

## Analysis by Simulation of Different Control Algorithms of A Digital Hydraulic Two-Actuator System

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Many hydraulic systems have losses, which could be avoided with new technology. Because component efficiency can be optimized to a certain operation point, hydraulic machines are no worse than other machines. More important than the peak efficiency values of each individual component in a system is the efficiency of the whole power transfer line. In a system where the amount of required power and the velocity/force ratio are variables, components may but seldom operate at their optimal design points. A typical approach to mobile work hydraulics is to use a load-sensing pump for a hydraulic multi-actuator system. This approach is efficient but seldom, if many actuators are used simultaneously. Our recent prototype of an improved hydraulic power supply system is the Digital Hydraulic Power Management System (DHPMS), which can serve many actuators at optimised supply pressure but is also capable of motoring and transforming. This functionality holistically reduces losses in the system. Losses can be further reduced by using distributed valve systems with sophisticated control algorithms together with the DHPMS. In this study, we used digital hydraulic valves, which efficiency strongly depends on the control algorithms used. We studied here different control methods for a system with two actuators, a DHPMS, and digital valves.

**Keywords:** digital hydraulic valve; Digital Hydraulic Power Management System; efficiency

### 1. Introduction

Though the energy efficiency of hydraulic systems is often poor, hydraulic booms are widely used in various mobile machines for practical reasons. The key benefits of using hydraulics in mobile machines are the high power/weight ratio and the flexibility of installing the systems. However, the efficiency of commonly used hydraulic booms with load sensing hydraulics can be extremely low, e.g., only 4% (Virvalo, 2000). This together with the facts that hydraulic machines have typically high power prime movers, and that many of them are operated even in three shifts results in huge total energy losses. As the price of energy continues to increase, cutting down on these losses is becoming an economic necessity. In addition, as some demanding consumers are also concerned about environmental issues, improved efficiency will add to the equipment's market value.

In state-of-the-art load sensing (LS) systems, the supply pressure is normally adjusted to 1.5 – 3.0 MPa over the highest load pressure; therefore, the pressure difference over the control valve may be kept at a somewhat acceptable level, but only over the valve that controls the highest load. Because the pressure differential of the other valves is not adjusted, major losses may occur. The worst case is when one actuator requires a high force while another, a fast moving one, requires a low force. Now supply pressure must be set to the demand of the highest force, which then causes huge losses in the pressure compensator of the fast

moving actuator, which requires less force. Furthermore, a typical single spool control valve has back pressure at the from-actuator-to-tank control notch, which is especially high to maintain good controllability with overrunning loads.

Recently, various ways have been studied to improve efficiency. One such is to use a separate pump to feed optimal pressure and flow to each actuator. For example, Artemis Intelligent Power Ltd. has patented a pack of pumps capable of doing this (Rampen, et al., 2008). The pump also features high partial flow efficiency because its structure is not based on the swash plate design; instead the number of pumping pistons is affected by disabling the unnecessary ones. The drawback of this solution is that it increases the pulsation of the flow. Optimal pressure can be generated for each actuator by using hydraulic switching converters (Scheidl, et al., 2008) or hydraulic transformers. The latter could be either rotary (Vael, et al., 2003) or linear (Bishop, 2009) versions of a similar principle. The effect of transformers in a multi-actuator system has been simulated, e.g., by (Sgro, et al., 2010), and the results indicate a major decrease in losses. An alternative to a constant pressure system and transformers with normal actuators is to use a constant pressure system with secondary controlled actuators. Secondary controlled pumps/motors have been used in mooring applications, but secondary controlled cylinders are an innovation (Linjama, et al., 2009). In valve controlled systems, losses caused by an unnecessary back pressure can be minimized by using

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separate-meter-in-separate-meter-out (SMISMO) control. Multi-actuator proportional SMISMO control with a simple control algorithm has been case-studied with a mid-size power tractor (Borghi, et al., 2010) with 17% energy savings. Even higher energy savings, 33-63%, were achieved with a multi-actuator system controlled with digital hydraulic valves and intelligent online optimization (Huova, et al., 2010). Also researched has been the SMISMO-principle in multi-actuator systems with a new type Negative Load Sensing function (NLS) (Erkkilä, et al., 2009).

In our previous paper (Karvonen, et al., 2011), we ran simulations to compare our recently developed prototype of the Digital Hydraulic Power Management System, the DHPMS (Linjama, et al., 2009), over an LS-pump to improve the efficiency of a multi-actuator hydraulic boom controlled with proportional valves. Papers have also been published on the basic features of a pump with independently controlled piston valves (Tammisto, et al., 2010) and on control methods of the DHPMS (Heikkilä, et al., 2010). Furthermore, laboratory measurements have been reported on a six-piston DHPMS in test bench conditions (Heikkilä, et al., 2010). More ideas to use the DHPMS concept to improve the efficiency of hydraulic systems are available in (Linjama, et al., 2009).

Previously, controllers for digital hydraulic valves have been tested mostly with single-actuator systems, and so too the DHPMS alone but not as a part of a whole system. Consequently, holistic analysis is necessary for a system with a DHPMS for power source/sink and equipped with digital valves. Hence previous controllers were modified and parameters were set for all controllers to work together for the end result of a well functioning, efficient system. Of all the tested controllers, good tracking of the given reference was required, and any differences in their energy consumption were carefully monitored. Before any testing, simulations should be run to verify hypotheses and test the controllers, and this paper is about the results of those simulations.

## 2. Digital Hydraulic Valve

A digital hydraulic valve is based on digital flow control units, DFCUs. One DFCU contains several parallel connected on/off valves. Because the valves may differ in size, different flow rates are achieved by varying the opening combinations. With similar valves, only the number of opened valves affects the flow. Thus to create a 4-way valve, four DFCUs are needed (schematics shown in Figure 1).

The possible control features of digital hydraulic valves are similar to those of any valve with separate control notches, but the simplicity of its base units, the on/off valves, enables accurate model-based control and change of control mode during movement. In addition, to realize a digital hydraulic system requires special multi-channel power electronics and a sophisticated controller platform.

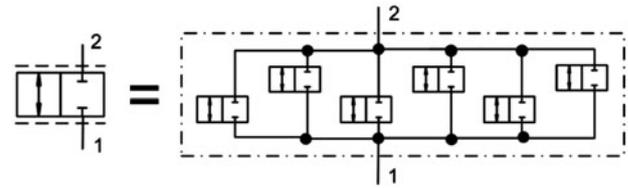


Figure 1. A digital 6-bit control notch, a DFCU, and its simplified drawing symbol. The dashed proportional line represents discrete valued functionality.

