Design Rules for High Damping in Mobile Hydraulic Systems

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Abstract

This paper analyses the damping in pressure compensated closed centre mobile working hydraulic systems. Both rotational and linear loads are covered and the analysis applies to any type of pump controller. Only the outlet orifice in the directional valve will provide damping to a pressure compensated system. Design rules are proposed for how the system should be dimensioned in order to obtain a high damping. The volumes on each side of the load have a high impact on the damping. In case of a small volume on the inlet side, the damping becomes low. However, the most important thing is to design the outlet orifice area properly. There exists an optimal orifice dimension for maximized damping; both smaller and larger orifice areas give lower damping independently of the volumes. This paper presents a method to dimension the outlet orifice area and the load volumes in order to obtain a desired system damping. Experimental results, which confirm the theoretical expectations, are also presented. The conclusions are that it is possible to obtain a high damping contribution from the outlet orifice if the system is dimensioned correctly. However, the energy efficiency needs to be considered while improving the damping.

Keywords: Damping, compensator, outlet orifice, efficiency

1 Introduction

In most fluid power systems, directional valves control the speed of the actuators. Two different types of valves are commonly used in mobile machines; open centre and closed centre. This paper addresses closed centre valves. Closed centre valves are commonly used in systems where the pump is actively controlled, for example in constant pressure and load sensing systems [1] [2] [3].

Several loads often share one common pump in mobile machines. If the system pressure level is adapted to the highest load and several functions are operated simultaneously, load interference challenges will occur. This means that the pressure level at the highest load will affect the velocity of the lighter loads. In applications where velocity control is an important property, the valves are often equipped with pressure compensators. The pressure drop across all directional valves will then be constant and the flow will be proportional to the opening area of the valve.

A drawback with pressure compensators is that the load will be poorly damped [4]. To obtain damping from a valve, the flow has to increase when the pressure drop across the valve increases and vice versa. A pressure compensated valve endeavours to achieve low influence on the flow from the load pressure. Except for secondary effects such as leakage, stiffness of the compensator spring and load friction, only the outlet orifice in the directional valve will provide damping to a pressure compensated actuator. A hydraulic system with poor damping has a tendency to oscillate, which has a negative impact on both the productivity of the application and the comfort of the operator [5].

This paper analyses the damping in systems using pressure compensated closed centre valves. Both rotational and linear loads are covered. The pump controller can be of any type; it does not affect the analysis. Design rules for how the system should be dimensioned in order to obtain a high damping are proposed. Experimental results, which confirm the theoretical expectations, are also presented.

2 Damping

A valve will with contribute damping if the flow increases when the pressure drop across the valve increases and vice versa. An example of a hydraulic system with high damping is the constant pressure system, see figure 1. The pump is controlled to maintain a constant pressure before the valve. If the load starts to oscillate, the pressure in the cylinder will also oscillate. Increased cylinder pressure, which means an accelerating force on the piston, results in decreased pressure drop across the valve and consequently also decreased flow through the valve. The acceleration will then be slowed down and the oscillations will decrease, as shown in figure 1.

If the directional valve is complemented with a pressure compensator, a constant flow valve is realized. The function of the pressure compensator is to maintain a constant pressure drop across the directional valve, regardless of pressure variations

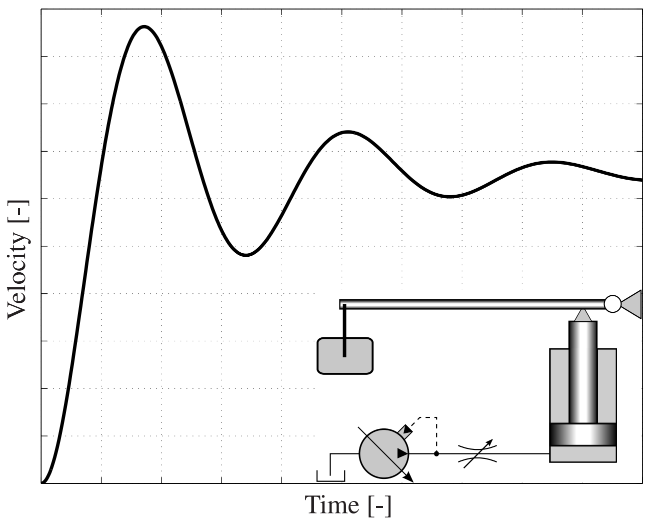

Figure 1: Simulation of a constant pressure system. The oscillations are decreased relatively fast which means that the damping is high.

