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A flexible working hydraulic system for mobile machines

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ABSTRACT
This paper proposes a novel working hydraulic system architecture for mobile machines. Load sensing, flow control and open-centre are merged into a generalized system description. The proposed system is configurable and the operator can realize the characteristics of any of the standard systems without compromising energy efficiency. This can be done non-discretely on-the-fly. One electrically controlled variable displacement pump supplies the system and conventional closed-centre spool valves are used. The pump control strategies are explained in detail. Experimental results demonstrate one solution to the flow matching problem and the static and dynamic differences between different control modes.

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Introduction

Fluid power systems have been used successfully in mobile machines for several decades. Because of the machines' versatility, different hydraulic systems have been developed for different applications. Important properties of hydraulic systems include energy efficiency, controllability, damping and system complexity. The choice of working hydraulic system is often a compromise between these properties. An energy-efficient and flexible working hydraulic system is proposed in this paper. With the proposed system, it is possible to change static and dynamic characteristics on-the-fly to fit a specific machine, working cycle or operator.

In the field of mobile hydraulic systems, one hot research topic is to eliminate the control valves and dedicate one pump to each actuator. Multiple concepts have been developed, including pump controlled actuators (Rahmfeld and Ivantysynova 2001, Heybroek 2008), hydraulic transformers (Achten *et al.* 1997) and electro-hydrostatic actuators (Gomm and Vanderlaan 2009). Such systems are not yet common commercially in mobile machines but can be found in, for example, the aerospace industry (Raymond and Chenoweth 1993). Valveless systems improve energy efficiency compared to single-pump systems, especially when multiple functions are operated simultaneously. However, one has to bear in mind that valveless systems may require several valves to handle, for example, asymmetric cylinder actuation and meet safety requirements (Williamson and Ivantysynova 2007, Heybroek 2008). Furthermore, since all actuators have their own dedicated pump in the valveless concepts, each has to be sized to handle maximum speed. In single-pump systems, the pump can be downsized since not every load is actuated at full speed simultaneously very often. For these reasons, the total installed displacement tends to be high in valveless systems compared to single-pump systems. Unlike these valveless concepts, this paper focuses on single-pump systems.

Another interesting area in mobile hydraulic system research is systems in which the inlet and outlet orifices in the directional valve are decoupled. Numerous configurations for individual metering systems have been developed, both in academia and in industry (Eriksson 2010). However, similar to valveless systems, these systems are not yet common commercially in mobile machines, mainly because of the control complexity and cost. In this paper, the focus is on systems using conventional spool valves.

Mobile working hydraulic systems

Today, most working hydraulic systems in mobile machines are operated with open-centre valves and fixed displacement pumps. Such systems can be considered to be relatively simple, robust and cost-effective, but also often energy-inefficient. These systems suffer from load interference, which means that the pressure level at one load can significantly influence the velocity of other actuators. Furthermore, the flow rate is not only dependent on spool position, but also

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on load pressure, often referred to as load dependency. From a controllability point of view, this is often considered a drawback. From a dynamics point of view, load dependency gives the system a high damping, which means that the system is less prone to oscillations. Damping is a preferred property when handling large inertia loads, for example the swing function of a mobile crane.

Load sensing systems improve energy efficiency compared to open-centre systems by continuously adapting their pressure just above the highest load. This means that a specific spool displacement results in a certain flow, independent of the load pressure. This is also true for simultaneous movements of loads if pressure compensators are used. The pressure insensitivity makes load sensing systems easy to operate for velocity or position control of low inertia loads. However, with high inertia loads, the operation becomes jerky because of the low damping. Furthermore, the closed-loop control mode for the pump might lead to stability issues (Krus 1988). An early review of load sensing systems was made by Andersson (1980).

To improve energy efficiency but still maintain load dependency and high damping, systems based on variable displacement pumps and open-centre valves have been developed, i.e. negative control (Andersson 1997). The controllability is similar to open-centre systems. Power losses are generally higher than in load sensing systems but not as high as in open-centre systems because of the variable pump. However, open-centre variable pump systems have power losses in neutral while load sensing systems do not.

A step forward from conventional hydro-mechanical pump controllers is to use an electrically controlled pump. This makes it possible to realize an electrical load sensing system (Hansen *et al.* 2010). Another possibility is to control the pump displacement setting based on the operator's command signals rather than feedback signals from the loads. One system solution is to control the pump displacement setting according to the sum of all requested load flows, here referred to as flow control. Advantages with flow control compared to load sensing are higher energy efficiency because of a decreased pump pressure margin at most points of operation (Djurovic 2007) and better dynamic characteristics because of the open-loop control (Latour 2006, Finzel 2010). However, the flow is statically pressure independent in flow control systems, giving the system a low damping (Axin 2013). A review of flow control systems has been made by Axin *et al.* (2014a).

One problem with flow control systems using compensators which control the absolute flow through the directional valve is flow matching (Eriksson and Palmberg 2010). The pump flow has to be matched against the sum of all load flows. If this is not the case, two situations may arise:

The pump flow is too low: The compensator spool at the highest load will open completely, resulting in a decrease in speed for that load.

The pump flow is too high: Both compensator spools will close more and the pump pressure will increase until the system relief valve opens.

