

# Fault-Tolerant Control of a Multi-Outlet Digital Hydraulic Pump-Motor

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Fault tolerance is the most important feature in safety-critical applications, including aircraft flight controls, nuclear systems, and medical devices, but it is a desirable property of any mechatronic system. In this paper, the fault tolerance of a multi-outlet digital hydraulic pump-motor is studied. This machine has actively controlled on/off valves to independently connect each piston to the tank or one of its outlets. Furthermore, the pump-motor can control an actuator directly without having directional control valves in the system; thus, the on/off control valves of the machine are the most vulnerable components of failure. A valve can either become jammed on (not able to close) or off (not able to open), whether the fault is electrical or mechanical. The effect of a defective valve is studied through simulations, and a method for fault compensation is proposed with a control algorithm adapted for each fault case. The simulations and experimental results show that the valve faults can be effectively compensated for by reconfiguring the software. Only slight degradation in the control performance can be expected.

**Keywords:** Digital hydraulic power management system, energy efficiency, fault tolerance, motor, pump, transformer

**Target audience:** Digital Hydraulics, Mobile Hydraulics

## 1 Introduction

### 1.1 Fault-tolerant hydraulic systems

Safety-critical applications, such as aircrafts and nuclear power plants, have redundant hydraulic systems. For example, Boeing airplanes have three independent hydraulic supply systems for their flight actuators: left, right, and centre /1/. Either the left or the right system can be replaced by the centre system if hydraulic power is lost. Even if all three systems fail to produce hydraulic pressure, a ram air turbine-driven pump can ensure a safe landing.

Fault-tolerant remote handling operations at ITER (an international nuclear fusion research and engineering megaproject) were studied in /2/. The water hydraulic manipulator under investigation has a redundant servo valve system. A faulty valve can be isolated from the hydraulic circuit, and the operation is continued using the redundant valve.

Usually, the hydraulic systems of mobile working machines are considered less safety-critical. However, component failures may cause economic losses for the owner if the fault prevents the machine from operating. Digital hydraulics is a solution toward more reliable machines, as it utilizes simple on/off components. Moreover, the parallel connected components make digital hydraulics configurable. Hence, a single component failure can be compensated for with an intelligent control algorithm. The fault tolerance of a digital flow control unit (DFCU) was studied in /3/. A basic principle of compensating for a single valve fault is to reconfigure the controller. In case of an off-jammed valve, the controller excludes all DFCU states related to the faulty valve. On the other hand, the distributed valve system allows also compensation for the effect of an on-jammed valve: the additional volume flow can be cancelled by using another flow path.

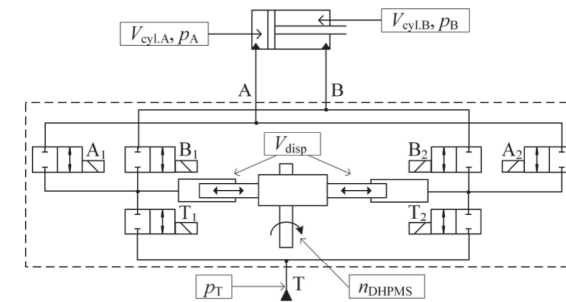


Figure 1. Simplified system diagram: connection between a DHPMS and a double-acting cylinder.

In digital displacement machines, the control valves are switched on and off at high frequency; thus, it is probable that the valves fail in process of time /4/. Therefore, fault tolerance of a digital hydraulic power management system (DHPMS) is worth studying. This research focuses on compensation of a single valve fault in a multi-outlet pump-motor. Methods for identification of the valve faults are not considered in this study. However, similar approaches (based on electrical and pressure measurements) that are presented in /5, 6/ can also be applied for the digital pump-motors.

### 1.2 Digital power management

Digital displacement machines have increasingly been studied in recent years. Artemis Intelligent Power Ltd is a pioneer of the digital pumping technology /7, 8/ and they have interested other institutes to study and apply the technique as well /9, 10, 11, 12/. A multi-outlet digital hydraulic pump-motor, the DHPMS, has actively controlled on/off valves to connect the pumping/motoring pistons independently to the tank or one of the outlets /13/. Thus, the volume flows at the outlets are controlled with the required number of pistons while the remainder are left to idle. This piston-by-piston control of digital machines also enables good efficiency at partial displacement; thus, the overall efficiency of a digital pump-motor can reach over 0.9 for the entire operating range /14/. Additionally, the multi-outlet solution enables the hydraulic energy to be transferred from one outlet to another without reducing the efficiency, as verified by the first prototype /15/. As both the pressure and flow can be altered at the outlets, the DHPMS can operate as a hydraulic transformer.

The DHPMS can be used for controlling the system pressures, whereas the actuators are controlled by directional valves. The ability to control several outlet pressures is profitable, especially in multi-actuator systems, because the supply pressure of each actuator can be set according to the load pressures independently of one another. Thus, the pressure-matching losses that are typical for traditional load-sensing (LS) hydraulics can be avoided /16/. Alternatively, the DHPMS can directly actuate a hydraulic cylinder by controlling the volume flows at the outlets. This approach comes with small hydraulic losses and the system is capable of full-scale energy recovery /17/. Moreover, a hybrid system can be implemented by attaching a hydraulic accumulator to one outlet of the DHPMS. The accumulator can be used as an energy source/sink, and allows the prime mover to be a smaller size /18/.

However, the good performance and efficiency of the DHPMS places high demands on the on/off control valves, which must have 1) high durability, 2) leakage-free construction, 3) fast response time with low variance, 4) high flow capacity, and 5) low electric energy consumption. In addition, the geometrical displacement of each individual pumping/motoring piston must be small to avoid extensive flow and torque oscillations. Therefore, a large number of pistons is needed or a high rotational speed must be used to achieve good controllability and sufficient volume flow /19/.