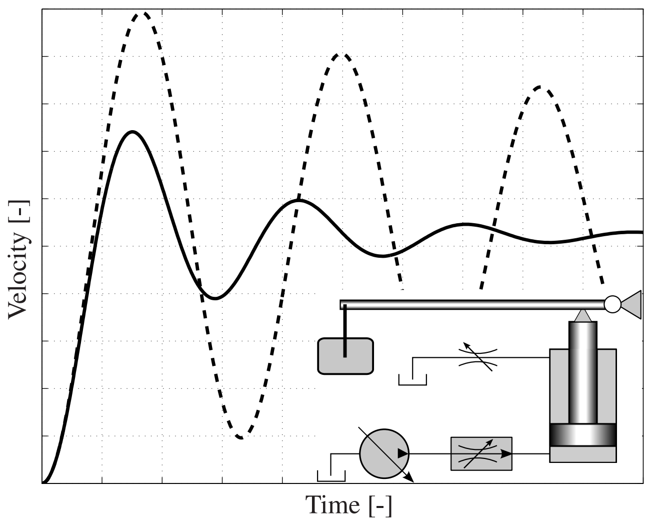

Figure 3: Simulation of a system with a constant flow valve and an outlet orifice. A good system design results in a high damping and a bad design results in a low damping.

on the pump and load side. The flow through the valve will then not change when the cylinder pressure oscillates. Therefore, nothing will dampen the accelerating force. An ideal pressure compensated hydraulic system with a closed centre valve is therefore completely undamped, which means that the oscillations will not decrease, see figure 2.

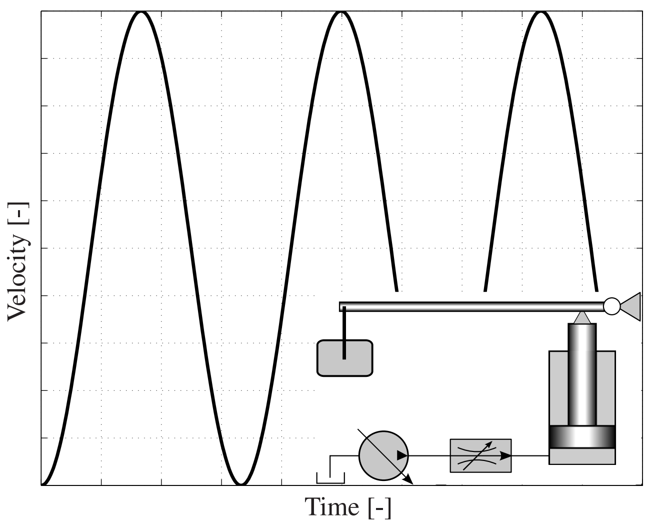

Figure 2: Simulation of a system with a constant flow valve. The pump controller can be of any type; it does not affect the results. Ideally, the system is completely undamped.

The simulation model in figure 2 is simplified to a pressure source, a constant flow valve, a cylinder and a load. The outlet orifice in the directional valve is thus missing. If an outlet orifice is included in the simulation, it is possible to obtain a high damping, see the solid line in figure 3. However, a bad system design results in a low damping according to the dashed line in figure 3.

It is important to design the system correctly in order to obtain a high damping from the outlet orifice. This paper will systematically describe how to design the system and propose design rules for how to achieve a high damping. Rotational and linear loads will be discussed. The difference between rotational and linear loads is that the volumes on the inlet and outlet side of the load will change during the stroke in case of a linear load. A rotational load has constant volumes. Experimental verifications of the findings are presented in section 5.