A great deal of research solving this flow matching problem has been presented (Fedde and Harms 2006, Djurovic 2007, Grösbrink *et al.* 2010, Xu *et al.* 2015). These solutions include additional sensors or a bleed-off valve to tank. Another solution is to use flow-sharing pressure compensators, distributing the entire pump flow relative to the individual valve openings (Finzel and Helduser 2008). In some applications, however, it is not desired to distribute the flow when the pump is being saturated. Furthermore, flow sharing compensators makes the highest load disturb all lighter loads (Lantto 1994).

A flexible hydraulic system

As described in the previous chapter, different hydraulic systems have different system characteristics. In some applications, smooth control with high damping is desired while high energy efficiency and handling capabilities with precise position control are important in others. A flexible system solution using an electrically controlled variable displacement pump is proposed in this paper. It is possible to realize open-centre, load sensing and flow control, but also a mix of the three systems. Conventional closed-centre spool valves are used, which results in high energy efficiency.

Pump controller

The pump controller used in the flexible working hydraulic system is shown in Figure 1. Sensors measure pump pressure, maximum load pressure, shaft speed and pump displacement setting. Input signals from the operator to the electric controller are pressure and flow commands. It is thus possible to control pressure in a closed-loop and control flow in an open-loop without any feedback signals from the system.

Combining load sensing and flow control

Both load sensing and flow control have their respective pros and cons. One drawback with load sensing is that the pump controller is a part of the closed-loop control gain (Krus 1988). Improving the pump's response time will decrease the stability margins of the complete system. Flow control has no such issues, but other challenges arise instead. For example, it is problematical to combine a flow control pump with traditional pressure compensators (Eriksson and Palmberg 2010). The solution proposed in (Axin *et al.* 2014b) is to combine

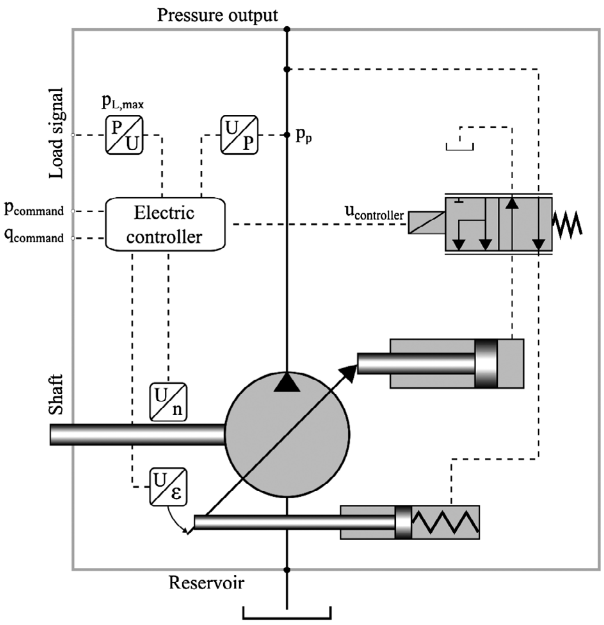

Figure 1. Electronic pump controller measuring pump pressure, maximum load pressure, shaft speed and displacement setting (P1 axial piston pump, product catalogue). Inputs from the operator are pressure and/or flow commands.

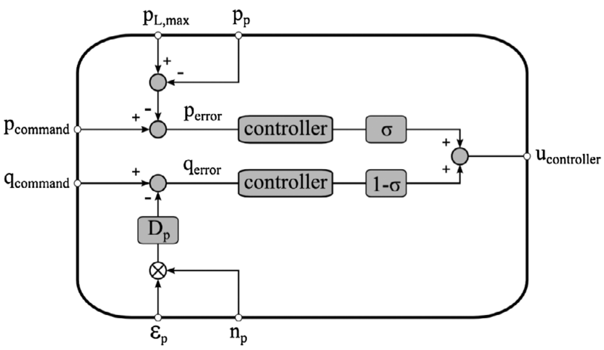

Figure 2. Schematic block diagram of the electric controller used in this paper. The output signal receives influence from both the pressure controller and the flow controller. The parameter σ set by the operator determine how much influence should come from the pressure and the flow parts of the controller, respectively.

pressure and flow control, thereby taking advantage of the respective benefits of the two systems and at the same time avoiding their drawbacks.

Similar ideas have been proposed by other researchers. A pump controller which consists of an electro-hydraulic valve controlling the flow and a hydro-mechanical valve controlling the pressure were studied by Grösbrink and Harms (2009) and Grösbrink *et al.* (2010). The valves are combined in such a way that the minimum pump displacement of the two controllers is selected. This means that the pump will be flow controlled as long as the pump flow demand is not too high. If the electro-hydraulic controller demands more flow from the pump than the valves can handle, the pressure will rise and the pump controller will automatically be switched to pressure control mode. Xu *et al.* (2012) and (2015) have studied a similar approach. The difference is that both flow and pressure control are realized electro-hydraulically. Both approaches use flow control as the primary control mode and pressure control as a safety control mode. Switching controllers might cause stability problems as shown in Xu *et al.* (2015). Hansen (2009) and Hansen *et al.* (2010) proposed an electronic load sensing design with a pressure controller, in which a feed forward from the joystick command signal was added.

This paper proposes summarizing the flow controller and the pressure controller in order to obtain influence from both pressure and flow. Furthermore, it is possible for the operator to choose how much influence should come from the pressure and the flow parts of the controller, respectively. This is done using a parameter, σ, see Figure 2. $\sigma = 1$ results in a pure pressure controller and $\sigma = 0$ results in a pure flow controller. $0 < \sigma < 1$ results in a combination of pressure and flow control. It is thus possible to control the pump continuously from pressure control to flow control.