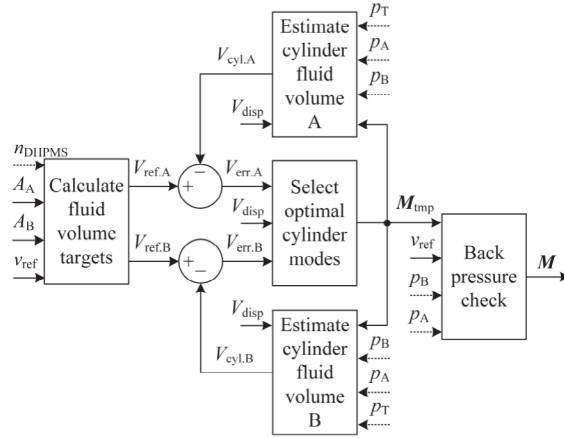


Figure 2. Control block diagram for the direct displacement control using the DHPMS.

### 1.3 Direct displacement control of an actuator

Displacement control was used to control the lift cylinder of a small excavator boom directly using a 6-piston DHPMS in /17/. Pumping and motoring modes (T, A or B) are selected for each DHPMS piston once per revolution. Hence, six mode-selection instants occur during one revolution, as the pumping and motoring modes are selected simultaneously for a pair of pistons. The selected modes realize when the corresponding valve (T, A or B) for each piston is opened in the beginning of the pumping/motoring stroke. Figure 1 shows the connection between the pumping/motoring pistons and the actuator via on/off control valves.

A control block diagram for the mode selection is shown in Figure 2. Targets for the cylinder fluid volumes A and B are calculated according to the velocity reference of the actuator ( $v_{ref}$ ) and effective piston areas ( $A_A$  and  $A_B$ ). The cumulative volume for the piston side is

$$V_{ref,A} = \int v_{ref} \cdot A_A \quad (1)$$

whereas the equation for the rod side volume is

$$V_{ref,B} = \int -v_{ref} \cdot A_B \quad (2)$$

Additionally, extrapolation is used to estimate the target volumes at the end of the pumping/motoring cycle. Thus, a faster response can be achieved by assuming that the velocity reference and rotational speed are unchangeable between the sequential mode-selection instants. Fluid volume errors at the outlets ( $V_{err,A}$  and  $V_{err,B}$ ) are further determined as differences between the target volumes and volume estimates ( $V_{cyl,A}$  and  $V_{cyl,B}$ ).

The resulting errors in fluid volumes are calculated for every mode combination, as stipulated in Table 1 where the valves  $T_1$ ,  $A_1$ , and  $B_1$  are related to the pumping piston and the valves  $T_2$ ,  $A_2$ , and  $B_2$  to the motoring piston respectively. For example,  $V_{err1}$  is the total fluid volume error for the mode combination [T, T]; hence, neither the pumping nor the motoring pistons are connected to the outlets. In the case of  $V_{err2}$ , the pumping piston is connected to outlet A yielding that the overall fluid volume at that outlet is increased by the geometrical piston displacement  $V_{disp}$ , whereas the error at outlet B remains unchangeable i.e. the mode combination is [A, T]. It can be seen that in normal operation, contemporary pumping and motoring to the same outlet is prohibited by defining the resulting error: “infinite”. Finally, the mode combination,  $M_{tmp} = [M_{P,tmp}, M_{M,tmp}]$ , which minimizes the total fluid volume error, is preselected. For example, if the choice to pump to outlet B and simultaneously motor from the outlet A ( $V_{err7}$ ) results in the smallest combined fluid volume error at the outlets, the mode combination  $M_{tmp} = [B, A]$  is selected. The compressibility of fluid in the DHPMS piston chambers is considered

when calculating the change in cumulative fluid volume estimates ( $V_{cyl,A}$  and  $V_{cyl,B}$ ) according to the decided modes. For example, if the piston is filled from the tank and it is decided to pump to outlet A, the new fluid volume estimate for that outlet is calculated as  $V_{cyl,A}(k) = V_{cyl,A}(k-1) + V_{disp} - C_h \cdot (p_A(k) - p_T(k))$ , where  $C_h$  is determined hydraulic capacitance for the pumping cylinder.

$M_{pump}$	$M_{motor}$	Total fluid volume error	Valve states ( $T_1, A_1, B_1, T_2, A_2, B_2$ )
T	T	$V_{err1} =  V_{err,A}  +  V_{err,B} $	(1, 0, 0, 1, 0, 0)
A	T	$V_{err2} =  V_{err,A} - V_{disp}  +  V_{err,B} $	(0, 1, 0, 1, 0, 0)
T	B	$V_{err3} =  V_{err,A}  +  V_{err,B} + V_{disp} $	(1, 0, 0, 0, 0, 1)
A	B	$V_{err4} =  V_{err,A} - V_{disp}  +  V_{err,B} + V_{disp} $	(0, 1, 0, 0, 0, 1)
B	T	$V_{err5} =  V_{err,A}  +  V_{err,B} - V_{disp} $	(0, 0, 1, 1, 0, 0)
T	A	$V_{err6} =  V_{err,A} + V_{disp}  +  V_{err,B} $	(1, 0, 0, 0, 1, 0)
B	A	$V_{err7} =  V_{err,A} + V_{disp}  +  V_{err,B} - V_{disp} $	(0, 0, 1, 0, 1, 0)
A	A	$V_{err8} = \infty$	(0, 1, 0, 0, 1, 0)
B	B	$V_{err9} = \infty$	(0, 0, 1, 0, 0, 1)

Table 1. Volume error calculation for different mode combinations in normal operation.