3 Rotational Load

A system consisting of a constant flow valve, a rotational inertia load and an outlet orifice, see figure 4, can be described by equations (1)-(4). The viscous friction has been ignored to simplify the analysis and the valve is considered to be much faster than the rest of the system. The valve dynamics are therefore ignored. The dynamics of pressure compensated valves have been studied in, for example, [6] and [7].

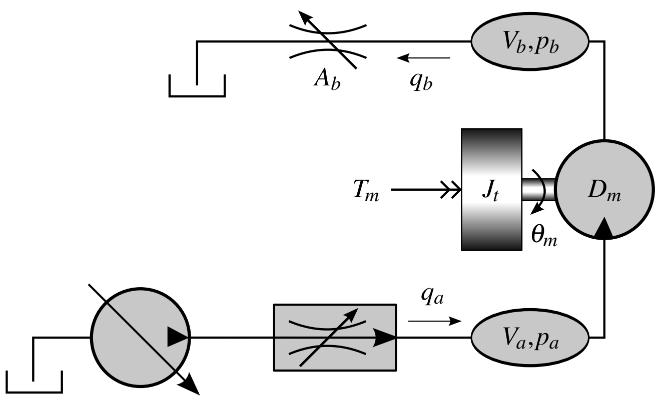

Figure 4: The system analysed in this section, a constant flow valve, a rotational inertia load and an outlet orifice.

$$q_a - D_m \frac{d\theta_m}{dt} = \frac{V_a}{\beta_e}\frac{dp_a}{dt} \tag{1}$$

$$J_t \frac{d^2\theta_m}{dt^2} = D_m(p_a - p_b) - T_m \tag{2}$$

$$D_m \frac{d\theta_m}{dt} - q_b = \frac{V_b}{\beta_e}\frac{dp_b}{dt} \tag{3}$$

$$q_b = q_a = C_q A_b \sqrt{\frac{2}{\rho} p_b} \tag{4}$$

Linearized and Laplace transforming equations (1)-(4) result in equations (5)-(8). The derivation of the equations is shown in [8].

$$Q_a - D_m s\theta_m = \frac{V_a}{\beta_e} s P_a \tag{5}$$

$$J_t s^2 \theta_m = D_m(P_a - P_b) - T_m \tag{6}$$

$$D_m s\theta_m - Q_b = \frac{V_b}{\beta_e} s P_b \tag{7}$$

$$Q_b = K_{c_b} P_b \tag{8}$$

where

$$K_{c_b} = \frac{\partial q_b}{\partial p_b} = \frac{C_q A_b}{\sqrt{2\rho p_b}} \tag{9}$$

An expression for K_{c_b} where maximum damping is obtained has been derived in [9] for a linear load. Substituting to a rotational load gives the expression in equation (10).

$$K_{c_{b_{opt}}} = D_m \sqrt{\frac{V_b}{\beta_e J_t (\gamma - 1)}} \gamma^{3/4} \tag{10}$$

where

$$\gamma = 1 + \frac{V_a}{V_b} \tag{11}$$

The maximum damping of the system can be calculated using equation (12), also derived in [9].

$$\delta_{h_{max}} = \frac{1}{2}(\sqrt{\gamma} - 1) \tag{12}$$

The most common system design is that the volumes on each side of the load are equal. In that case, the maximum damping of the system is $\delta_h = 0.21$ according to equations (11) and (12). To obtain the optimal damping, the required outlet orifice area can be calculated using equations (4), (9) and (10), resulting in equation (13).

$$A_{b_{opt}} = \sqrt{\frac{q_a K_{c_{b_{opt}}} \rho}{C_q^2}} \tag{13}$$

The outlet orifice opening area should according to equation (13) increase with the square root of the inlet flow rate; all other parameters are constant. Since the valve is assumed to be pressure compensated, the inlet flow is directly proportional to the inlet orifice opening area. The damping as a function of the outlet orifice area is shown in figure 5.