By using a combination of pressure and flow control, the pump displacement setting is determined partly by the load pressure feedback and partly by the flow command signal. A low load pressure feedback gain can be used to solve the flow matching problem. When too much flow is demanded by the pump and the system pressure rises, the pressure controller will reduce the pump displacement setting, thereby avoiding an undesired pressure build-up. Furthermore, since the pressure controller only has to contribute a small part of the output signal to the displacement control valve, stability margins are gained.

Combining open-centre and flow control

Even though flow control has no stability issues attached to the pump controller, the damping is still often low. One way of increasing the damping is to introduce a load dependency into the system. Open-centre systems have this load dependency in terms of an open-centre channel. The losses, however, are often substantial. Changing to a variable pump but still maintaining open-centre valves improve efficiency (Andersson 1997). This paper proposes mimicking the behaviour of a conventional open-centre system by using the electrically controlled pump and closed-centre valves. This will increase energy efficiency further compared to variable pump systems using open-centre valves.

In the proposed solution, the open-centre flow is reproduced virtually by controlling the variable pump. The flow that would go through the open-centre path in a conventional open-centre system is calculated by measuring the pump pressure and having a model of the opening area in the open-centre channel according to Equation (1).

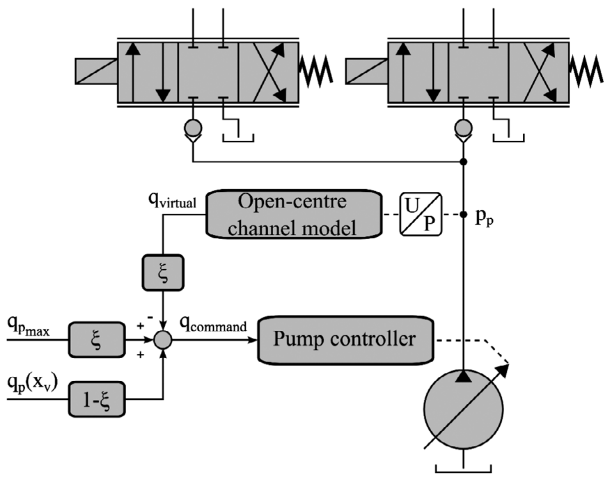

Figure 3. Proposed system solution using a flow controlled pump. $\xi = 1$ results in open-centre mode, $\xi = 0$ results in flow control mode and $0 < \xi < 1$ results in something in-between.

$$q_{\text{virtual}} = C_q A_{oc_1} \sqrt{\frac{2}{\rho}\left(p_p - p_{oc_1}\right)} = \cdots = C_q A_{oc_n} \sqrt{\frac{2}{\rho} p_{oc_n}} \Leftrightarrow q_{\text{virtual}} = C_q A_{oc_1} \sqrt{\frac{2}{\rho} p_p \left(1 - \frac{\sum_{k=2}^{n} \frac{1}{A_{oc_k}^2}}{\sum_{k=1}^{n} \frac{1}{A_{oc_k}^2}}\right)} \tag{1}$$

The virtual flow through the open-centre path is then subtracted from the maximum flow rate of the pump and the result is the command flow sent to the pump controller, see Figure 3.

When no valve is activated, the reference flow will be zero. That can be compared with all flow going through the open-centre channel. Activating a valve will decrease the opening area of the virtual open-centre channel, thus allowing a small flow to be sent by the pump, increasing the pump pressure. At a certain pressure level, the reference flow will find its equilibrium, only compensating for its own leakage. Activating the valve more will continue to increase the pressure until the pump pressure becomes higher than the load pressure. There will then be a flow to the load. Increasing the spool stroke further will decrease the opening area in the virtual open-centre channel, which means increased flow from the pump. When the valve is completely opened, the pump will be at maximum displacement, sending all flow to the load. A conventional open-centre system has exactly the same working principle, although control is accomplished hydraulically instead of electrically (Axin *et al.* 2014c).

Since electronic control is used, it is possible to have an arbitrary model of the virtual open-centre channel. For example, it would be possible to continuously decrease it in order to reduce the load dependency. Here, it is proposed to have a parameter, ξ, which is a multiplication coefficient on the virtual flow. At the same time, ξ will also change the input signal to the system, see Figure 3. Instead of being the maximum flow rate, the input signal will be dependent on the joystick command signals from the operator. The extreme case is when no load dependency exists at all, $\xi = 0$, resulting in a flow control system. By changing the value of ξ, it is possible to realize a system with open-centre characteristics, a flow control system or something in-between.

A similar commercial system design is the Virtual Bleed Off System from Bosch Rexroth (Virtual Bleed Off, webpage). However, it does not have the possibility to tune the load sensitivity online. Another similar solution has been patented by Filla (2014). A conventional load sensing pump is used and the directional valves are actively controlled in order to achieve open-centre characteristics.