The final pumping and motoring modes,  $M = [M_P, M_M]$ , are decided after a back-pressure check. A non-load chamber of the actuator is also kept pressurized for the system stiffness and to avoid cavitation. Based on the former research, the changes in actuator-loading and the leakages through the DHPMS can also be effectively compensated for by controlling the back-pressure /19/.

The actuator pressures have user-defined limits and the pressure is raised or lowered according to the rules defined in Table 2. In general, the pressure can be raised by pumping to the expanding chamber or by restricting flow from the contracting chamber. For example, if the cylinder (actuator) pressure goes below the user-defined minimum pressure while the cylinder is extending ( $v_{ref} > 0$ ) and  $p_{back} = p_A$ , motoring from outlet B is restricted by selecting  $M_M = T$ . The rules for lowering the pressure are the opposite. For example, if the cylinder pressure exceeds the user-defined maximum pressure while the cylinder is stationary or extending ( $v_{ref} \geq 0$ ) and  $p_{back} = p_A$ , motoring is enforced from outlet B by selecting  $M_M = B$ .

Case	Condition	Decision
low	$p_{back} = p_B$ and $v_{ref} \geq 0$	$M = [A, M_{M,tmp}]$
low	$p_{back} = p_B$ and $v_{ref} < 0$	$M = [M_{P,tmp}, T]$
low	$p_{back} = p_A$ and $v_{ref} > 0$	$M = [M_{P,tmp}, T]$
low	$p_{back} = p_A$ and $v_{ref} \leq 0$	$M = [B, M_{M,tmp}]$
high	$p_{back} = p_B$ and $v_{ref} > 0$	$M = [T, M_{M,tmp}]$
high	$p_{back} = p_B$ and $v_{ref} \leq 0$	$M = [M_{P,tmp}, A]$
high	$p_{back} = p_A$ and $v_{ref} \geq 0$	$M = [M_{P,tmp}, B]$
high	$p_{back} = p_A$ and $v_{ref} < 0$	$M = [T, M_{M,tmp}]$

Table 2. Final mode selection after a back-pressure check in case of too-low or too-high pressure.

In this study, the controller is implemented using MATLAB/Simulink, whereas the control algorithm is processed by dSPACE DS1005 PPC board. The sampling rate of 20 kHz is used to ensure accurate valve timing. A DS4001 digital I/O board controls the valves.

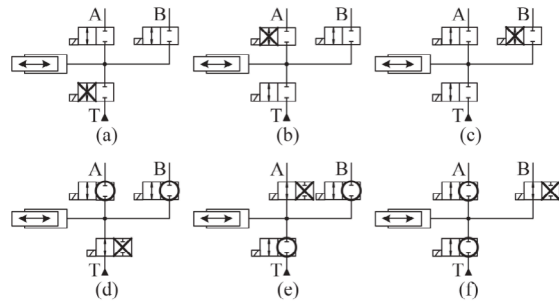


Figure 3. Fault compensation: the valve position that cannot be used (X) and the position used after fault compensation (O).

## 2 Proposed fault-tolerant control method

### 2.1 Effect of an uncompensated valve fault

Two kinds of control-valve faults are considered: a valve is jammed fully in either the on-position (not able to close) or the off-position (not able to open). Additionally, it is assumed that only one defect can occur at a time. Hence, there are six different fault cases for the studied DHPMS:

- Inlet T-valve jammed off
- Outlet A-valve jammed off
- Outlet B-valve jammed off
- Inlet T-valve jammed on
- Outlet A-valve jammed on
- Outlet B-valve jammed on

An off-jammed valve causes the DHPMS chamber pressure to rise excessively during the pumping stroke. A pressure-relief valve should therefore be installed in each pumping piston of the DHPMS. During the motoring stroke, an off-jammed valve prevents the chamber from being filled with oil and causing cavitation. If not compensated, an off-jammed outlet valve also affects the actuator behaviour due to incomplete pumping or motoring strokes. Jamming on again causes energy-wasting cross-flow through the valves and can make the system uncontrollable.

### 2.2 Fault compensation

Generally, an off-jammed valve means that the inlet or outlet related to the valve can no longer be used. On the other hand, an on-jammed valve means that the inlet or outlet related to the valve must be used constantly. Hence, compensation of an off-jammed valve rules out the port of the defective valve, whereas compensation of an on-jammed valve closes the remaining ports in the related piston. The valve faults also decrease the effective geometrical displacement of the DHPMS.

The studied valve faults and actions for fault compensation are shown in Figure 3. A cross (X) on top of a control valve indicates the flow path that cannot be used, whereas a circle (O) indicates the forced valve position after fault compensation. Figure 3a shows an off-jammed T-valve. Thus, outlets A and B can still be used for that piston. Similarly, inlet T and outlet B can be used in the case of an off-jammed A-valve (Figure 3b), and an off-jammed B-valve prevents operation of that outlet (Figure 3c).

$M_{pump}$	$M_{motor}$	Total fluid volume error
T	T	$V_{err1} =  V_{err,A}  +  V_{err,B} $
A	T	$V_{err2} = \infty$
T	B	$V_{err3} =  V_{err,A}  +  V_{err,B} + V_{disp} $
A	B	$V_{err4} = \infty$
B	T	$V_{err5} =  V_{err,A}  +  V_{err,B} - V_{disp} $
T	A	$V_{err6} =  V_{err,A} + V_{disp}  +  V_{err,B} $
B	A	$V_{err7} =  V_{err,A} + V_{disp}  +  V_{err,B} - V_{disp} $
A	A	$V_{err8} = \infty$
B	B	$V_{err9} = \infty$

Table 3. Volume error calculation for the mode combinations in case of an off-jammed A-valve in a pumping piston.