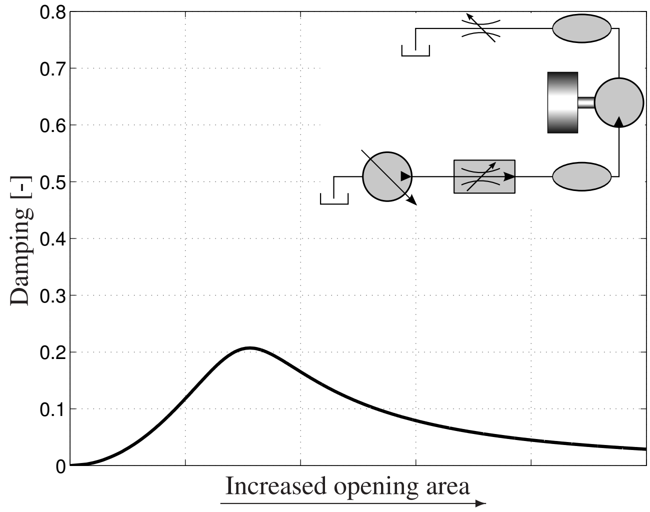

Figure 5: The damping as a function of the opening area for the outlet orifice. The volumes on each side of the load are assumed to be equal.

It is possible to obtain a higher damping than $\delta_h = 0.21$ by changing the volumes on each side of the piston. To increase the damping, the volume on the inlet side should be large compare to the volume on the outlet side according to equations (11) and (12). As can be seen in figure 6, it is possible to obtain a high damping contribution from the outlet orifice by increasing the volume on the inlet side. For example, the maximum damping becomes $\delta_h = 0.72$ if the inlet volume is five times larger than the outlet volume.

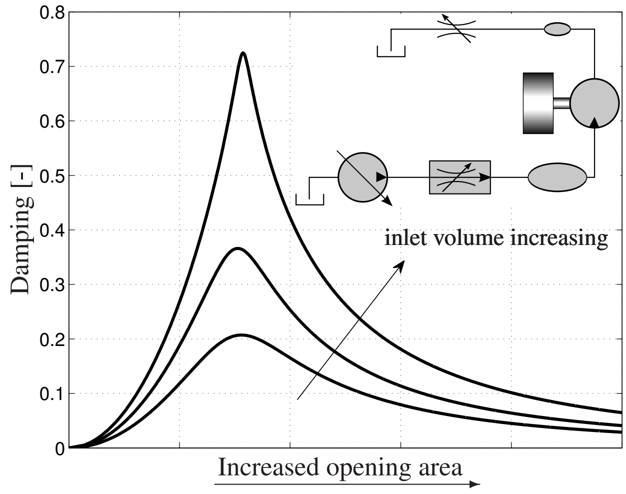

Figure 6: A higher damping can be obtained if the inlet volume is large compared to the outlet volume.

Increasing the inlet volume will increase the damping. However, there are drawbacks with this approach. In case of changing the flow direction, and thereby the direction of the rotation, there will be a large volume at the outlet side instead. Consequently, the damping will be low, see figure 7. If the outlet volume is five times larger than the inlet volume, the maximum damping is $\delta_h = 0.05$.

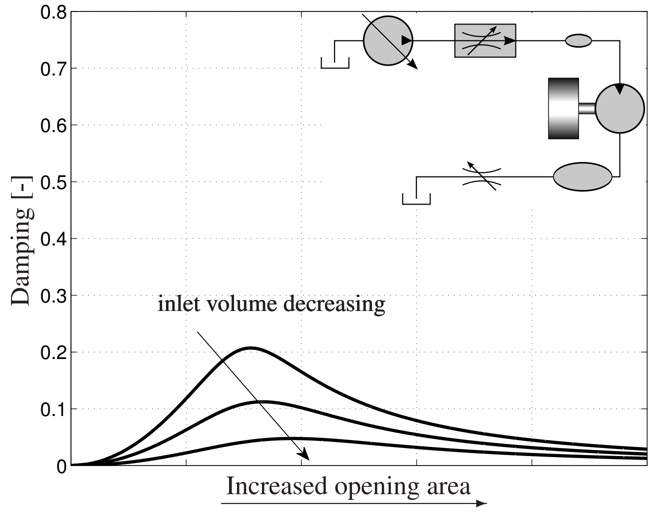

Figure 7: The damping will be low if the inlet volume is small compared to the outlet volume. This will be the case if the flow direction changes compared to figure 6.