Combining open-centre and load sensing

In a conventional open-centre system, the operator controls the pump pressure by activating a valve. The pump pressure is determined by the opening area in the open-centre path and the magnitude of the open-centre flow. This paper proposes actively controlling the pump pressure using the variable pump. The same virtual model of the opening area in the open-centre path as in the previous section is used. The virtual flow through the open-centre path is calculated by measuring the current pump displacement setting and rotational speed, see Equation (2). The pump pressure can then be calculated according to Equation (3).

$$q_{\text{virtual}} = q_{p_{\max}} - q_p = D_p n_p (1 - \varepsilon_p) \tag{2}$$

When no valve is activated, the reference pump pressure will be close to zero. This is the case when all flow is going through the open-centre path in a conventional

$$q_{\text{virtual}} = C_q A_{oc_1} \sqrt{\frac{2}{\rho}\left(p_p - p_{oc_1}\right)} = \cdots = C_q A_{oc_n} \sqrt{\frac{2}{\rho} p_{oc_n}} \Leftrightarrow p_p = \frac{\rho q_{\text{virtual}}^2}{2C_q^2} \left(\sum_{k=1}^{n} \frac{1}{A_{oc_k}^2}\right) \tag{3}$$

open-centre system. Activating a valve will decrease the opening area of the virtual open-centre channel, which will increase the reference pump pressure. At a certain pressure level, equilibrium will be found and the pump will only compensate for its own leakage. The pump displacement setting will then be close to zero, which means that all flow is still going through the open-centre channel. Activating the valve more will continue to increase the pressure until the pump pressure becomes higher than the load pressure. There will then be a flow to the load and the pump displacement setting will increase to maintain the pressure. This reduces the virtual open-centre flow according to Equation (2).

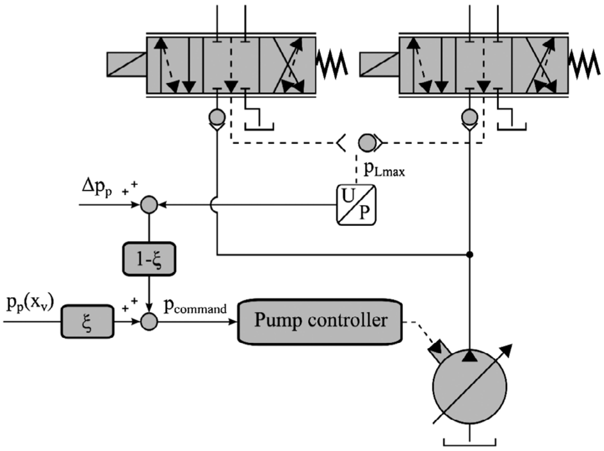

Figure 4. Proposed system solution using a pressure controlled pump. $\xi = 1$ results in open-centre mode, $\xi = 0$ results in load sensing control mode and $0 < \xi < 1$ results in something in-between.

Increasing the spool stroke further will decrease the opening area in the virtual open-centre channel and decrease the virtual open-centre flow, allowing more flow to the load. When the valve is completely opened, the pump will be at maximum displacement, sending all flow to the load.

Similar to the previous section, it is possible to reduce the load pressure dependency. This is done by the same parameter, ξ, which in this case will change the reference pump pressure. Instead of calculating the reference pump pressure according to Equation (3), it will also be influenced by the maximum load pressure and an additional load pressure margin, see Figure 4. The extreme case is when no load dependency exists at all, $\xi = 0$, resulting in a load sensing system. By changing the value of ξ, it is possible to realize a system with open-centre characteristics, a load sensing system or something in-between.

A similar commercial system design is the 3G valve from Nordhydraulic (Andersson 2013). However, it does not have the possibility to tune the load sensitivity online and a small excess flow is needed for the functionality. Differences between the 3G valve and the solution proposed in this paper are that system control is accomplished purely hydraulically and that it is possible to include compensators, which eliminates load interference issues.

Complete system solution

In the previous sections, three different system solutions have been proposed. All use the electronically controlled pump described above. In this section, the three solutions are combined in order to realize a flexible and energy-efficient working hydraulic system.

In the complete system solution, the parameter σ determines if the pump should be pressure controlled, flow controlled or something in-between and the parameter ξ sets the level of load dependency. Figure 5 shows the complete block diagram from input signals to displacement control valve signal, $u_{controller}$. No additional sensors are needed, only those available for the electronic pump controller. With the complete system solution, it is possible to realize a load sensing system, a flow control system, an open-centre system or something in-between, see Figure 6. Compared with only having the possibility to choose between the three original systems, this expands the design space and opens up the possibility for optimal control characteristics to fit a specific machine, working cycle, load or operator.

Experimental results

A test rig has been designed in order to validate the performance of the flexible working hydraulic system. It is a lorry crane with four actuators: boom, jib, telescope and swing, supplied by a commercially available electrically controlled pump that can be operated in both pressure and flow control mode (P1 axial piston pump, product catalogue), see Figure 7. The closed-centre directional valves are prepared for use with compensators or check valves. Pressure sensors are attached on the supply side, on the load sensing port of the directional valve and on both sides of all cylinders. The cylinders are also equipped with position sensors, a flow sensor is attached on the pump hose, and the pump is equipped with a displacement sensor. Both pump and valve are controlled by the commercial software IQAN by Parker Hannifin. The hardware data is shown in Table 1.

