and switched off when the pressure exceeds 80 bar.

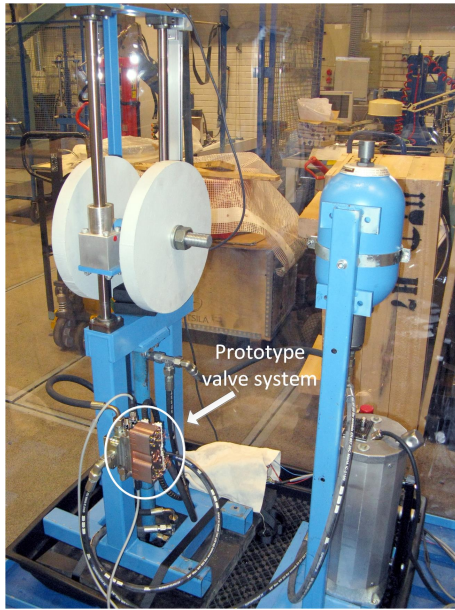


**Figure 8.** The hydraulic circuit of the test system

The control system is implemented with a dSpace DS1103 controller board. Pressures are measured with Trafag NAH transmitters with the pressure range of 250 bar. The position of the cylinder piston is measured with a Heidenhain LS477 linear encoder with a resolution of 0.5  $\mu\text{m}$ . The natural frequency of the system is about 120 rad/s in the middlestroke.

### Reference method

Pulse Number Modulation (PNM) is a basic modulation method with equal coded valve systems. It means that the output is controlled by the number of opened valves. The main advantage of this method is that it uses only static openings. The drawback is that the resolution is rather low, the same as the number of valves in the DFCU. This is used as a reference method when estimating the performance of the novel switching scheme. The PNM controller is based on the accurate valve models and predefined cost function that involves terms for the velocity error and for the number of switchings. The opening state for the next sampling period is that which minimizes the cost function. The controller includes steady-state valve models for each of



**Figure 9.** A photograph from the test system

the 16 valves and calculates cylinder's steady-state velocities for each allowed opening combination. Both DFCUs have eight allowed opening states, which are the openings of  $n$  first valves. Thus, there are 17 allowed states including the zero opening.

### Tuning of controllers

**Motion controller** The motion controller tuning is based on the assumption that the transfer function from  $v_{ref,c}$  to  $Q$  is  $A_{cyl}$  in the applied frequency range (see the block diagram in Fig. 4). Now, the plant model, including the model based velocity controller, the novel switching controller, the valve system, the cylinder and the load, can be written as:

$$\frac{x}{v_{ref,c}} = A_{cyl} \frac{\frac{1}{A_{cyl}}}{s \left( \frac{s^2}{\omega_h^2} + \frac{2\xi_h s}{\omega_h} + 1 \right)}, \quad (12)$$

where  $\omega_h$  is the hydraulic natural frequency and  $\xi_h$  is the damping factor.<sup>21</sup> Since the motion controller uses the Eq. (8), tunable parameters are  $K_p$  and  $\tau$ . When  $\tau$  is significantly larger than  $1/\omega_h$ , the dynamics of the cylinder and load system can be neglected and the open loop transfer function from  $x_{ref}$  to  $x$  is approximately

$$\frac{x}{x_{ref}} = \frac{K_p}{s(\tau s + 1)}. \quad (13)$$

An addition of feedback with unity gain results in a second order transfer function with a closed loop natural frequency and closed loop damping factor as written in Eq. (14) and (15).<sup>20</sup>

$$\omega_c = \sqrt{K_p/\tau} \quad (14)$$

$$\xi_c = 1/(2\sqrt{K_p\tau}) \quad (15)$$

To keep these equations valid,  $\tau$  is selected to be  $3/\omega_h$ . Then,  $K_p$  is selected to be 30, yielding the closed loop damping factor of 0.58 and closed loop natural frequency

of 35 rad/s. Controller and DFCU model parameters are gathered by Table 1 and 2, respectively.