## 3. Digital Hydraulic Power Management System – The DHPMS

### 3.1. Principle of the machine

The origin of the DHPMS lies in a utopia of an ideal sink/source of hydraulic power capable of independent feeding or draining power to or from any of numerous actuator lines at any pressure and any flow rate and of storing energy for later use. A two-outlet version of such a machine can at some level be realized by modifying the valve plate of a piston pump so that each piston can be individually connected to either port-A, port-B, or the Tank line. This enables each piston to individually either pump to or motor from any port. Modes between pumping, motoring, and transforming are switched by using on/off valves in a correct order and timing. The pumping and motoring modes are those of individual piston elements and their working principle is straightforward. The transformer function requires many piston elements and can be executed by, e.g., motoring from port A with four piston elements while pumping to port B with two piston elements. Power transfer could be used in certain suitable load conditions but also to increase the energy capacity of an accumulator by enabling it to utilize a wider pressure range than usual whereby the accumulator is directly connected to a pressure line (Linjama, 2010). However, regeneration or transforming is not possible if the valves controlling the actuators cannot route power in all required directions (a hydraulic diagram of the simulated system is shown in Figure 2).

### 3.2. Control of the DHPMS

The control of the DHPMS seeks to maintain pressures in the actuator supply line at target values, which are set, e.g., according to ELS-functions. In the studied boxer pump unit, six mode decision instants occurred during one pump revolution, and at each of them the pumping mode was selected for one piston and the suction mode for the other (Heikkilä, et al., 2010). Evidently, in the pumping and suction phase, pistons could be connected to either actuator supply line or to the tank line, resulting in three possible pumping and suction modes (T, A, B) for each piston. Active digital valves were further controlled according to piston modes.

Figure 4 shows the pressure control logic of the DHPMS. The block diagram presents a model-predictive

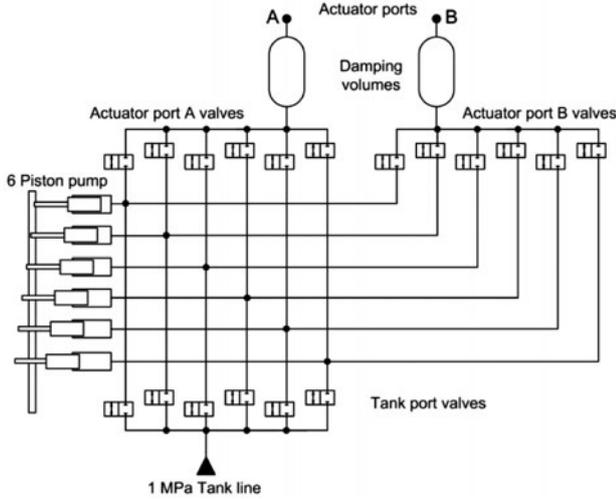


Figure 2. A hydraulic diagram of a DHPMS with two actuator ports. Constant volumes of 5 liters have been attached to the supply lines to dampen pressure ripple. This 6-piston pump is a “boxer,” not “inline,” as shown in the simplified graph.

mode selection logic for a pair of pistons with opposite phases. First, a change in the supply line fluid volumes is estimated against previously selected modes and actuator flows. Then, linear extrapolation as a function of piston angles is used to determine an additional fluid volume due to incomplete strokes. As a result, the vector  $\Delta V_{1\_est} = [\Delta V_{A\_1\_est}, \Delta V_{B\_1\_est}]$  is formed. Similarly, the actuator flow volumes  $\Delta V_{A\_2\_est}$  and  $\Delta V_{B\_2\_est}$  are extrapolated till the stroke end by assuming that actuator flows remain unchangeable. Actuator flow estimates  $Q_{2\_est} = [Q_{A\_2\_est}, Q_{B\_2\_est}]$  can be calculated from actuator velocity references owing to the rapid pressure response in transient states.

For all mode combinations, errors in the supply line pressure are calculated considering measured pressures  $p = [p_A, p_B]$  and their target values  $p_{ref} = [p_{A\_ref}, p_{B\_ref}]$  when the change in the fluid volumes  $\Delta V_{est} = [\Delta V_{A\_est}, \Delta V_{B\_est}]$  and hydraulic capacitances  $C_h = [C_{h\_A}, C_{h\_B}]$  are known. In this study, the supply line capacitances were considered constant, and the equation can be written as

$$|p_{err\_est}| = |p_{ref} - p - (\Delta V_{est} + \Delta V_{mode})/C_h| \quad (1)$$

where  $\Delta V_{mode} = [\Delta V_{A\_mode}, \Delta V_{B\_mode}]$ . Possible values of  $\Delta V_{A\_mode}$  and  $\Delta V_{B\_mode}$  are shown in Table 1, where  $V_{disp}$  is the geometric piston displacement. Because we excluded the mode combinations of pumping and suction chosen for the same supply, the optimal mode combination was a search amongst seven candidates by minimizing pressure errors. Optimal modes were then routed to the valve switching controller, and correct valve timing was determined in relation to pressures and angular velocity to optimize pre-compression and pressure release times and to compensate for valve delays (Heikkilä, et al., 2010). The sample time of the DHPMS controller was 50  $\mu$ s.

Table 1. Possible mode combinations and their effect on supply line volumes

| Pumping | Suction | $\Delta V_{A\_mode}$ | $\Delta V_{B\_mode}$ |
|---------|---------|----------------------|----------------------|
| T       | T       | 0                    | 0                    |
| A       | T       | $V_{disp}$           | 0                    |
| B       | T       | 0                    | $V_{disp}$           |
| T       | A       | $-V_{disp}$          | 0                    |
| A       | A       | NOT USED             |                      |
| B       | A       | $-V_{disp}$          | $V_{disp}$           |
| T       | B       | 0                    | $-V_{disp}$          |
| A       | B       | $V_{disp}$           | $-V_{disp}$          |
| B       | B       | NOT USED             |                      |

#### 4. Simulation Models

We used simulation models similar to those in (Karvonen, et al., 2011) with MATLAB/Simulink and SimMechanics for simulations. Equations are shown in this section and detailed parameters in appendix A. Of controller parameters, only those representing physical reality are included. Stiff solver is used.

##### 4.1. Mechanical model

Figure 3 shows a CAD image of the machine. Only lift and tilt functions were modeled, and desired load mass was modeled for the tip of the machine instead of the bucket. Inertia ellipsoids of the mechanical model are shown in Figure 5. Cylinders were not modeled as bodies but only as force sources. The mechanical model’s cylinder connection points were used to calculate directions for the forces applied. In addition, cylinder lengths and velocities were calculated from the connection point coordinates.

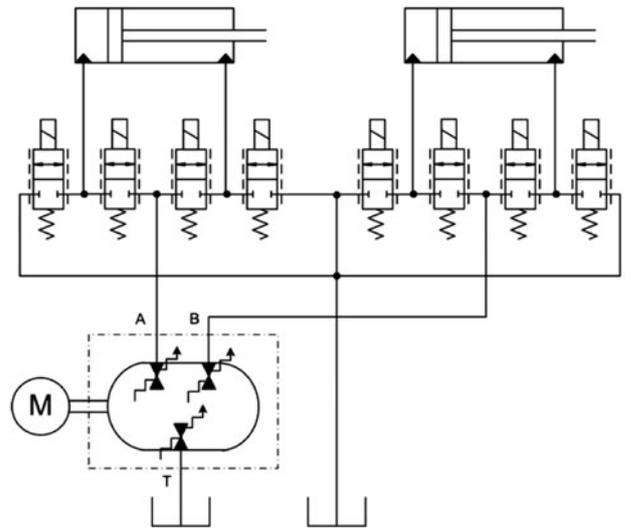


Figure 3. A hydraulic diagram of a DHPMS connected to two separate supply lines for actuators controlled by digital hydraulic valves. Each control notch is composed of six parallel connected on/off valves. The marked area represents the system in Figure 2.