To dimension the outlet orifice area in order to get a desired damping, the load needs to be known, see equation (10). In mobile hydraulic applications, the load situation varies greatly over time in a typical working cycle. It is therefore important to consider the load when designing the outlet orifice area. It is only possible to optimize the damping for a specific load. The damping will no longer be optimal if the load situation alters. In figure 8, it is shown how a load change will influence the damping. It is assumed that the outlet orifice area is dimensioned to obtain the highest possible damping for a nominal load. It is then shown how an increased/decreased load will affect the damping.

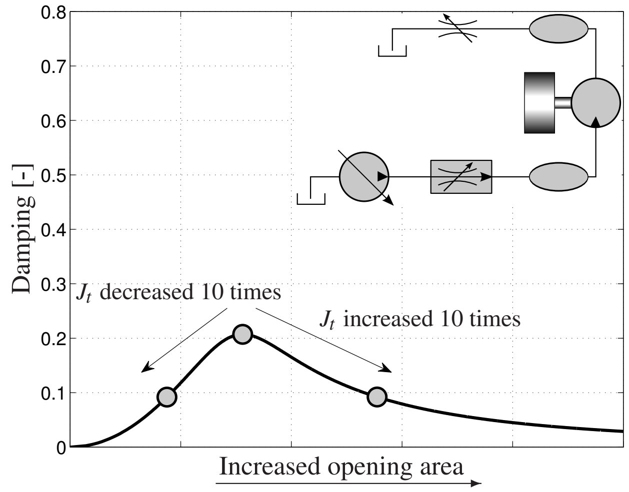

Figure 8: The outlet orifice area is designed to obtain the optimal damping for a nominal load. It is then shown how the damping will change if the load situation alters.

As can be seen in figure 8, the damping will decrease if the load changes. From an efficiency point of view, it is advantageous to have as large opening area of the outlet orifice as possible. The pressure drop across the valve can then be kept small, resulting in low losses. It is therefore recommended to design the opening area to be optimal for the minimum load. By doing so, the damping will be optimal for that load and then decrease for higher loads. Designing the opening area optimally for the minimum load maximizes the optimal outlet orifice area according to equations (10) and (13).

4 Linear Load

A system consisting of a constant flow valve, a linear load with a gear ratio and an outlet orifice, see figure 9, can be described by similar equations as the rotational load in section 3. Equations (14)-(17) describe the linearized system. The viscous friction in the cylinder has been ignored to simplify the analysis.

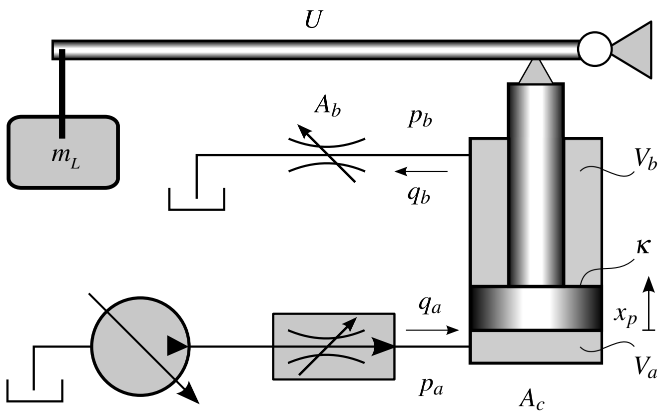

Figure 9: The system analysed in this section, a constant flow valve, a linear load with a gear ratio and an outlet orifice.

$$Q_a - A_c s X_p = \frac{V_a}{\beta_e} s P_a \tag{14}$$

$$U^2 m_L s^2 X_p = A_c P_a - \kappa A_c P_b - F_p \tag{15}$$

$$\kappa A_c s X_p - Q_b = \frac{V_b}{\beta_e} s P_b \tag{16}$$

$$Q_b = K_{c_b} P_b \tag{17}$$

The expression for K_{c_b} where maximum damping is obtained is shown in equation (18) and the maximum damping is calculated in the same ways as for a rotational load, see equation (12) [9].