Flow matching problem

In this section, it is demonstrated how a combination of pressure and flow control can solve the flow matching problem. The directional valves are equipped with traditional pressure compensators, controlling the absolute flow rate to the loads. The flow command to the pump controller is increased from a correct level to 10% more than the valves are expecting. As can be seen from Figure 8(a), the pump pressure margin in flow control mode ($\sigma = 0$) then increases from a level slightly above 10 bar to about 55 bar. Theoretically, the pressure would increase until the system's main relief valve opens but secondary effects such as increased pump leakage stabilize the pressure. By introducing a load pressure feedback into the pump controller, the system will find equilibrium on a lower pressure level. Figure 8(a) shows how 2 and 5% load pressure feedback will affect the system. In load sensing mode ($\sigma = 1$), the system is insensitive to an incorrect flow demand since the pump is controlled only by the load pressure feedback. Figure 8(b) shows how the pressure

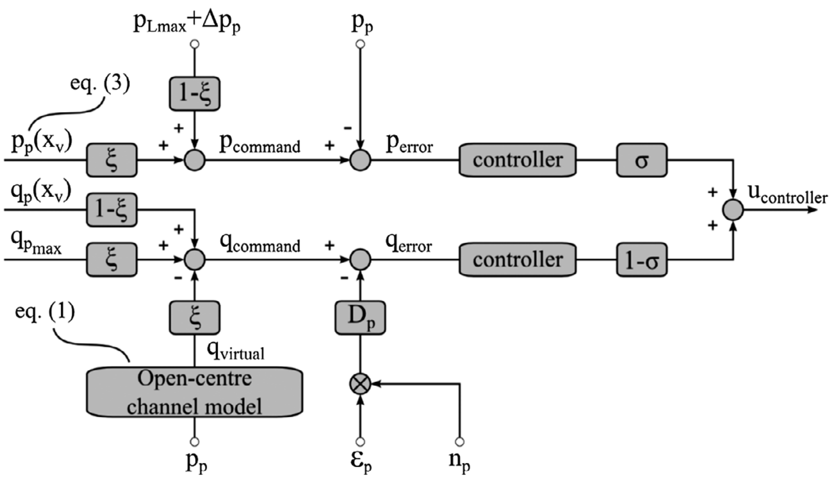

Figure 5. Complete block diagram for the proposed system design from input signals to displacement control valve signal, $u_{controller}$. σ determines if the pump should be pressure controlled, flow controlled or something in-between and ξ sets the level of load dependency.

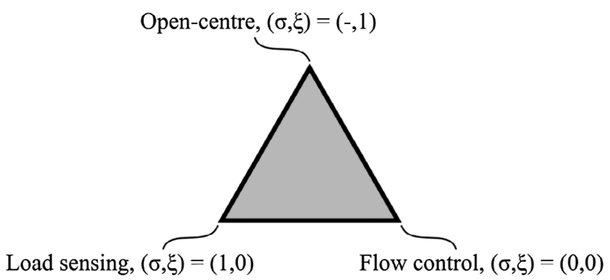

Figure 6. Design space for the proposed system design.

Figure 7. The crane used for experiments. The boom cylinder controls the first arm, the jib cylinder controls the second, telescope cylinders can extend the second arm and the swing cylinders can rotate the crane. The valve packages can be seen lower right.

equilibrium changes with the value of σ. By avoiding low values of σ, the pump pressure will not increase higher than in the load sensing mode when the flow command to the pump controller is higher than the valves are expecting.

Dynamic characteristics

A step is made in the boom function to demonstrate some dynamic differences between load sensing, flow control and solutions in-between. The directional valves are equipped with check valves instead of pressure compensators. It can be seen from Figures 9(a) and (b) that flow control mode ($\sigma = 0$) gives a faster response than load sensing mode ($\sigma = 1$). This is because the chain of signals to the pump controller is shorter. In load sensing mode, the joystick generates a pilot pressure which displaces a directional valve. The highest load pressure can then be sent electrically to the pump controller, which generates flow and thereby a pressure build-up in the pump hose. In flow control mode, a flow demand is sent directly to the pump controller when a joystick is activated. When controlling the pump with a combination of pressure and flow ($\sigma = 0.5$), the response time is between flow control and load sensing. Figure 9(c) shows that the initial delay increases approximately linear with the value of σ.

In Figures 9(a) and (b), it is also possible to observe system stability. In load sensing mode, the pump controller is a part of the loop gain, which gives low stability margins and an oscillatory behaviour. By decreasing the value of σ, and thereby decreasing the loop gain, the pump displacement setting is partly determined by the load pressure feedback signal and partly by the flow command signal. The pressure controller therefore only has to contribute a small part of the total output signal to the displacement control valve, which means that stability margins are gained. The oscillations are therefore lower for $\sigma = 0.5$. In flow control mode, the oscillations are similar to $\sigma = 0.5$, which means that both systems have high stability margins. This can also be observed in Figure 9(d). By

Table 1. Hardware data on the crane used for experiments.