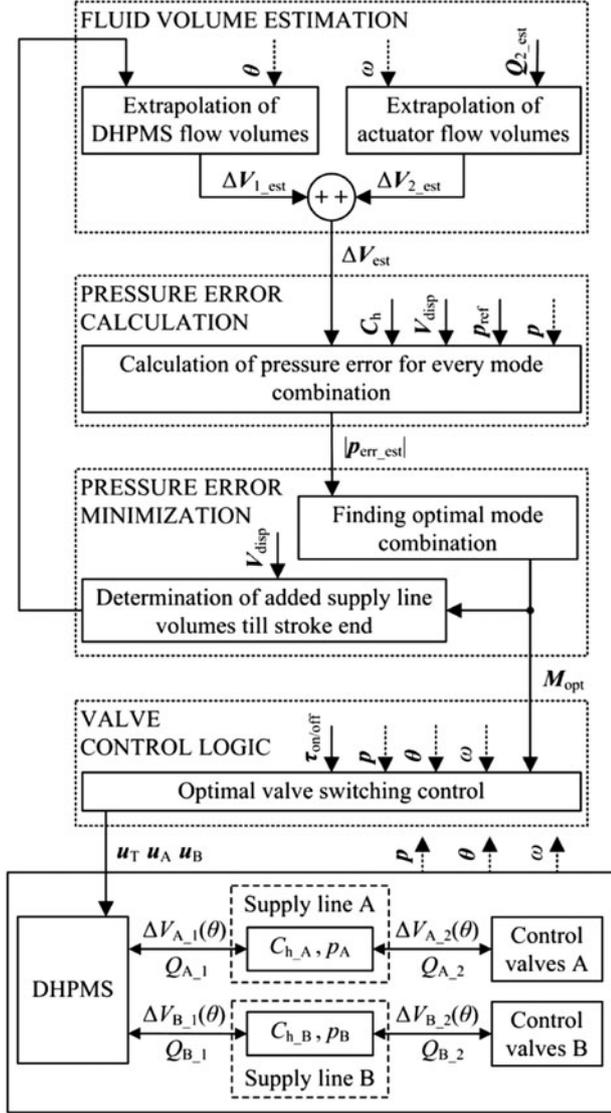


Figure 4. Mode control logic of the DHPMS

#### 4.2. Hydraulic models

Cylinders were modeled as two volumes linked together so that the volumes of the cylinder chambers corresponded to reality. Eq. 2 shows an equation for the pressure dynamics. Cylinder friction was modeled with a friction model based on a hyperbolic tangent, as shown in Eq. 3. Friction parameters are trivial in view of appendix A, containing the simulation parameters.

$$\frac{dp}{dt} = \frac{B_{eff}}{V} \left( \sum Q - \frac{dV}{dt} \right) \quad (2)$$

$$F_{fric} = \tanh(K \cdot \dot{x}) \cdot (F_c + (F_s - F_c) \cdot e^{-(\dot{x}/v_s)^2}) + b \cdot \dot{x} \quad (3)$$

On/off valves were modeled with the empirical equation of a turbulent orifice:

$$Q = \text{sign}(\Delta p) \cdot u \cdot K_v \cdot |\Delta p|^x \quad (4)$$

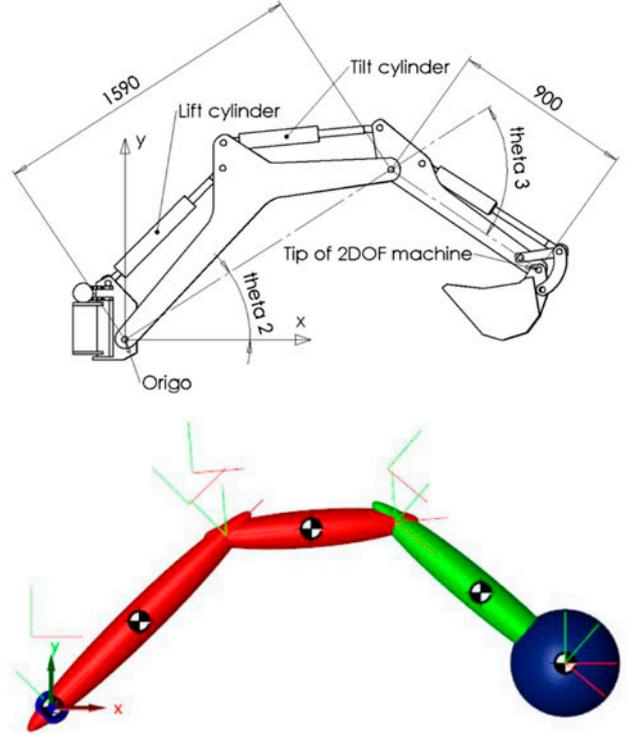


Figure 5. Upper: A sketch of the modeled machine. Lower: Inertia ellipsoids from SimMechanics. Cylinders were not modeled as bodies. Cylinder piston positions, velocities, and generated torques were analytically calculated.

where  $u$  represents the binary control value. On/off valve dynamics were modeled only with a delay and a rate limiter. Parameters are reported in appendix A.

#### 4.3. The DHPMS model

Our digital hydraulic power management system is based on volume models that change according to the crank shaft angle and orifices that connect the chambers to one of the actuator supply lines or to the tank line. When the piston's trajectory is sinusoidal, its position can be solved from equation 5,

$$x(t) = \frac{s}{2} \sin(\omega t + \theta) + \frac{s}{2} \quad (5)$$

where  $\omega$  is angular velocity,  $\theta$  the phase shift, and  $s$  the stroke of the piston. The instantaneous pressure of each cylinder chamber can be integrated from equation 2. The DHPMS was modeled by combining in parallel six piston units with a phase shift of sixty degrees. Supply lines A and B were modeled as static 6.6 dm<sup>3</sup> volumes overall with DHPMS and actuator flows as inputs. In the supply lines, a rigid volume of 5 dm<sup>3</sup> was for added linear pressure control (Heikkilä, et al., 2010) with the hose volumes (1.6 dm<sup>3</sup>) also included in the overall volume. Parameters are given in appendix A and the values are based on the real system.

#### 4.4. DFCU parameterization

The digital valve models we used in simulations were based on an empirical orifice model with diameters set in a series of [0.6, 0.8, 1.1, 1.6, 2.4, 2.5] mm. The diameters were chosen for the series to provide fair resolution, high enough flow, a small variation in step size, and authenticity to real components. A flow characteristic estimate can be calculated by assuming that a square root model is valid. All the control notches were similar. The DFCU containing six valves had 64 states, but state space was pre-reduced to minimize the calculation load in the model-based controller. Pre-reduction was done so that from state 13 on, the smallest valve remained open, and from state 22 on, both the smallest and the second smallest bits remained open (flow series shown in Figure 6). Resolution decreases in higher states, which is a drawback in pre-reducing search space. A full search space is unnecessary and only increases computational costs during the controller time step. Henceforth in this paper, the states of any DFCU are those of reduced search space and thus represent no binary code as they would without pre-reduced search space. For example, state 26 stands now for opening vector [111111], not for [011010] as in a binary system with a full search space.