$$K_{c_{b_{opt}}} = \kappa A_c \sqrt{\frac{V_b}{\beta_e U^2 m_L (\gamma - 1)}} \gamma^{3/4} \tag{18}$$

where

$$\gamma = 1 + \kappa^2 \frac{V_a}{V_b} \tag{19}$$

The outlet orifice area where optimal damping is obtained is derived according to equation (20).

$$A_{b_{opt}} = \sqrt{\frac{\kappa q_a K_{c_{b_{opt}}} \rho}{C_q^2}} \tag{20}$$

The difference between rotational and linear loads is that the volumes on the inlet and outlet side of the load will change during the stroke in case of a linear load. In the following example, a dead volume of 20% is assumed on each side of the piston. While the piston moves upwards, the inlet volume will increase and the outlet volume decrease. The damping will therefore increase during the stroke. In figure 10, the damping as a function of the outlet orifice opening area is shown during a whole stroke.

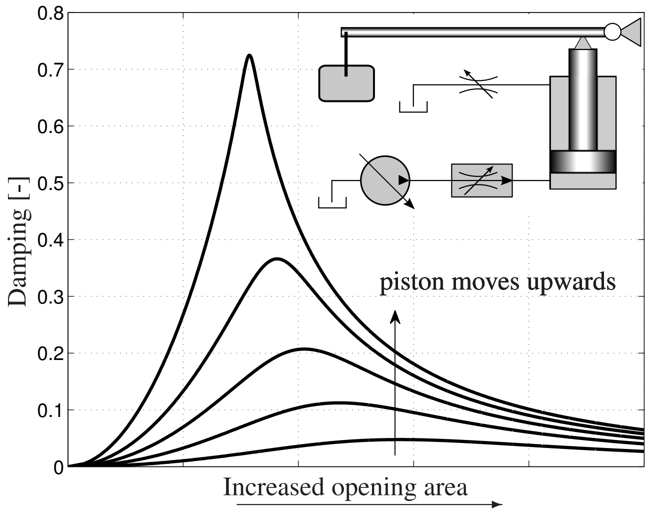

Figure 10: The damping will increase during the cylinder stroke. A dead volume of 20% on each side of the piston is assumed.

One design approach is to maximize the damping in the worst case scenario. In that case, the outlet orifice area should be dimensioned optimally at the piston's lower end position. The damping will then increase while the piston moves upwards, see the solid line in figure 11. It is possible to design the outlet orifice area smaller if a higher damping is required. However, drawbacks with that approach are slightly lower damping at the piston's lower end position, see the dashed line in figure 11, and higher losses across the orifice. A large opening area will reduce the losses at the outlet orifice. This, however, is at the expense of a lower damping, see the dashed-dotted line in figure 11. There is no point in designing the opening area to the left of the peaks in figure 10, since the damping will then be low while the losses are large. In applications where damping is important, it is appropriate to design the outlet orifice somewhere between the optimal area at the piston's lower end position and the optimal area at the piston's higher end position.

It is possible to obtain a higher damping than $\delta_h = 0.05$ at the piston's lower end position by increasing the dead volumes in the cylinder. For example, in case of 50% dead volume, the optimal damping would increase to $\delta_h = 0.11$ at the lower end position, see figure 12. Drawbacks with this approach are that the system response will decrease and that more space will be required. By increasing the dead volumes further, the volume change can eventually be ignored and the damping becomes constant, similar to the rotational load in figure 5.