$M_{pump}$	$M_{motor}$	Total fluid volume error
T	T	$V_{err1} = \infty$
A	T	$V_{err2} =  V_{err,A} - V_{disp}  +  V_{err,B} $
T	B	$V_{err3} = \infty$
A	B	$V_{err4} =  V_{err,A} - V_{disp}  +  V_{err,B} + V_{disp} $
B	T	$V_{err5} = \infty$
T	A	$V_{err6} = \infty$
B	A	$V_{err7} = \infty$
A	A	$V_{err8} =  V_{err,A}  +  V_{err,B} $
B	B	$V_{err9} = \infty$

Table 4. Volume error calculation for the mode combinations in case of an on-jammed A-valve in a pumping piston.

Figure 3d shows an on-jammed tank valve. In this case, A- and B-valves must be kept closed to prevent cross-flow; hence, the piston is continuously connected to the tank. Correspondingly, the piston is constantly pumping to and motoring from outlet A when that valve is jammed on (Figure 3e). On the other hand, T- and A-valves are always kept closed in case of an on-jammed B-valve (Figure 3f).

### 2.3 Controller adaptation

In order to minimize the influence of a valve fault on system performance, the control algorithm must adapt to the limited controllability. A certain fault has an effect on the optimal mode selection, as well as on the back-pressure control. The rules for the fluid volume error calculation in case of an off-jammed A-valve in a pumping piston are shown in Table 3. As the fluid cannot be pumped to outlet A, each related mode combination results in “infinite” volume error. Additionally, simultaneous pumping and motoring for outlet B is prohibited, as the inlet can be used for idling. Hence, the best mode combination is selected from among the remaining five choices. Correspondingly, mode combinations [T, A], [B, A], [A, A], and [B, B] would be banned if a motoring stroke was considered. The rules for compensate an off-jammed valve are also similar in the case of valves T and B. However, mode combinations [A, A] and [B, B] are exploitable when the T-valve is jammed off. Between these two combinations, the one connected to the back-pressure is chosen when idling.

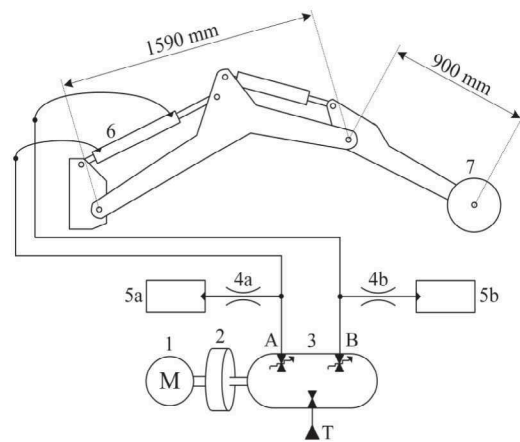


Figure 4. The studied system: an excavator boom with direct lift cylinder actuation using the DHPMS.

Table 4 shows the eligible mode combinations when the A-valve of the pumping piston is jammed on. As valves T and B must be kept closed, the modes minimizing fluid volume errors have to be selected from among the combinations [A, T], [A, B], and [A, A]. The last one is used instead of normal idling [T, T]. In this case, the effect of unavoidable pumping can be compensated for by motoring simultaneously, which results in zero volume flow at the outlet. For the motoring stroke, only the mode combinations [T, A], [B, A], and [A, A], respectively, would be allowed.

Similar rules also apply to the back-pressure control, as pumping to or motoring from an outlet with an off-jammed valve cannot be executed. In addition, an on-jammed T-valve prevents the pressure control of both outlets.

### 3 Test system: a small excavator boom

The measurement system is the same as that introduced in /17/, in which the first experimental results of a direct displacement control of a boom using the DHPMS are presented. The hydraulic system actuating the boom lift cylinder (Ø63/36–500) is shown in Figure 4. A six-piston DHPMS (3) with a geometrical displacement of 30 cm<sup>3</sup> is driven by an induction motor (1), which has a rated power of 18.5 kW at a rotational speed of 1450 r/min. However, a rotational speed of 850 r/min is used throughout the tests. The speed is set by a frequency converter.

A flywheel (2) having inertia of 0.15 kgm<sup>2</sup> is installed in the motor axis smooths the torque pulsation caused by the DHPMS. The outlets of the DHPMS connect to the lift cylinder (6) ports via supply line hoses having the length of about 2.8 m. In addition to these hose volumes of about 1.4 l in each line, the rigid wall volumes (5a and 5b) of 5 l are used to increase hydraulic capacitance. The volumes are sized such that the pressure fluctuation caused by the piston unit does not exceed 1 MPa. The flows of the volumes are throttled using needle valves (4a and 4b) to increase the system damping. The orifices have a fixed setting: the volume flow equals 3.4 l/min when the pressure difference is 0.5 MPa.

The load mass (7) at the boom tip is 200 kg and the tilt cylinder is hydraulically locked close to its minimum length, yielding a natural frequency of about 2 Hz for the system. Prototype valves (18 in total) having response times of about 1.5 ms are used to control the DHPMS pistons. Relatively small flow capacity of the valves (23 l/min @ 0.5 MPa) decreases the mechanical-hydraulic efficiency of the DHPMS /15/.