Primary power source	Electrical motor: 30 kW
Pump size	75 cc
Pump speed	1000 rpm
Directional valve sizes	Maximum area: 31,64 mm^2
Cylinder boom	Piston diameter: 125 mm Piston rod diameter: 90 mm
Cylinder jib	Piston diameter: 100 mm Piston rod diameter: 70 mm
Mass moved by the crane	140 kg

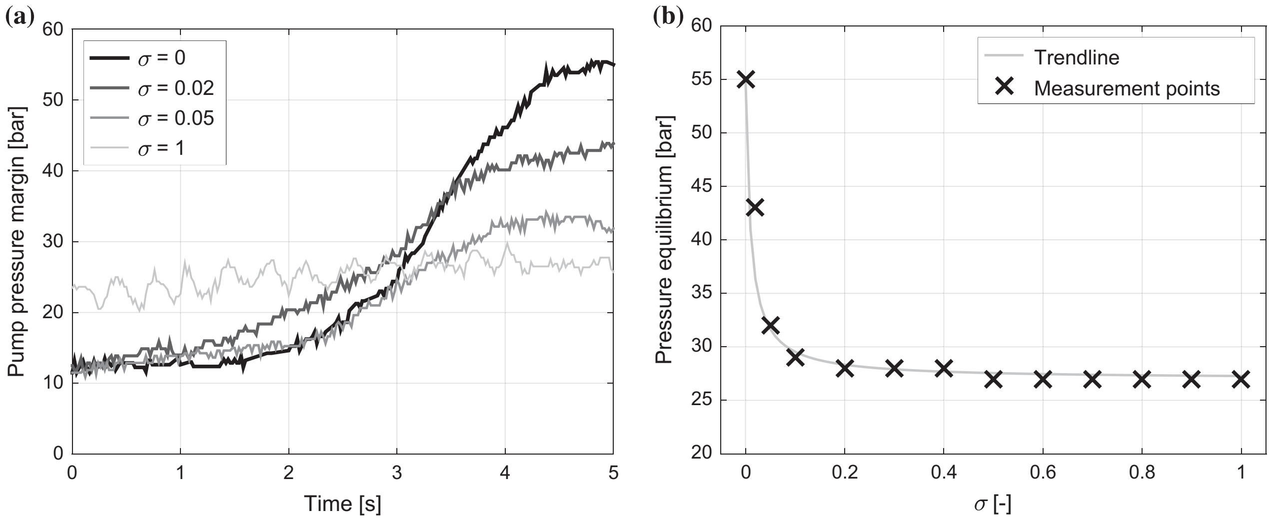

Figure 8. The flow demand is increased from a correct level to 10% more than the valve expects. A higher value of σ will make the pump controller less sensitive to an incorrect flow demand. (a) Pump pressure margin increase when too much flow is demanded by the pump for different values of σ. (b) Pressure equilibrium from Figure 8(a) as a function of σ. By avoiding low values of σ, the flow matching issues can be eliminated.

avoiding high values of σ, the pressure oscillations are approximately constant.

Similar results have also been reported in (Finzel and Helduser 2008) where a hydro-mechanical load sensing controller is compared with an electrical flow controller.

Load dependency

This section describes how different levels of load dependency affect the static and dynamic characteristics of the flexible hydraulic system. The pump is in flow control mode ($\sigma = 0$) and the load dependency is set according to $\xi = 0$, 0.5 and 1, respectively. The joystick command signal to the jib function is constant and a step is made in the boom function at 1 s. At 4 s, the boom joystick signal is set to 0 again. Figures 10(a), (c) and (e) show the pump pressure and the highest load pressure for different levels of load dependency. While moving, the boom function has the highest load pressure. Otherwise, the highest load pressure is the jib function. Figures 10(b), (d) and (f) show the boom and jib velocity and also the pump displacement setting.

The pump displacement setting is independent of the load pressure for $\xi = 0$. While increasing the value of ξ, the flow becomes more pressure-dependent. This static difference can be seen in Figures 10(b), (d) and (f). Since the jib function has a relatively low load pressure, the virtual flow through the open-centre path will be small according to Equation (1). This results in a higher velocity for the jib function during the first second when increasing the value of ξ. When the boom function is actuated, the pressure is increased to a relatively high level, increasing the virtual flow. The boom velocity therefore decreases with a higher value of ξ. When the boom stops moving, the pressure is reduced again. Because of the crane geometry, the jib function now requires a slightly higher pressure than between 0 and 1 s. As can be seen from Figure 10(b), this does not affect the static jib velocity when $\xi = 0$. However, when the load dependency is increased, the static jib velocity is slightly lower because of a higher pump pressure, resulting in a higher virtual flow through the open-centre path, see Figures 10(d) and (f).

The level of load dependency will also affect the dynamic characteristics. When making a step in the boom function at 1 and 4 s, the pump displacement setting and the system pressure levels will change. Because of the pump controller dynamics, this results in an overshoot and a few oscillations in the displacement setting when there is no load dependency, see Figure 10(b). When a load dependency exists, oscillations in the