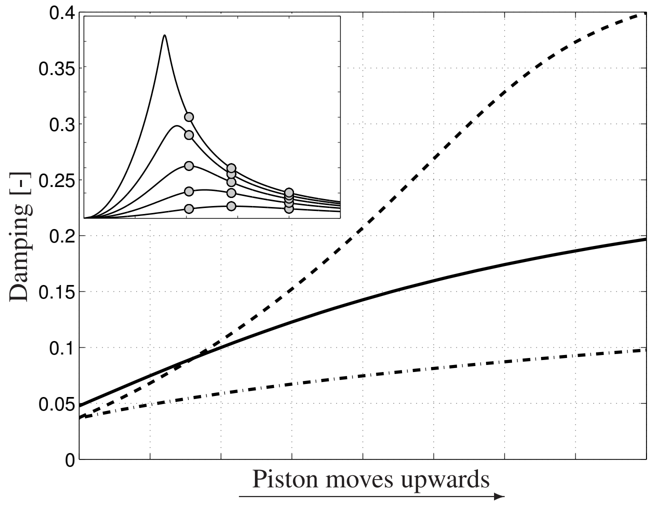

Figure 11: Three different designs of the outlet orifice area. The solid line shows an orifice area where the damping is optimized at the piston's lower end position, the dashed line shows a smaller orifice area and the dashed-dotted line a larger orifice area. The losses will increase with a decreased orifice area.

Similar to the rotational load case, the load needs to be taken into consideration when designing the outlet orifice, see equation (18). It is shown how a load change affects the damping in figure 13. The damping is assumed to be optimized for a nominal load when the piston is at its lower end position. It is then shown how an increased/decreased load will affect the damping.

For the linear load case, it is not obvious to design the opening area to be optimal for the minimum load. This is because the optimal opening area will change while the piston moves upwards. Designing the opening area to be optimal for the minimum load will minimize the losses since the opening area can be designed large, see equations (18) and (20). However, the damping will become small in case of higher loads according to figure 13. If the opening area is designed to be optimal for a higher load, it is possible to obtain a higher damping when the load decreases. It is, however, important to ensure that the damping does not shift to the left of the peaks in figure 13 when the load decreases. Drawbacks with designing the opening area optimally for a higher load are a larger pressure drop across the outlet orifice and lower damping at the piston's lower end position. To obtain the highest possible overall damping during the cylinder stroke, the outlet orifice area should be designed to be optimal somewhere between the minimum and maximum load.

5 Experimental Results

A test rig has been constructed to validate the damping contribution of the outlet orifice. It consists of a traditional pressure compensated valve on the inlet side, a cylinder with a mass load and a servo valve on the outlet side, see figure 14. Different designs of the outlet orifice can be achieved by controlling the opening area of the servo valve. A constant pressure pump

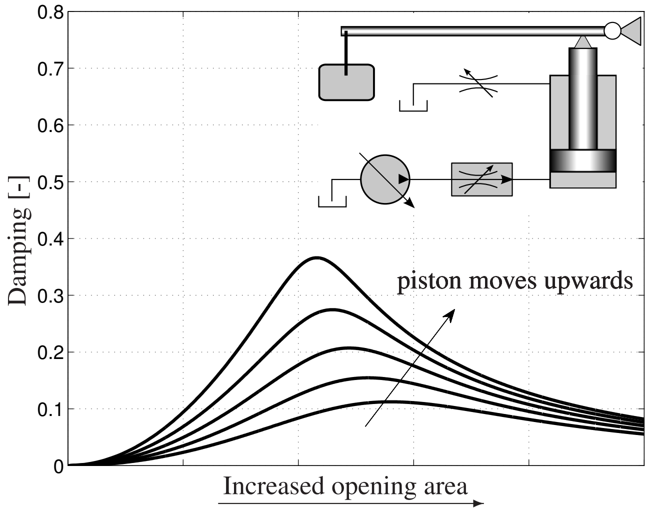

Figure 12: The damping in the worst case scenario can be increased by increasing the dead volumes in the cylinder. A dead volume of 50% on each side of the piston is assumed.