Valve dynamics were modeled with a delay and a rate limiter such that the opening delay was 6 ms and the closing delay 10 ms. The armature movement time was 4 ms open and 5 ms close, resulting in switching times of 10 ms open and 15 ms close which approximates the values of a certain commercial valve driven with proper power electronics. Because the controller time step must be selected so as to secure safe state transitions before

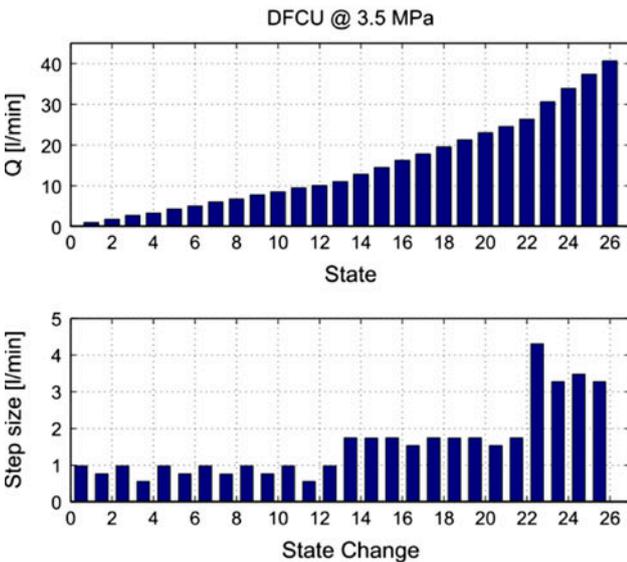


Figure 6. Static characteristics of the DFCUs. Upmost is the flow vs. state, middle the step size vs. state change, and at bottom the characteristic curves of the valves. N.B. Due to pre-reduced search space, state 26 stands now for opening vector [1 1 1 1 1 1], not for vector [1 1 0 1 0], as it would with a full search space.

states are updated with new values, we selected position and valve controller time steps of 20 ms.

## 5. Control Systems

In this study, different control systems were compared. In each simulated case, models representing a mechanical part were kept the same. The digital hydraulic approach has the advantage that system functionality remains in the control code, not in solid mechanical parts. In the emulated proportional system, an ideal pressure compensator was added in the valve model, though compensating functions could also appear in controller sub-models. In all systems, the high level position controller remained the same, but the core level was changed. The valve controllers differ in their mode-choosing logic and pump pressure references.

### 5.1. Upper level position control

The upper level position controller utilized velocity feed-forward and a filtered P-controller in the feedback loop. The reference trajectory contains Cartesian  $x$  and  $y$  coordinates, from which cylinder position references are obtained with inverse kinematics and trigonometry. The cylinder velocity references are discrete derivatives of position references. The filter is tuned to reduce proportional gain at frequencies higher than  $1/3$  of the system's lowest natural frequency. In literature, such a controller is defined as the PT1-type. It is robust against high frequency perturbations but cannot guarantee stability if the lowest natural frequency drops, as it may during a moment of cavitation. (Linjama, et al., 2005).

### 5.2. The discrete proportional valve controller

The simplest digital hydraulic valve controller consists of proportional gains for each control notch such that two control notches are controlled simultaneously, as is done in a traditional analog proportional valve. The laws controlling the notch states are expressed below in Eq. 5:

$$\begin{aligned} u_{PA} &= K_{PA} \cdot (x_{ref} - x), 0 \leq u_{PA} \leq (2^n - 1) \\ u_{AT} &= -K_{AT} \cdot (x_{ref} - x), 0 \leq u_{AT} \leq (2^n - 1) \\ u_{PB} &= -K_{PB} \cdot (x_{ref} - x), 0 \leq u_{PB} \leq (2^n - 1) \\ u_{BT} &= K_{BT} \cdot (x_{ref} - x), 0 \leq u_{BT} \leq (2^n - 1) \end{aligned} \quad (5)$$

where  $u$  is the control signal (state) for a notch,  $K$  is proportional gain, and  $x$  is the cylinder position.  $n$  stands for the number of valves in a DFCU. The  $K$ -gains are fixed and affect the valve similarly to spool geometry in a traditional proportional valve. The  $K$ -gains are tuned to guarantee cavitation free actions with all but the highest load mass. Henceforth, this controller is referred to as the "Emulated Proportional" as it mimics the behaviour of a proportional valve. The following table lists the gain values.

Because this controller contains no pressure compensation functions, an ideal pressure compensator was modelled in the valve model. In reality, this could be done with a digital valve and a traditional pressure compensator. Since the valve parameters are known, flows of the states at a certain pressure differential can be calculated and the states put in order. A selector block is used to find the row that corresponds to the control value, and the binary state vector in that row represents the desired state.

The above controller has limited functionality, but it can still “do the work.” Its functions are simple enough for formal methods to be used to guarantee its functionality, which would be tricky with a complex model-based controller. Parallel use of a simple “safe” controller and a complex optimal “not-guaranteed-to-be-safe” controller was studied in (Huova, et al., 2012), but there the pressure compensator was in the software.

### 5.3. Model-based optimal valve control

High level position control gives a velocity reference as output. The model-based digital valve controller (MBC\_{Orig}, later in results) takes the velocity reference and supply and cylinder chamber pressures as inputs and gives as output optimal control signals to all separate metering edges. One model-based valve controller gives command signals to four DCFUs, all with six on/off valves. Two MBC controllers are needed for the valves of two actuators.

Introduced in (Linjama, et al., 2007), the model-based controller comprises the following main parts:

#### Signal processing

- Pressure signals are measured at 1 kHz
- Pressure signals are low-pass filtered
- The load force estimate is generated from filtered chamber pressure levels

#### The target mode is proposed

- The direction of the desired actuator velocity is defined from the velocity reference. The velocity reference must be over a certain threshold, or the “Stop Mode” is proposed.
- A differential mode to the direction is proposed, if it is feasible within an allowed system pressure range and if chamber pressures do not exceed the highest allowed pressure level.
- Otherwise the inflow/outflow mode is proposed, but only if the required supply pressure and chamber pressures do not exceed the highest allowed pressure level.
- If the load requires a force that exceeds the highest force achievable at the maximum system pressure, the “Stop Mode” is used.
- The required/necessary supply pressure level of a feasible target mode is calculated.