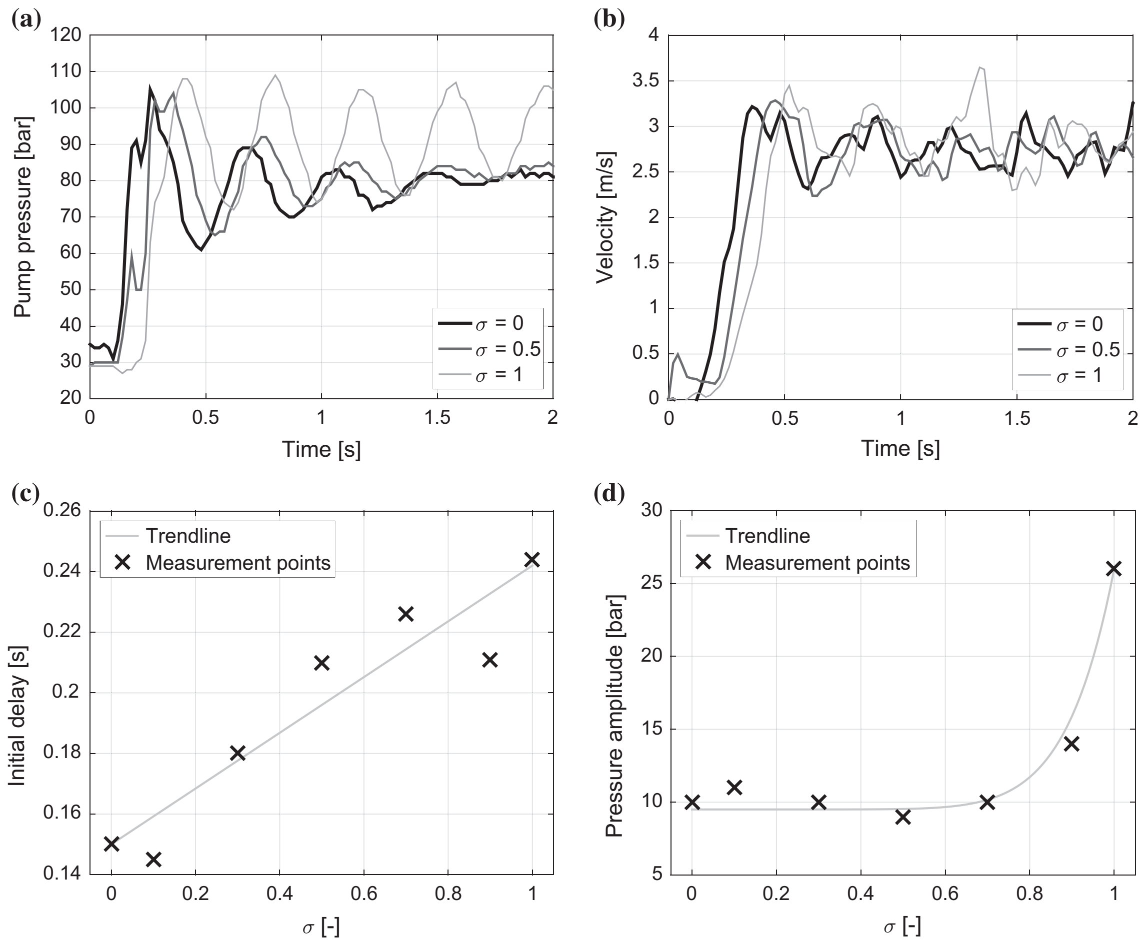

Figure 9. Dynamic comparison of the boom function for different values of σ. A step is made at 0 s. A lower value of σ improves the response time and decreases the oscillations. (a) Pump pressure as a function of time. (b) Crane velocity as a function of time. (c) Initial delay as a function of σ. (d) Pressure oscillation amplitude as a function of σ.

pump pressure will affect the pump displacement setting. As the pressure rises, creating an accelerating force, the pump decreases its displacement. The acceleration will then be slowed down, resulting in a system with more damping. As can be seen from Figures 10(a), (c) and (e), the pressure oscillations decrease while increasing the load dependency. It can also be seen that the displacement setting is actively controlled in order to reduce the pressure oscillations in Figures 10(d) and (f).

Discussion

A flexible system solution using an electrically controlled variable displacement pump and conventional closed-centre spool valves has been presented in this paper. With the complete system solution, it is possible to realize load sensing, flow control, open-centre or a mixture in-between. Compared with only having the possibility to choose between the three original systems, this expands the design space and opens up the possibility for optimal control characteristics to fit a specific machine, function or working cycle.

When optimizing control characteristics, it is also important to consider the operator. For example, the load sensitivity of an open-centre system is said to give the operator a better feel of the machine. A skilled operator can use this information feedback from the system to advantage and increase the machine's controllability. A non-skilled operator, however, might experience this load dependency as an inconsistency and it can then be regarded as a disturbance. With the flexible system layout proposed in this paper, it is possible for each operator to obtain their optimal control characteristics.

The energy efficiency of the proposed system design is similar to that of load sensing systems. The pump pressure is adjusted according to the highest load and high losses might occur when loads with different pressure demands are operated simultaneously. When controlling the pump by flow, however, the pressure drop between pump and load is given by the resistance in the hoses and in the valves rather than a prescribed pump pressure margin, which results in slightly higher efficiency.