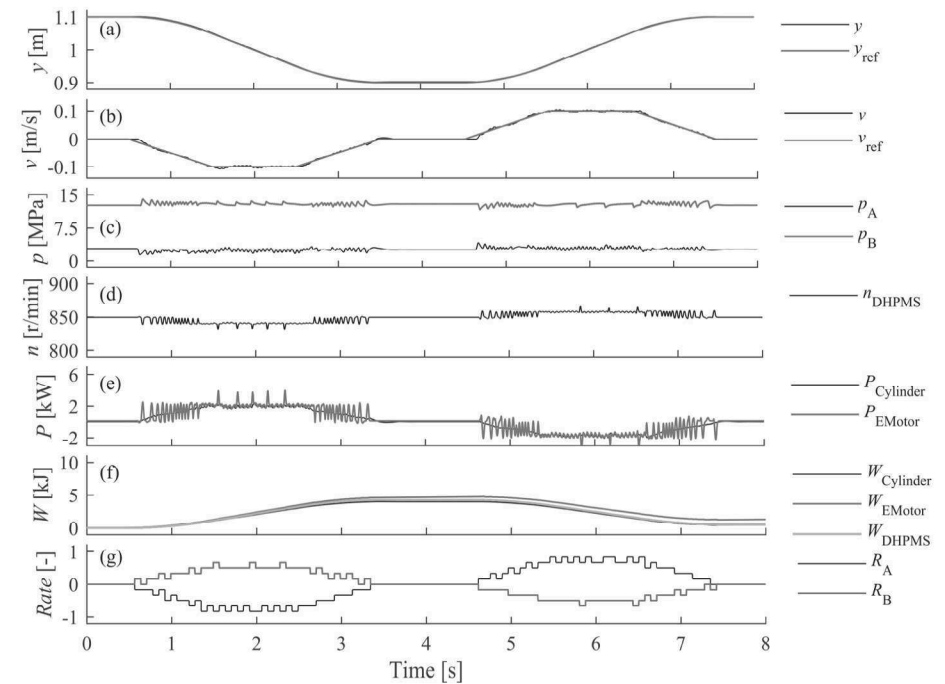


Figure 5. Simulation results: normal operation.

### 4 Simulation results

The system is modelled using MATLAB/Simulink and the SimMechanics Toolbox, based on the principles presented in /16, 17/. The lift cylinder is controlled by the DHPMS without using position feedback. Figure 5 shows the system behaviour in normal operation; thus, no faults exist. The boom is first lifted by completing a retracting movement of 0.2 m. At the end of the trajectory, the boom returns to its original orientation. Figure 5a shows that the position tracking of the piston is good, as is the accuracy of the velocity tracking (Figure 5b); only slight oscillation can be seen in piston velocity. The cylinder-side pressure (back-pressure) is approximately 3 MPa, while the pressure in the rod-side (load pressure) is approximately 12.5 MPa as shown in Figure 5c.

The rotational speed of the electric motor is set to 850 r/min to produce sufficient flow even if only five out of six DHPMS pistons can be used. During the movement, the rotational speed has a slight ripple due to torque variation in spite of the attached flywheel (Figure 5d). Figure 5e shows the input and output powers of the system: the cylinder power is approximately 2 kW during boom lifting, and approximately -1.5 kW when the boom is lowered. The power output by the electric motor has quite a large ripple, but the average is close to the cylinder power. However, the model does not consider mechanical friction losses in the DHPMS. The losses caused only by compressibility of the fluid and the pressure drop in the on/off control valves have been taken into account on the DHPMS model. The cumulative energy outputs by the electric motor, DHPMS, and lift cylinder are shown in Figure 5f. At the end of the trajectory, the work done by the cylinder is 0.4 kJ, whereas this value is 0.5 kJ for the DHPMS and 1.3 kJ for the electric motor. Thus, the total energy loss of the system is 0.9 kJ for the studied trajectory, and the corresponding hydraulic loss is 0.1 kJ. The utilization rate of DHPMS outlets is shown in Figure 5g. The rate is expressed as selected pumping/motoring modes per revolution, so the rate has six levels. It can be seen that the maximum rate is ±0.83, which means that full pumping/motoring is not required for the trajectory.

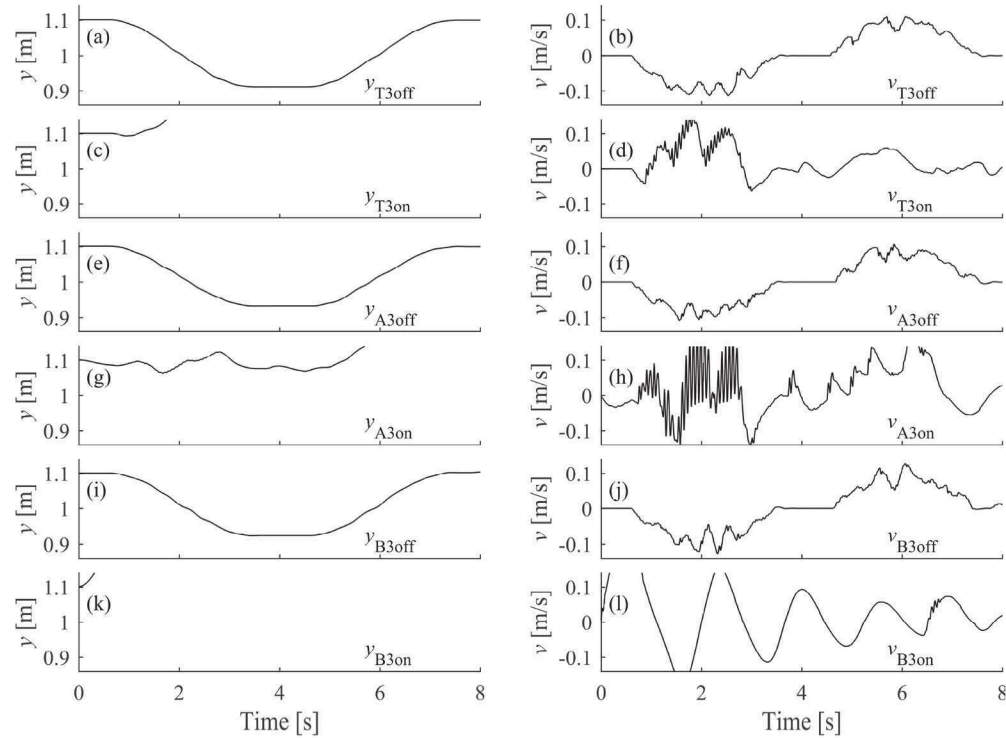


Figure 6. Simulation results: uncompensated faults.