The closed loop velocity reference  $v_{ref,c}$  from the motion controller is fed to the model based valve controllers which generate the best opening for the next sampling period. Two valve controllers are studied; the PNM controller and the novel switching controller.

**Table 1.** Controller parameters

Description	Value
Proportional controller gain $K_p$	30 1/s
Time constant of P-controller filter $\tau$	25 ms
Cut-off frequency of pressure filters	40 rad/s
Sample time of controllers	5 ms
Sample time of pulse generation	0.5 ms
Stopping threshold for PNM	9 mm/s
Stopping threshold for Switching	2 mm/s
Weight of the switching cost	0.01 m/s

**Table 2.** Parameters for DFCU models

Description	P-side	T-side
Time correction	2.7 ms	1.7 ms
Flow coefficient	$4.87 \cdot 10^{-9}$	$13.4 \cdot 10^{-9}$
Exponent	0.57	0.50
Cavitation choking	0.42	0.29

**PNM controller** Tunable parameters in the PNM controller is the weight of the switching cost and the stopping threshold. When  $|v_{ref,c}|$  is below the stopping threshold, all the valves are kept closed. The threshold is tuned as small as possible without the limit cycle oscillation. This yields different threshold values for both control method.

**Novel switching controller** Control pulse time for PFM control is 2 ms for both DFCUs. Time correction parameters were tuned with open loop tests with a medium size trajectory so that small position errors occur at the ends of the movements in both directions. Obtained values are 2.7 ms for the pressure side and 1.7 ms for the tank side. The PWM frequency is 200 Hz so the control values where PFM is switched to PWM are 0.94 for the pressure side and 0.74 for the tank side. The switching controller also has the stop mode implemented in the same way as the reference controller. However, better resolution of this controller yields smaller tolerance values. For valve model parameters, the average values of measured valve parameters were used.

### Trajectories

For the experimental study, the fifth order polynomial is used as a position reference. This trajectory shape, with the movement time of 2 seconds, is similar than those used in previous studies.<sup>7,16</sup> However, the length of the longest stroke differ from the references. The stroke lengths for this study are 15 mm, 50 mm and 100 mm.

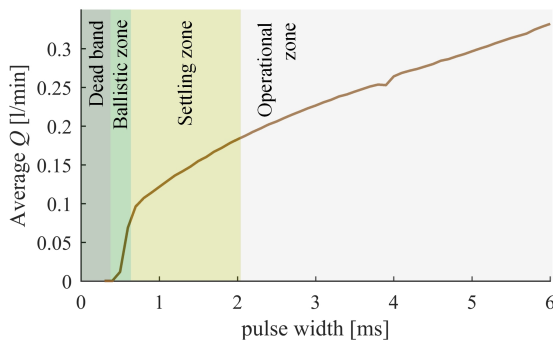
## Experimental results

### Verification of valve dynamics

The difference between opening and closing delays are modeled in the controller as constant. In order to verify that this approximation is effective in the operation range, the relation between the control (pulse width) and the output flow rate was studied.

The experiment was conducted with the test setup presented earlier. A single valve at a time was controlled with 50 Hz PWM signal. The average output flow rate was deduced from the cylinder displacement. Pulse width was varied by 0.1 ms steps. The supply pressure was kept between 84 and 85 bars. The average flow rate was plotted as a function of pulse width.

One typical result is seen in the Fig. 10. Four regions can be observed from the data. At the dead band, when the pulse width is shorter than the opening delay, there are neither mechanical response nor an observable flow rate. A ballistic zone appears when the pulse width is longer than delay but shorter than plunger needs for full movement. After about two milliseconds, the slope of the curve stays approximately constant. That means a constant gain from the pulse width to the flow rate. Supposedly, the constant gain is the consequence of the constant difference between opening and closing delay. This result supports the choice of 2 ms for minimum pulse length.