The actual mode is chosen

- The proposed target mode is chosen actual, if it is feasible under the current load and supply pressure conditions.
- If the target mode is not feasible, another feasible mode is used. This could occur, e.g., when the load is operated in the differential mode due to the maximum system pressure, but when the supply pressure has not yet reached the demand level.
- If, temporarily, the load cannot be operated in any mode, the “Stop Mode” is selected.
- Target values for cylinder chamber pressures are calculated based on the actual mode chosen. Cylinder chamber pressures are limited to avoid cavitation or to exceed the maximum allowed pressure level.

#### Optimal control

- Pre-reduced search space of valve opening combinations is determined as a subspace of combinations most likely to produce the desired velocity. This is done separately for each chamber based on candidates likely to match the flow reference.
- For all combinations of determined candidates, model-based estimates of cylinder velocity and chamber pressures are calculated according to the valve model.
- The cost function takes account of the velocity error, chamber pressure error, and the number of necessary valve switchings and energy consumption.
- The combination of the states that minimizes the cost function is selected as current controller output.

#### 5.3.1. Mode-choosing logic for individual actuators

A single actuator has four possible actuator modes. In the following list, letters P, A, B, and T are paired to indicate flow direction in the order of the letters. The control notches used can be seen from the letter pairs. Whenever the letter P, which represents a supply pressure line, is the latter, it indicates the regenerative mode because the flow is towards that port.

- **Mode 0**, Stop mode
- **Mode 1**, Extending inflow/outflow (PA&BT)
- **Mode 2**, Retracting inflow/outflow (PB&AT)
- **Mode 3**, Extending differential (PA&BP)
- **Mode 4**, Retracting differential (AP&PB) (recuperative mode)

With this mode-choosing logic, both actuators are operated separately, and the MBCs produce their own pressure references to the DHPMS controller. With two separate supply pressures, pressure references are used as they are. With only one supply pressure version, the highest pressure reference is used. Unnecessary pressure, if present, is throttled down at the outflow control notch while the pressure differential at the inlet control notch is to remain at the target  $\Delta p$  of the control valve.

### 5.3.2. Mode-choosing logic for two actuators sharing the same supply pressure

Mode-choosing logic can be turned into a cleverer version of the same logic selecting modes for both actuators and producing a pressure level reference to optimize a two-actuator system pressure reference to minimize the power requirement. The modes possible for a single actuator are the same as in the previous case. The trick behind this logic is to calculate the model-based estimates into a power requirement for all possible mode combinations of two actuators. The combination estimated to require least power is then selected. This controller is usable only for single pressure, multi-actuator machines. This controller has 16 different mode combinations, and the combination most likely to minimize the energy consumption estimate is selected. This modification of the Original MPC is henceforth referred to as the MBC\_{TwoActMod}.

### 5.3.3. Mode-choosing logic for a pressurized tank line

If flow is available from the tank line—which requires a pressurized tank line—new modes can be used. The new set of modes is

- **Mode 4**, Extending the pressure side differential (PA&BP)
- **Mode 3**, Extending regenerative inflow/outflow (TA&BP) (recuperating mode)
- **Mode 2**, Extending the tank side differential (TA&BT)
- **Mode 1**, Extending inflow/outflow (PA&BT)
- **Mode 0**, Stop mode
- **Mode -1**, Retracting inflow/outflow (AT&PB)
- **Mode -2**, Retracting the tank side differential (AT&TB) (recuperative mode)
- **Mode -3**, Retracting regenerative inflow/outflow (AP&TB) (recuperative mode)
- **Mode -4**, Retracting the pressure side differential (AP&PB) (recuperative mode)

Note that recuperative capabilities are greatly increased due to the new modes. Modes 4 and -4 were not used because they cause a discontinuous supply pressure reference, which does not improve the efficiency of single actuators. This controller, henceforth marked as the MBC\_{PresTankMod}, has been described in detail in (Huova 2012).

### 5.4. Pressure References

The DHPMS has two, individually pressurizable actuator lines. In simulations where only one pressure reference was used, the same reference was used for both supply lines. When two supply pressures were used, references were directly routed to corresponding controllers. Flow at the supply ports was estimated with the valve

controller, and the estimate was used as a feed-forward signal in the controller.

## 6. Simulations

### 6.1. Test cases

The circular reference trajectory was driven with five different loads and five different velocities, and a total of 25 combinations were simulated for five different controllers. The fastest trajectory was set so as to fulfill the maximum flow demand with the six-piston DHPMS prototype. DHPMS parameterization was based on a real machine.

Load masses 0, 75, 150, 225, and 300 kg, and trajectory times 10, 15, 20, 25, and 30 seconds are used. Peripheral velocity is kept constant, and no start- or end-smoothing functions are used. This result in somewhat jerky behavior at end and start instances, but it simplifies trajectory generation. Also jerks at the start and end are insignificant in terms of energy efficiency. Furthermore, it is important that the system maintains stability also in case of sudden changes in references.

The circular trajectory had the same start and end point, and because the load mass was kept constant during simulation, no actual work was done. At a certain time step, output power was then calculated from the product of the cylinder net force, which is the sum of the products of pressures and areas multiplied by velocity. Hydraulic input power was obtained from the pressures and flows of the DHPMS and mechanical input power from the angular velocity and torque of the DHPMS. Work from the power was obtained by a trapezoidal integration method.

### 6.2. Results

The following Figures 8 through 14 illustrate the 15 s trajectory and 300 kg load mass and contain matrices of subplots with the left column for cylinder 1, the lift cylinder, and the right column for cylinder 2. The topmost row displays cylinder position and position reference. The second row is for velocity and velocity reference, the output of the high level position controller. The third row for Mode has signals for both target and actual modes. As the emulated proportional valve controller used only inflow/outflow modes, no mode selection logic was needed. The DFCU states in the fourth row are control notch openings. The following rows stand for chamber pressures and supply pressure and its reference. The last row indicates hydraulic power consumed by the line and the output power of the actuator. Negative output power means overrunning load, and negative input power occurred if the DHPMS was motoring. Input work for a line is shown in the middle of the subplot. The last graph has bars for total input, hydraulic input, and output energies summed over all simulated cases with varying trajectory time and load mass.

Table 2. Virtual spool geometries for control notches. The value defines the state used if the position error is one meter.

|          | Cyl 1 [1/m]   | Cyl 2 [1/m] |    |     |
|----------|---|-------------|----|-----|
| $K_{PA}$ | 220   | 220         |    |     |
| $K_{AT}$ | 180   | 180         |    |     |
| $K_{PB}$ | 180 </tr <tr> <td><math>K_{BT}</math></td> <td>86</td> <td>180</td> </tr> | $K_{BT}$    | 86 | 180 |
| $K_{BT}$ | 86  | 180         |    |     |

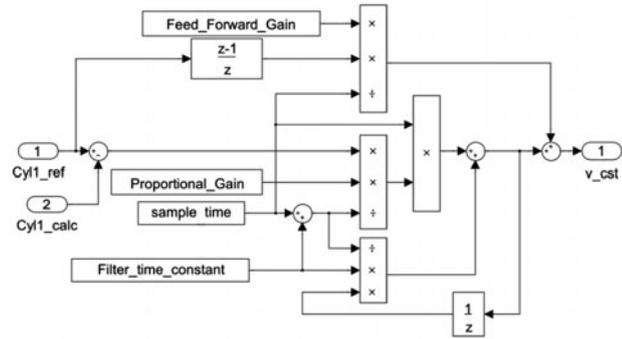


Figure 7. High Level Position Controller. Discrete PT1 type with velocity-feed-forward.