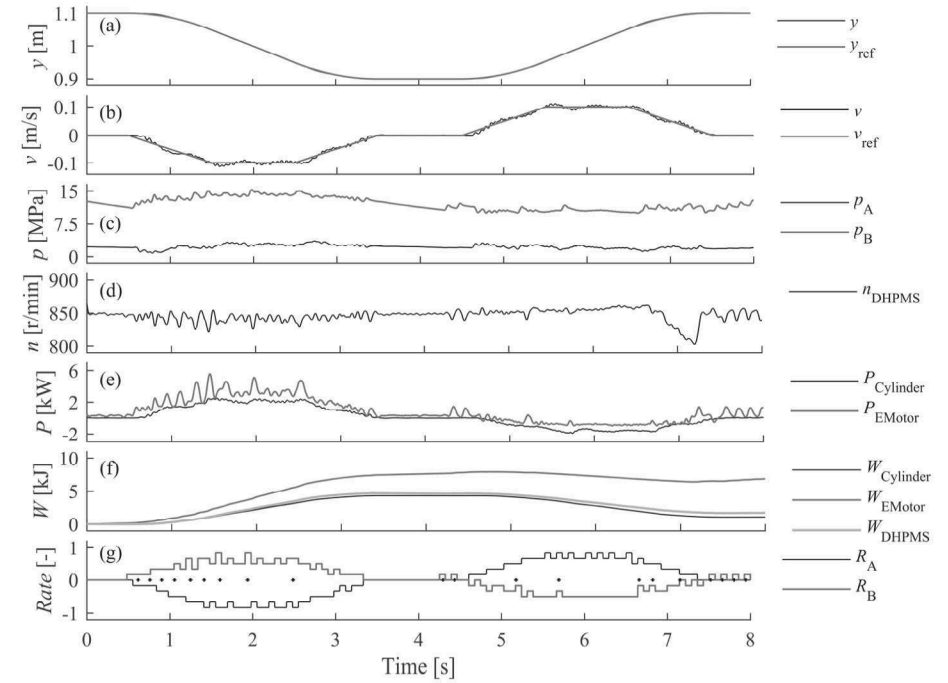
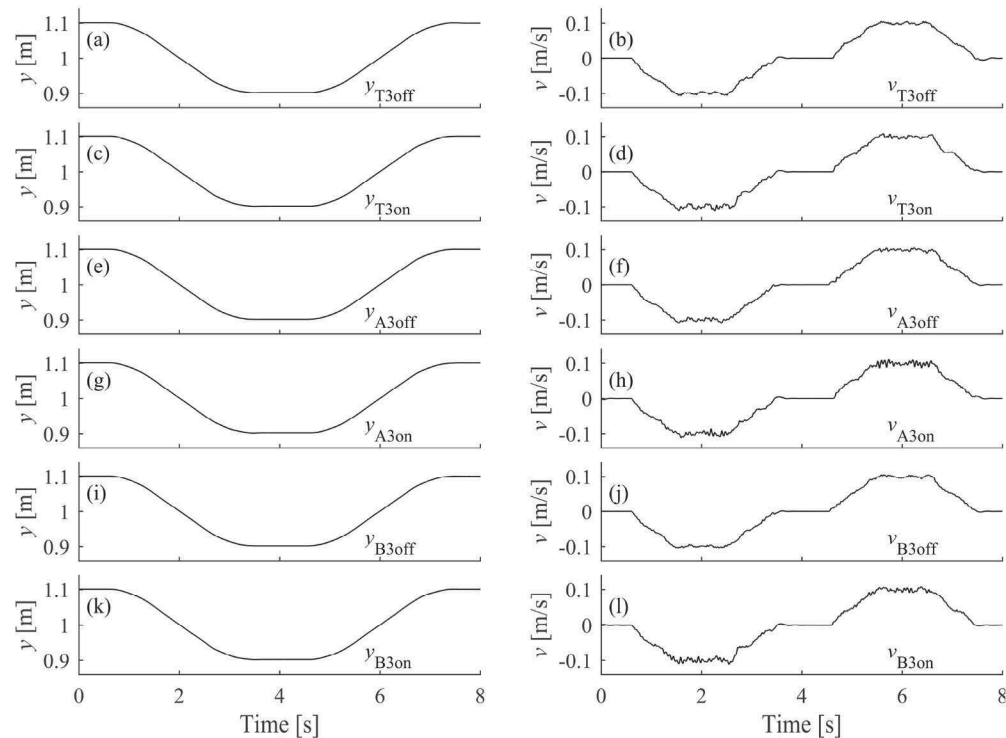


Figure 8. Measurement results: normal operation.

Figure 6 shows the simulated position and velocity tracking without valve fault compensation. An off-jammed T-valve causes a shorter piston movement of the cylinder as the controller calculates the fluid volumes, assuming that each pumping and motoring stroke is completed (Figure 6a). The effect of an off-jammed A-valve (Figure 6e) or B-valve (Figure 6i) is similar; the piston position falls behind its reference value. The missed pumping and motoring strokes obviously also influence the velocity tracking, as shown in Figures 6b, 6f, and 6j.