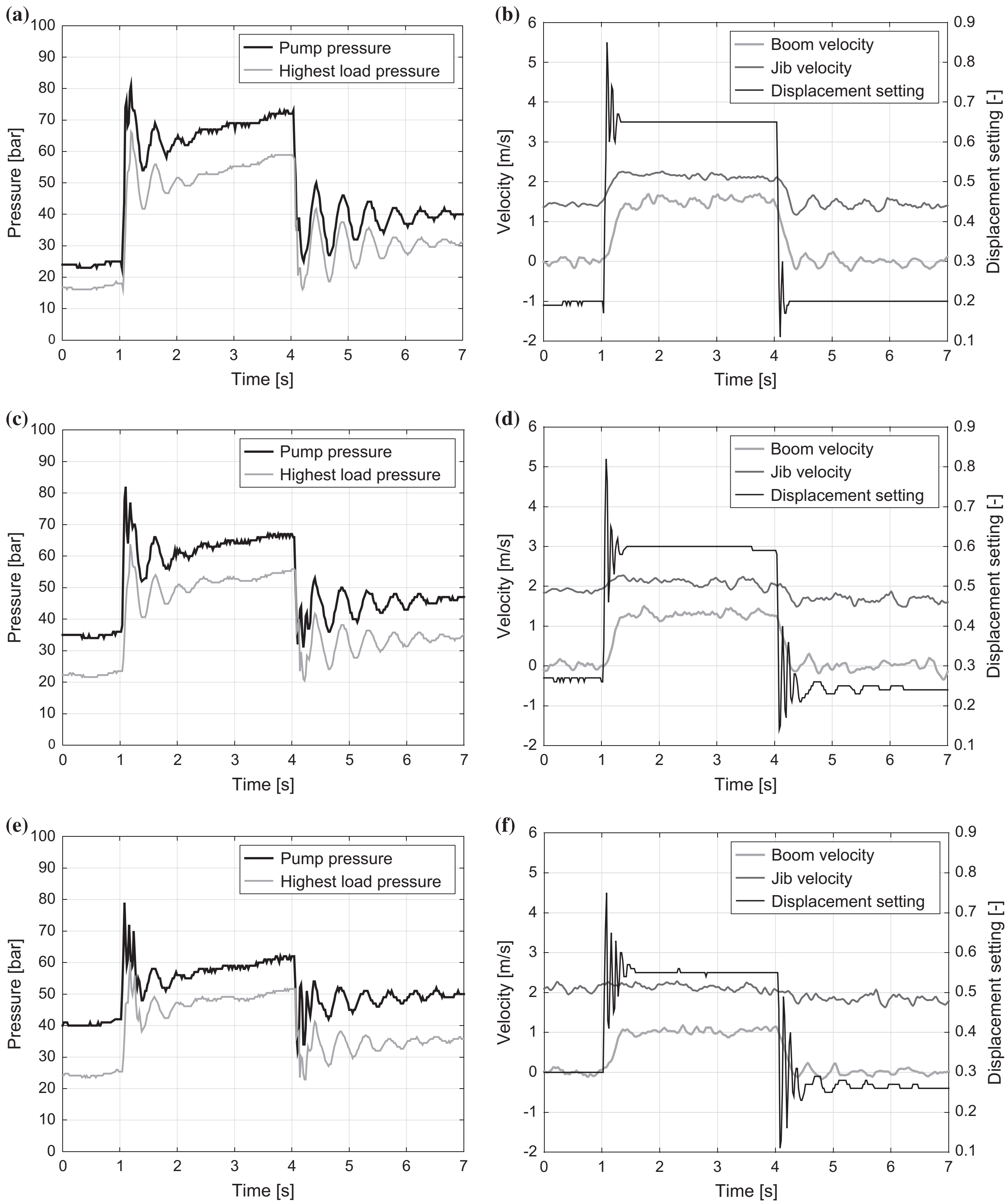

Figure 10. System characteristics for different values of ξ. Decreasing the value of ξ decreases the load dependency. (a) Pressure as a function of time for $\xi = 0$; (b) velocity and displacement setting for $\xi = 0$; (c) pressure as a function of time for $\xi = 0.5$; (d) velocity and displacement setting for $\xi = 0.5$; (e) pressure as a function of time for $\xi = 1$; and (f) velocity and displacement setting for $\xi = 1$.

Table 2 summarizes the system characteristics for different control modes. As can be seen, all modes have their pros and cons. If force control with high damping is desired, it is impossible to avoid load dependency. In load sensing and flow control mode, it is possible to equip the valves with pressure compensators and thereby eliminate load interference issues. However, pressure compensators cannot be included in open-centre mode because the functionality requires a load dependency. Therefore, it makes no sense to quantify flow matching issues in open-centre mode.

Table 2. Comparison of system characteristics for different control modes.

	Load sensing mode	Flow control mode	Open-centre mode
Energy efficiency	+	++	+
Velocity control	+	+	-
Force control	– –	– –	+
Damping	–	–	+
Load dependency	+	+	–
Load interference	+	+	–
Flow matching issues	+	–	

Conclusions

A novel system architecture is proposed where flow control, load sensing and open-centre are merged into a generalized system description. The proposed system is configurable and by tuning two parameters the operator can realize the characteristics of any of the standard systems without compromising energy efficiency or performance. By controlling the pump with a combination of pressure and flow, it is possible to avoid unnecessary energy losses in case of a flow mismatch between pump and valve, and at the same time improve system response and increase stability margins compared to load sensing. This optimum is also highly robust and verified by experiments. Experiments also demonstrate that the electronically controlled pump is fast enough to satisfy dynamic requirements.

Nomenclature

Variable	Description	Unit
A_{oc}	Opening area in the open-centre path	m^2
C_q	Flow coefficient	–
D_p	Pump displacement	m^3/rev
n_p	Pump shaft speed	rev/s
p_{Lmax}	Maximum load pressure	Pa
p_{oc}	Pressure in the open-centre path	Pa
p_p	Pump pressure	Pa
q_p	Pump flow	m^3/s
$q_{p_{max}}$	Maximum pump flow	m^3/s
$q_{virtual}$	Virtual open-centre flow	m^3/s
$u_{controller}$	Pump displacement valve signal	–
x_v	Valve position	m
ε_p	Pump displacement setting	–
Δp_p	Pump pressure margin	Pa
ξ	Parameter	–
ρ	Density	kg/m^3
σ	Parameter	–

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