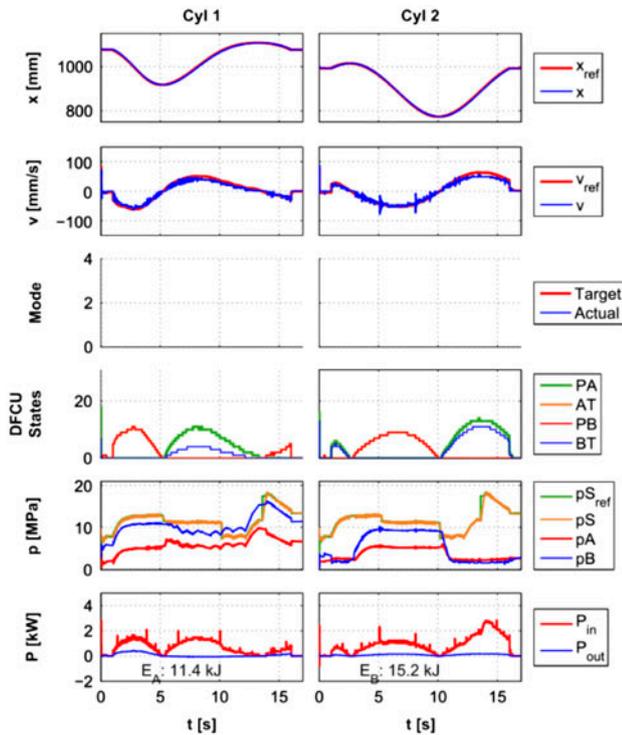


Figure 8. The Emulated Proportional valve with common supply pressure for both actuators. Ideal pressure compensator models were used.

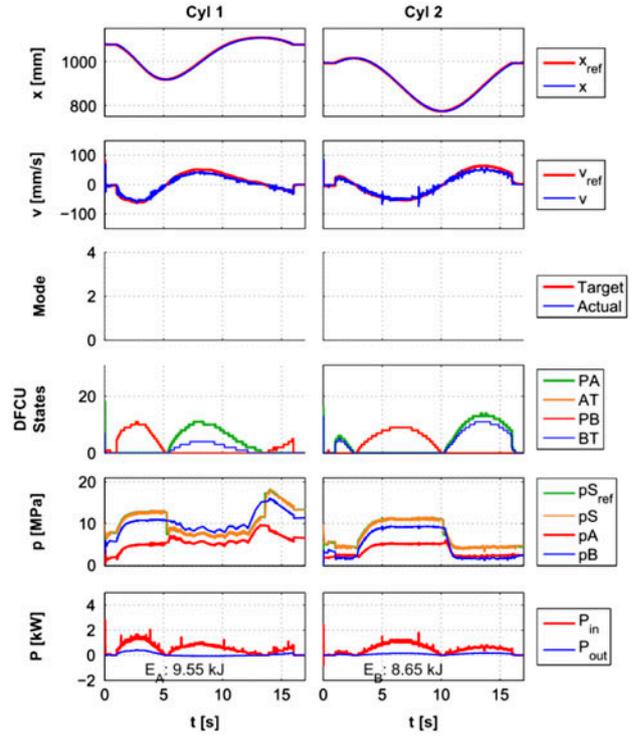


Figure 9. The Emulated Proportional valve with two separate supply pressures. Ideal pressure compensator models were used.

## 7. Analysis Of The Results

The results in Figure 15 show that the most efficient case is that of separate pressures and use of a pressurized tank line. The biggest losses occur with the least intelligent controller with only one supply pressure. All losses are throttling losses. Loss caused by cylinder friction is considered output work, because the trajectory is such that no actual work is done. Because friction forces and leakage were not modeled in the DHPMS, its losses are then only throttling losses in the port valves. The prototype's total efficiency measured 70 – 85%, and it remained at this level during the pumping, motoring, and transforming functions (Heikkilä, et al., 2010). Simulated results show similar total efficiency, indicating the DHPMS's most significant losses as being throttling losses.

The supply pressure signals show that at the start and end, actuator 1 demanded higher pressure and in the middle actuator 2, creating pressure compensation losses in the actuator with too much available pressure. The two-supply pressure system required no pressure compensation at the inlet notch.

During moments of cavitation (occurring only with the highest load), the pressure differential over the valve dropped to zero, causing a velocity error. In addition, the cavitating cylinder became less stiff, as seen in equation 6.

$$K_h = \frac{A_A^2 \cdot B_{Aeff}}{(x \cdot A_A + V_{Ao})} + \frac{A_B^2 \cdot B_{Beff}}{((L - x) \cdot A_B + V_{Bo})} \quad (6)$$

where  $A$  stands for area,  $B$  for bulk modulus,  $m$  for reduced mass,  $L$  for cylinder stroke, and  $V$  for dead volumes (Merritt, 1967). The equation states that the effective bulk modulus of one of the chambers drops virtually to zero, as occurred during the cavitation of one chamber, and therefore the stiffness is significantly reduced because one term becomes zero. Because the stiffness of one cylinder affects the system's natural frequencies, the system may at some operation point become unstable. With no cavitation present, the PT1-controller is tuned based on the principle of the system's lowest natural frequency. The controller is robustly stable against high frequency perturbations but does not guarantee stability if the system's natural frequency drops, as can be seen in the simulation results on the Emulated Proportional cases, where cylinder 1 chamber A cavitates for some time. During cavitation, close to mid-simulations, the system turned momentarily unstable, as evidenced by the strongly oscillating chamber pressure. A model-based controller can actively control chamber pressures so that chambers will not cavitate.

The more complex mode-choosing logic of the controller  $MBC_{\{TwoActMod\}}$  seemed to have no positive effect on holistic results. On further inspection, some simulation results show that benefits, if any, are case sensitive. Our hypothesis was that the  $MBC_{\{TwoActMod\}}$  should have resulted in less losses than the  $MBC_{\{Orig\}}$  with one supply pressure but not less than the  $MVC_{\{Orig\}}$  with two supply pressures. The hypothesis seems correct, though there is no significant difference in the net energies used between the single pressure cases of these controllers. This together with the fact that the controller comes with more complex mode-choosing logic does not support using this method for this application and trajectory.

An optimal supply pressure in a Model-Based Controller system makes it possible to use modes more effectively. Compare now the modes on results of actuator 2 in the case of the  $MBC_{\{Orig\}}$  at time of about 3 s (Figures 10 and 11). With only one supply pressure, the mode-choosing logic proposes that due to the load force the regenerative differential mode could be used. Because the supply pressure at the moment is too high, that mode cannot be chosen as the actual. With two supply pressures, pressures are set individually. The pressure is thus set to a correct level, and the target mode can be selected as the actual. The same functionality can also be seen in the  $MBC_{\{PressTankMod\}}$  with actuator 1 at time 5 to 10 s (Figure 13 and 14). The proposed target mode is regenerative inflow/outflow, but with only one supply pressure, this mode cannot be selected as actual, and thus normal inflow/outflow is used instead. With two supply pressures, pressures are set by the demands of both actuators, and target modes can be chosen as actual more often.