An on-jammed valve even has a drastic impact on the cylinder piston velocity. Due to cross-flow between the outlets and the tank, an on-jammed T-valve makes the velocity uncontrollable, as shown in Figure 6d. The cylinder piston also reaches its front end as the load pressure drops (Figure 6c). An on-jammed outlet valve has similar impact to the velocity, as shown in Figures 6h and 6l. However, oscillation is more intensive because the cross-flow occurs more often than in cases of on-jammed inlet valves. However, the influence on the cylinder piston position is the same, and eventually the front end is reached.

Figure 7 shows the trajectories after fault compensation. The simulations show that, regardless of a valve fault, the position tracking performance is as good as it is during normal operation (Figures 7a, 7c, 7e, 7g, 7i, and 7k). In the case of off-jammed valves, the velocity curves are also smooth, as shown in Figures 7b, 7f, and 7j. However, jamming on causes a slightly larger ripple to the velocity (Figures 7d, 7h, and 7l). The ripple is at its largest in cases where an outlet valve is jammed on, as the fluid is forced to pump to and motor from that outlet, independent of the movement. Nevertheless, the velocity is zero during idling, as the volume flow resulting from the on-jammed outlet valve is compensated for using the reverse DHPMS piston.

### 5 Experimental results

The system was also studied experimentally. Figure 8 shows the boom lifting/lowering cycle in normal operation, with no active faults. Position tracking is good, considering that the position is controlled open-loop, as shown in Figure 8a. Slight oscillation can be seen in velocity, but the tracking is satisfactory (Figure 8b).

Figure 8c shows the cylinder pressures during the trajectory. It is notable that the pressures decrease between the movements due to leakage through the DHPMS control valves. The rotational speed of the electric motor is set to 850 r/min, as shown in Figure 8d. At the end of boom lowering, the rotational speed drops due to back-pressure control. Figure 8e shows that the cylinder power is approximately 2 kW during boom lifting, but -1.5 kW when the boom is lowered. Correspondingly, the electric motor outputs 5.5 kW at maximum, while the minimum is approximately -1 kW. At the end of the measurement, the work done by the cylinder is 1.1 kJ, whereas this value is 1.7 kJ for the DHPMS and 6.8 kJ for the electric motor, as shown in Figure 8f. The utilization rate of the outlets remains below 1; hence, full pumping/motoring is not required to produce sufficient volume flows. The black (+) signs show the mode-changing instant, when the back-pressure control interferes due to the leakages.

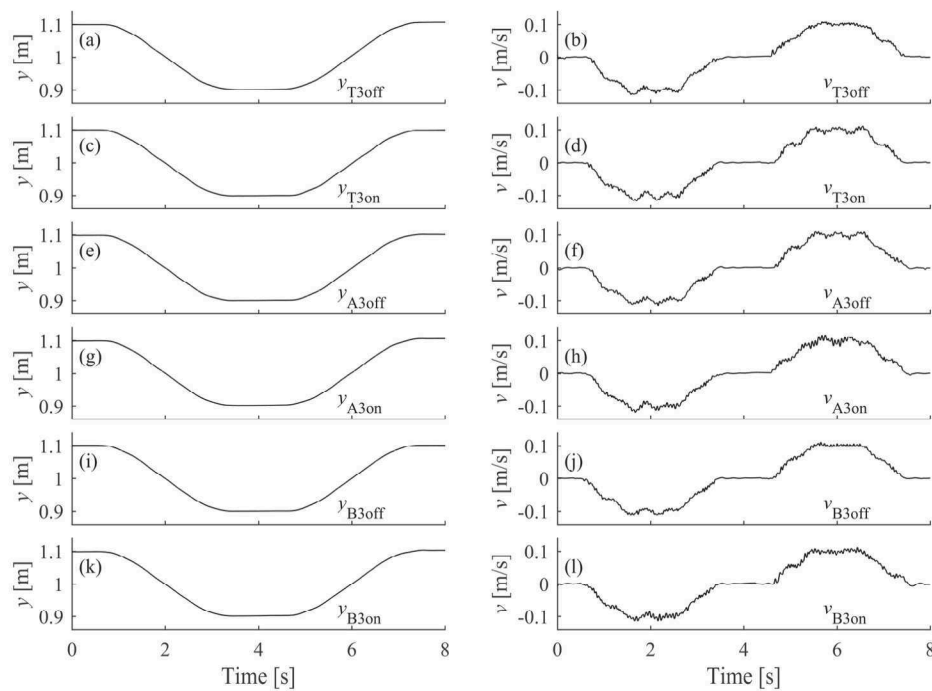


Figure 9. Measurement results: compensated faults.

The fault compensation is tested such that the valve faults are created with software and the adapted controller is utilized during the experiments. Figure 9 shows the piston movement and velocity for the fault cases: an off-jammed T-valve is shown in the first row, an on-jammed T-valve in the second row, an off-jammed A-valve in the third row, an on-jammed A-valve in the fourth row, an off-jammed B-valve in the fifth row, and an on-jammed B-valve in the sixth row. Position tracking is good in Figures 9c, 9e, 9i, and 9k. In cases of an off-jammed T-valve (Figure 9a) and an on-jammed A-valve (Figure 9g), the positioning at the end of the trajectory is less accurate, as the back-pressure controller cannot compensate for the leakage from the rod side to the cylinder side. The velocity curves hardly differ from one another. However, the on-jammed valves (Figures 9d, 9h, and 9l) cause somewhat coarser velocity tracking than do the off-jammed valves (Figures 9b, 9f and 9j).