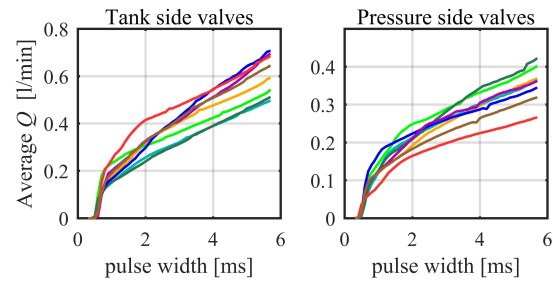


**Figure 10.** Effect of pulse width on average flow rate at the frequency of 50 Hz

All the results are plotted in Fig. 11. It can be seen that the curves differ quite a lot among the valves. That means different flow rates between the valves. That might result in pressure fluctuation. However, the simplified flow model that utilize average parameters can estimate the average velocity due to the fact that different valves are used frequently at high frequency. The feedback controller compensates the inaccuracies of the model as well.

### Tracking control

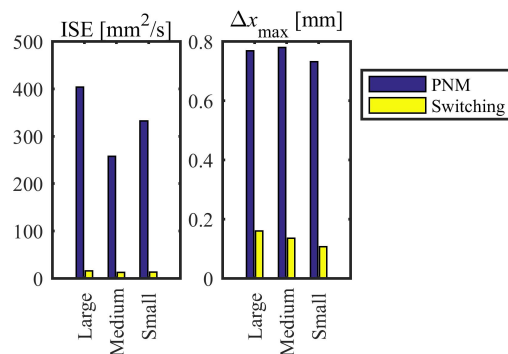
The three different size trajectories were measured with the three studied control methods. All measurements were carried out five times to increase reliability. Figures 14 and 15 depict the typical measured data from the medium size trajectory. Typical system behavior with both control methods can be seen on those figures. To compare the control methods, the tracking control performance is estimated with a maximum position error  $\Delta x_{\max}$  and an Integral of



**Figure 11.** All the results of the valve dynamics verification test measured with the frequency of 50 Hz

Square Error (ISE). Values are averaged from the repeated measurements and are presented with bar plots in Fig. 12. The size of the trajectory seems to have only a slight effect on the performance while the difference between control methods is clear. As a main result, the performance improvement with the novel switching control is significant.

For a supplementary result, Fig. 13 depicts an example of the novel switching scheme behavior. The data are obtained from the beginning of the medium size trajectory extending movement. The principle of the combined pulse frequency and pulse width modulation is clearly seen. With PWM control the system can follow the floating point control signal in 0.1 steps by varying the pulse width by 0.5 ms steps. The accuracy could be increased by increasing the resolution of the pulse width. In the figure, an inconsistency can be observed at the point where the modulation method changes. Pulses in PFM and PWM control are not synchronized, thus pulse and pause times at the switching point are incorrect. However, the influence seems to be minimal.



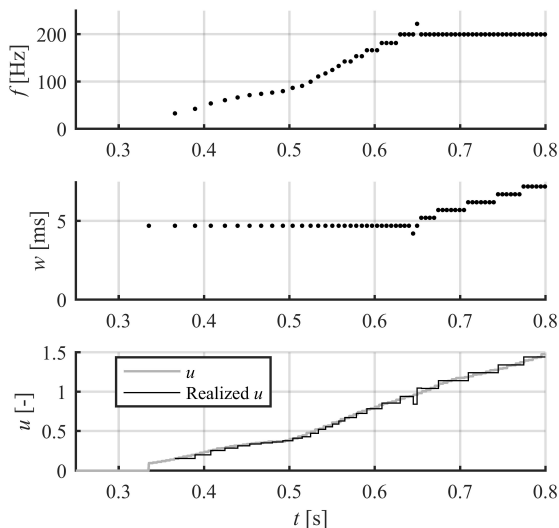
**Figure 12.** Measured tracking control performance of the two studied controllers

To compare the achieved tracking control performance to the corresponding result of Linjama and Vilenius, the ratios between the maximum position error  $\Delta x_{\max}$  and  $v_{\max}$  is calculated. The smaller the ratio is the better the tracking performance. The benefit of this performance indicator is the scalability. Different size systems and trajectories can be compared. The achieved ratio in this study is 1.7 (ms) while the ratio achieved by Linjama and Vilenius was 5.3 (ms).<sup>7</sup>