The control algorithm of the DHPMS worked well. Comparing it with previous DHPMS controllers, we found that an accurate velocity estimate obtained from a

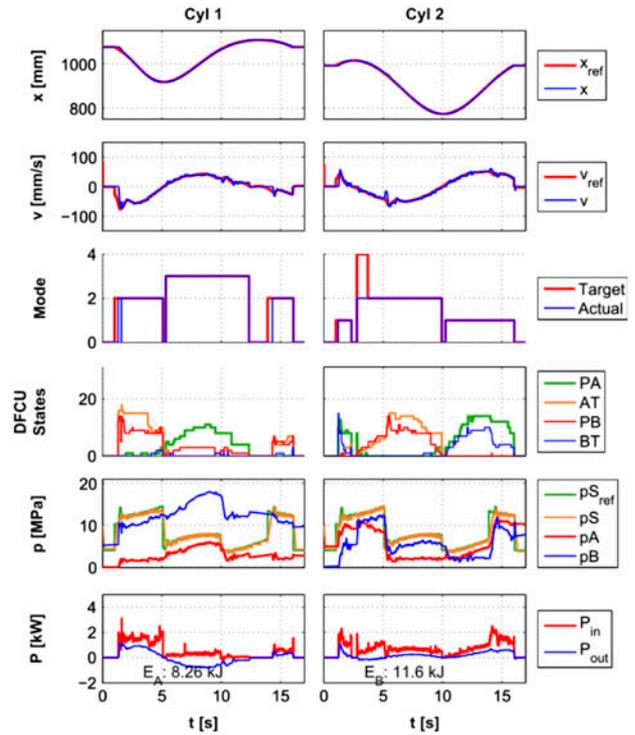


Figure 10. The  $MBC_{\{Orig\}}$  with common supply pressure

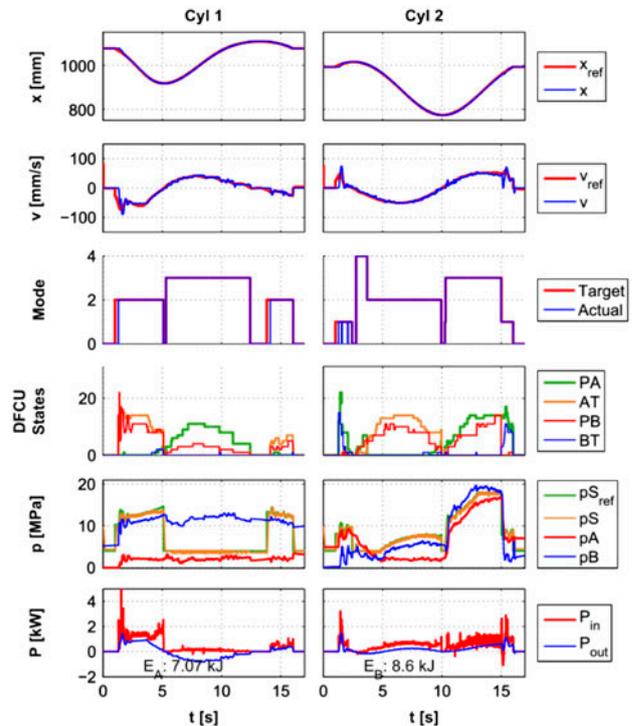


Figure 11. The  $MBC_{\{Orig\}}$  with two separate supply pressures.

model-based valve controller can be used to calculate a flow estimate for the DHPMS controller. As a result, a smoother pressure signal could be obtained by controlling only pressure by the feedback of the measured

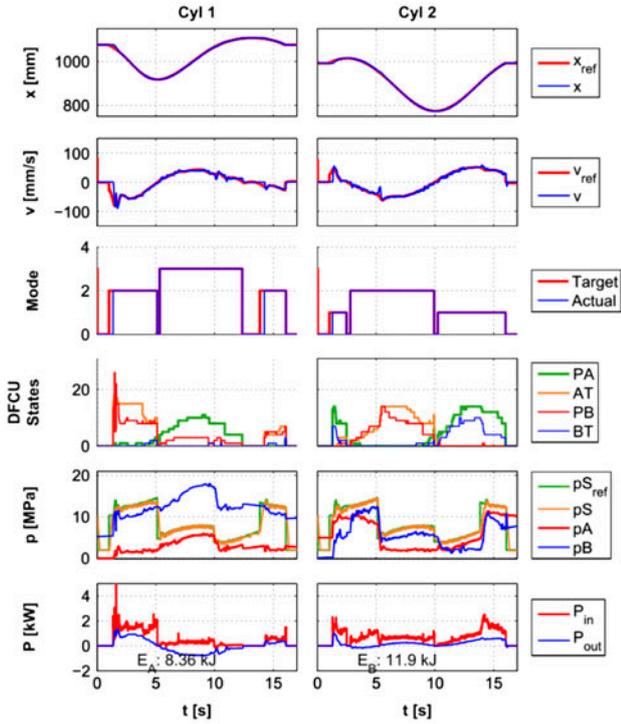


Figure 12. The  $MBC_{\{TwoActMod\}}$  with common supply pressure

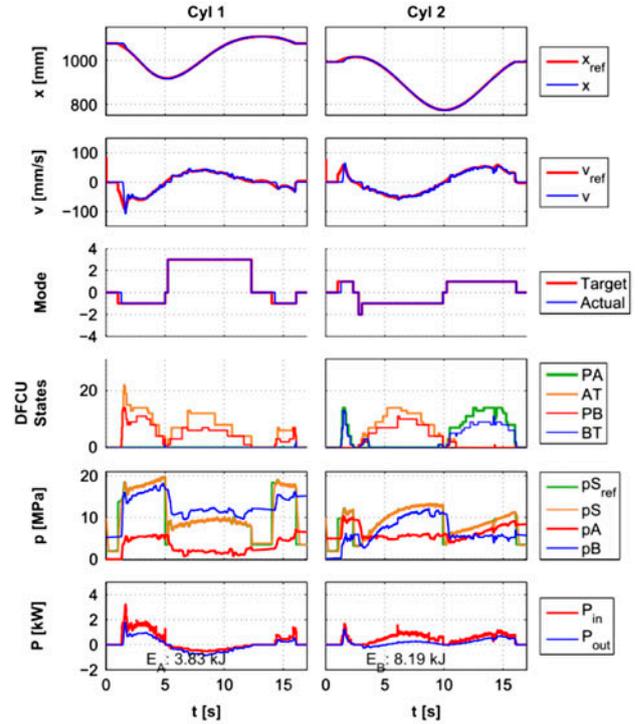


Figure 14. The  $MBC_{\{PresTankMod\}}$  controller using a pressurized tank line with two separate supply pressures.