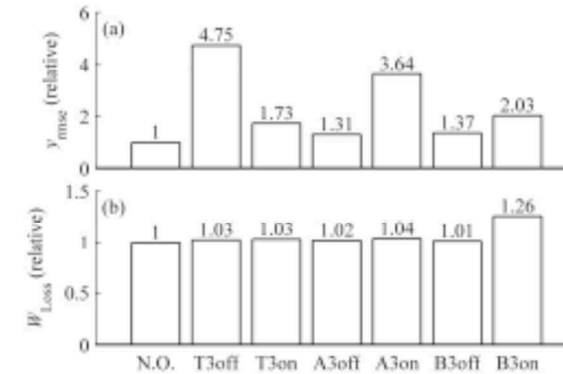


Figure 10. Relative RMSE for the piston position in each test (a) and relative energy losses for the corresponding cases (b).

## 6 Analysis of the results

The simulations show that an uncompensated off-jammed valves impairs the position and velocity tracking performance of the studied system, as the unrealized pumping/motoring strokes cause a jerky motion of the lift cylinder piston. An on-jammed control valve has an even more drastic impact on the boom, as the controllability can be completely lost. The cross-flow between the actuator and the tank line causes the load pressure to drop rapidly. Thus, the cylinder piston of the actuator moves toward its end position.

The impact of faults can be effectively compensated for by the controller adaptation: the position and velocity tracking remain good despite a faulty control valve. The simulation model assumes leakage-free control valves; hence, the positioning accuracy is as good as in normal operation. However, compensation of on-jammed valves causes a slightly larger ripple to the actuator velocity than in the normal operation.

The experimental tests verify the simulation results, showing that the valve faults can be effectively compensated for by using the controller adaptation. However, the leakages of the test system affect the position tracking accuracy. Figure 10a shows the relative root mean square error (RMSE) for the piston position in each test. An absolute RMSE for the normal operation is 0.8 mm, which is used as a reference value. Clearly, the error becomes large in cases where the DHPMS piston is constantly connected to outlet A. The leakage from the rod side to the cylinder side sustains the back-pressure; therefore, the leakage cannot be compensated for by the back-pressure controller during the active fault.

The relative losses for the system are shown in Figure 10b. The losses increase 26% when the faulty valve connects the DHPMS piston to load pressure. As the fault is compensated for by connecting the reverse piston to that outlet as well, both the mechanical-hydraulic and volumetric efficiency of the DHPMS are decreased. In the remainder of cases, however, the increased overall losses are within 4%.

## 7 Conclusions and future work

Digital hydraulic systems possess natural redundancy, as this technique uses parallel connected components. Thus, the systems are fault-tolerant as long as the defect can be detected and identified. The key is to reconfigure the controller to minimize the effects of the existing fault. Normally, the operation can be continued despite slightly degraded performance.

The fault-tolerant controller was validated using both simulations and experimental tests. The results showed that all valve faults could be effectively compensated for by using the adapted controller. Thus, the system is naturally fault-tolerant with no changes to the hardware needed. Moreover, the fault compensation requires no extra computing power compared with the normal operation. The leakage of the control valves, however, degraded the position tracking accuracy in some fault cases. The simulation model, which assumed leakage-free control valves, gave a better result: neither off-jammed nor on-jammed valves decreased the control performance, provided that the reconfigured controller was used.

In most cases, a defective valve in the multi-outlet digital hydraulic pump-motor decreases the potential volume flow at the outlets. The studied trajectory for the cylinder piston was selected such that the required maximum volume flow was about 80% of the maximum capacity. Thus, the produced flow remained adequate even though only five pumping/motoring pistons out of six could be used. Otherwise, the rotational speed of the electric motor should have been controlled as well. Hence, control of the rotational speed should be considered in future studies. Furthermore, applicability of existing fault detection methods should be verified.

In summary, displacement control using a multi-outlet digital pump-motor has shown potential for the development of more energy-efficient machinery. Despite challenges related to the new technology, these digital machines are worth studying as their programmability offers new capabilities, including features related to fault tolerance.

### Nomenclature

Variable	Description	Unit
$A_A$	Cylinder blind end area	[m <sup>2</sup> ]
$A_B$	Cylinder rod end area	[m <sup>2</sup> ]
$C_h$	Hydraulic capacitance	[m <sup>3</sup> /Pa]
$\mathbf{M}$	Final control mode vector for a pair of pumping/motoring pistons	[-]
$\mathbf{M}_{mp}$	Preselected control mode vector for a pair of pumping/motoring pistons	[-]
$n_{DHPMS}$	Rotational speed of the DHPMS axis	[r/s]
$p_A$	Cylinder A-chamber pressure	[Pa]
$p_B$	Cylinder B-chamber pressure	[Pa]
$p_{back}$	Cylinder back (non-load) pressure	[Pa]
$p_T$	Tank (DHPMS inlet) pressure	[Pa]
$V_{cyl,A}$	Cylinder A-chamber fluid volume	[m <sup>3</sup> ]
$V_{cyl,B}$	Cylinder B-chamber fluid volume	[m <sup>3</sup> ]
$V_{disp}$	Geometrical displacement of pumping/motoring piston	[m <sup>3</sup> ]
$V_{err,A}$	Cylinder A-chamber fluid volume error	[m <sup>3</sup> ]
$V_{err,B}$	Cylinder B-chamber fluid volume error	[m <sup>3</sup> ]
$V_{ref,A}$	Cylinder A-chamber target fluid volume	[m <sup>3</sup> ]
$V_{ref,B}$	Cylinder B-chamber target fluid volume	[m <sup>3</sup> ]
$v_{ref}$	Cylinder piston target velocity	[m/s]

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