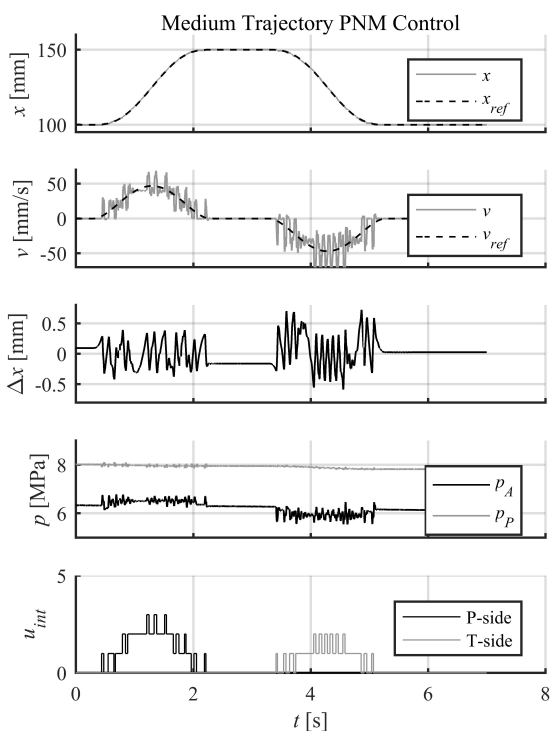
## Conclusion

The principle of high-resolution and high-linear digital hydraulic valve control is investigated. Results show state-of-the-art tracking control performance in the category of





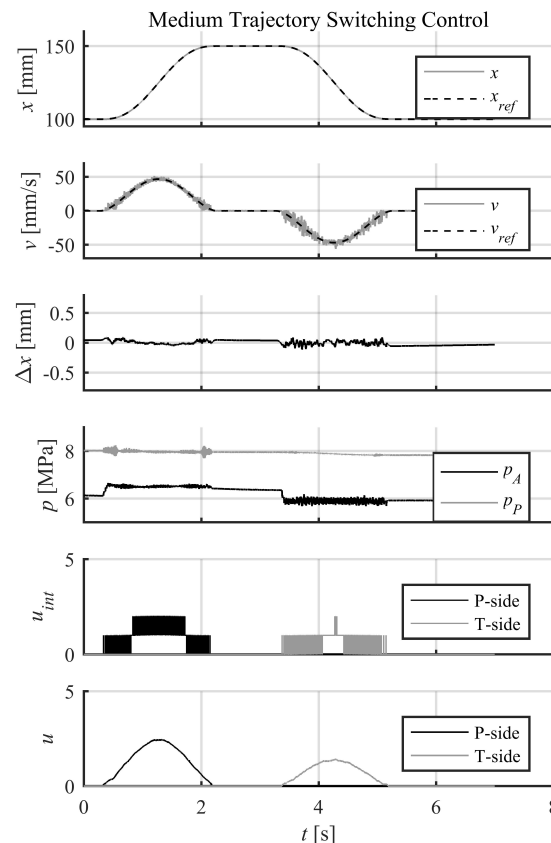
**Figure 13.** Realized pulse width and frequency at the beginning of the medium size trajectory extending movement. The dots in the upper diagrams depict single pulses.



**Figure 14.** Measured data from the medium size trajectory with PNM control

digital hydraulic valves, using only eight equally sized valves per DFCU and two-edge control. The key feature behind this achievement is to combine three digital hydraulic state-of-the-art methods: equal coding, parallel connected valve series, and switching control. This combination is enabled by the novel digital hydraulic miniature valves.

The proposed method generates pressure fluctuation and noise; however, the levels are lower than in the typical switching control because the full flow rate is not switched. The algorithm gives significantly better flow resolution than



**Figure 15.** Measured data from the medium size trajectory with novel switching control

former parallel connected solutions due to switching control. As a drawback, the switching control shortens a life time of the valves although the switching duty is distributed to many valves. The results of this paper show that former hypotheses about the required number of valves in an equal coded DFCU is overestimated.

### Declaration of conflicting interests

The Authors declare that there is no conflict of interest

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