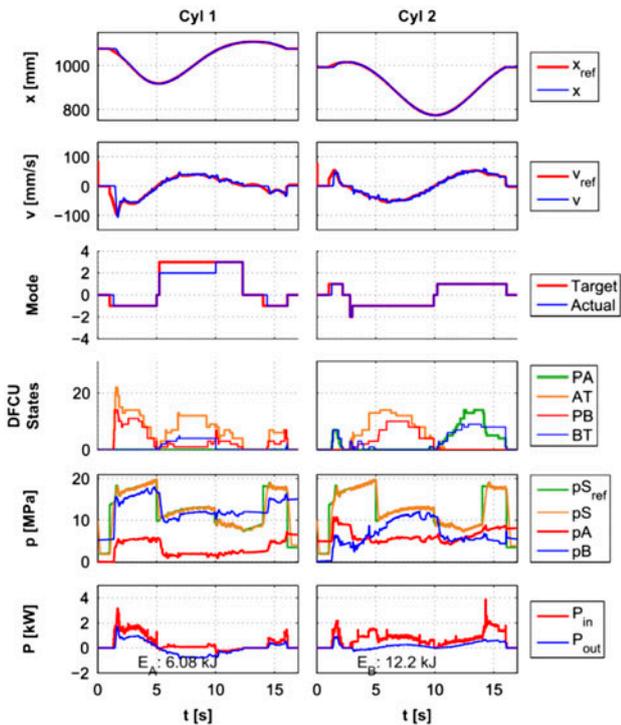


Figure 13. The  $MBC_{\{PresTankMod\}}$  controller using a pressurized tank line with common supply pressure.

pressure signal. The flow estimate is thus used in a fashion similar to the general use of feed-forward in controllers.

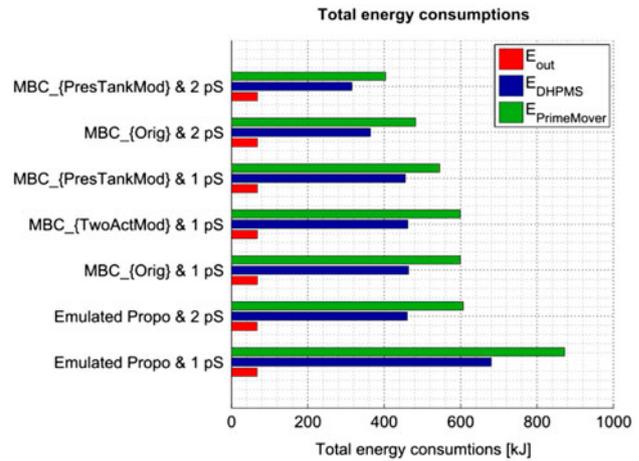


Figure 15. Total energy consumptions of all 25 test cases.  $E_{DHPMS}$  is hydraulic energy taken from the ports of the DHPMS.  $E_{PrimeMover}$  is the amount of energy taken from the prime mover axle.

### 8. Conclusion

The basic principle of digital hydraulics is to use simple components and intelligent control, i.e., to replace hydro-mechanical controllers and variable value components with simple and reliable and universal hydraulic components and a sophisticated controller. In this paper, those simple components were pumping small pistons, on/off valves, and large actuator pistons (cylinders). Controllers were based on a hierarchical controller with

a position controller at the highest level and various algorithms and functions at the core level to decide on when to switch a particular set or sequence of on/off valves.

Because digital hydraulic components are universal, only the control system needs to be changed to produce various types of systems. The same hardware models (except pressure compensators) and different software were used and system behavior was studied. Energy efficiency was our primary interest. The same set of work trajectories was driven on the same set of load masses. Results show that maximized efficiency requires a pressurized tank line and separate supply pressures; furthermore, two-actuator modification of the mode-choosing logic is not practical.

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This research was supported by the Academy of Finland (Grant No. 139540). Researchers Karvonen and Huova are also sponsored by the TUT President's doctoral program.

### Notes on contributors



Matti Karvonen (Born 20<sup>th</sup> July 1984) started as a research assistant at the Tampere University of technology (TUT) at fall of 2007 and took degree of M.Sc. (Tech.) in June of 2009. Since then he has been working as a researcher in the digital hydraulics research group. Topic of his ongoing PhD (Tech.) studies are supported by the doctoral programme of TUT's president. Main research area considers digital hydraulic solutions in MIMO-systems. Another major research area has been miniature valves.



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Matti Linjama Adjunct Professor Matti Linjama has worked 18 years at TUT and his major research topic has been digital hydraulics since 2000. He is leader of the digital hydraulics research group and has published 50+ scientific publications about the topic.

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## Appendix A

Simulation parameters in SI-units:

```
% Cylinders parameters
Cyl.Dd = [63e-3,36e-3];
Cyl.DeadVols = [0.05e-3,0.05e-3];
Cyl.HoseVols = [0.05e-3,0.05e-3];
Cyl.Oil_B = 1000e6;
Cyl.Hose_B = [300e6 300e6];
Cyl_1.stroke = 1000e-3;
Cyl_1.x_0 = 600e-3; %Cylinder min length
Cyl_2.stroke = 700e-3;
Cyl_2.x_0 = 400e-3; %Cylinder min length
%Static friction force
Fric.Fs = 800;
%Coulombian force
Fric.Fc = 750;
%Viscous coeicient
Fric.b = 300;
%Velocity of minimum friction
Fric.vmin = 0.02;
%Tanh-steepness coeicient;
Fric.K = 4000;

% Digital valve Dynamic parameters
%Delays & movement time
Valve.delay_open = 6e-3;
Valve.delay_close = 10e-3;
Valve.movtime_open = 4e-3;
Valve.movtim_close = 5e-3;
```

```
% DHPMS Model parameters
%Dampening volumes
DPMT.Vol_1 = 5e-3;
DPMT.Vol_2 = 5e-3;
%Volumes of supply lines
DPMT.V_hose1 = 4*pi/4*(3/4*25.4e-3)^2;
DPMT.V_hose2 = 4*pi/4*(3/4*25.4e-3)^2;
%Effective bulk modulus
const.B_e = 1000e6;
%Tank pressure
const.p_T = 1e6;
%Rotational speed of the prime mover
Pump.n = 1000; % [rpm]
% Number of pistons
piston.N = 6;
% Phase shift of pistons
piston.ph_rad = 2*pi / piston.N * (0:piston.N-1); % Phase shift of pistons
piston.S = 16e-3; % Stroke
piston.d = 20e-3; % Diameter
piston.A = (pi*piston.d^2)/4; %Area
piston.V_disp = piston.S*piston.A;
piston.V_0 = 20e-6; % Dead volume
%DHPMS valve parameters:
valve.nominal_pressure = 3e6;
valve.nominal_flow = 55/6e4;
valve.delay = 0.001;
valve.move_time = 0.0005;

% Mechanism:
%Body masses: 80 kg & 40 kg.
%Inertias calculated by equation of solid rod having diameter of 100 mm. Load inertia eye ( )
%In following figure, units are in [mm]
```