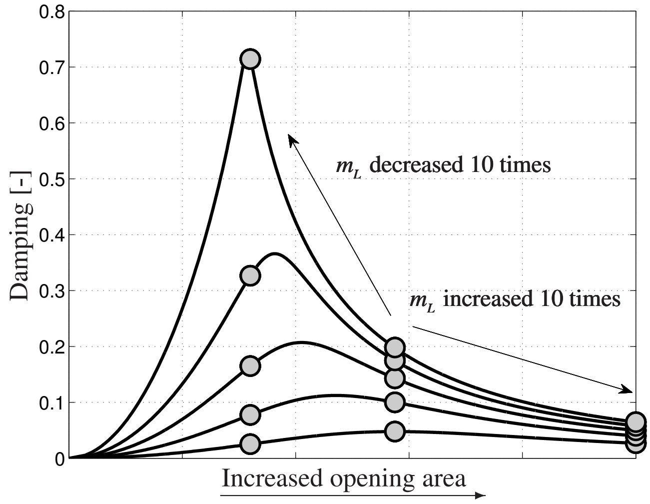

Figure 13: The outlet orifice area is designed to give the highest possible damping when the piston is at its lower end position. It is then shown how the damping will change if the load situation alters.

supplies the system. Pressure sensors are attached on the supply side and on both cylinder chambers. The cylinder and the servo valve are equipped with position sensors. External volumes are mounted on both sides of the piston. By using either one, it is possible to manipulate the dead volumes on either side of the piston.

In the experiments, a step is made in the flow by opening the inlet valve. Oscillations in the cylinder velocity are then studied. The experimental results are presented in figures 15-18. In tests **(a)** and **(b)**, there is a large volume on the inlet side, which means that a relatively high damping can be expected. In test **(a)**, the outlet orifice area is dimensioned close to the maximized damping. As can be seen in figure 15, there are almost no oscillations in the cylinder velocity. In test **(b)**, the outlet orifice area is larger than in test **(a)** and the damping becomes lower, see figure 16.

In test **(c)**, there is a large volume on the outlet side of the cylinder, which means that the damping is expected to be low. The outlet orifice area is dimensioned close to the maximized damping. Nevertheless, the damping is still low according to figure 17. This is consistent with the mathematical analysis according to equations (12) and (19).

In test **(d)**, the outlet orifice area is so large that it can be equated with having no outlet orifice at all. Theoretically, the hydraulic system will not contribute any damping without an outlet orifice, as shown in figure 2. This is almost the case in the measurements as can be seen in figure 18. The damping that is still obtained is due to secondary effects ignored in the mathematical analysis, such as friction and leakage.

6 Design Rules

This section summarizes the findings from sections 3 and 4 concerning how a pressure compensated hydraulic system should be designed in order to obtain a high damping contribution from the outlet orifice in the directional valve. Some

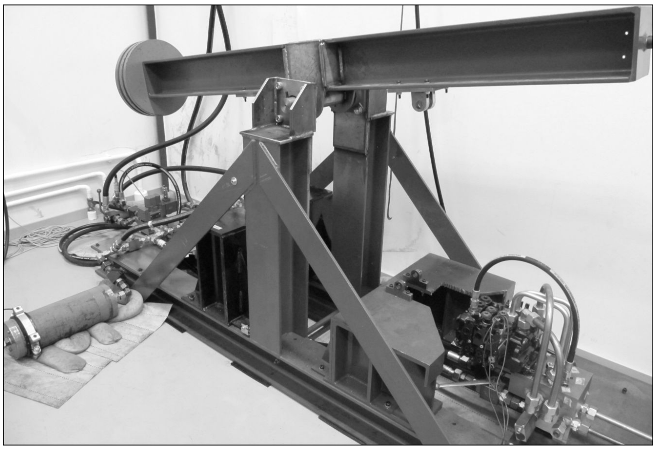

Figure 14: The experimental test stand. The pressure compensated valve can be seen at the lower right and one of the volumes to the left.

design rules are general and some are specific depending on whether the load is rotational or linear.

Design the outlet orifice area correctly There exists an optimal orifice dimension; both smaller and larger orifice areas give lower damping. The outlet orifice area which gives the highest possible damping can be calculated using equations (10) and (13) for a rotational load and equations (18) and (20) for a linear load.

Consider the volumes on each side of the load The damping will be higher if the inlet volume is large compared to the volume on the outlet side of the load. In case of a linear load, the volumes will change during the cylinder stroke. The outlet orifice can only be designed optimally for a specific volume. It is appropriate to design the outlet orifice optimally somewhere between the piston's lower and higher end positions.

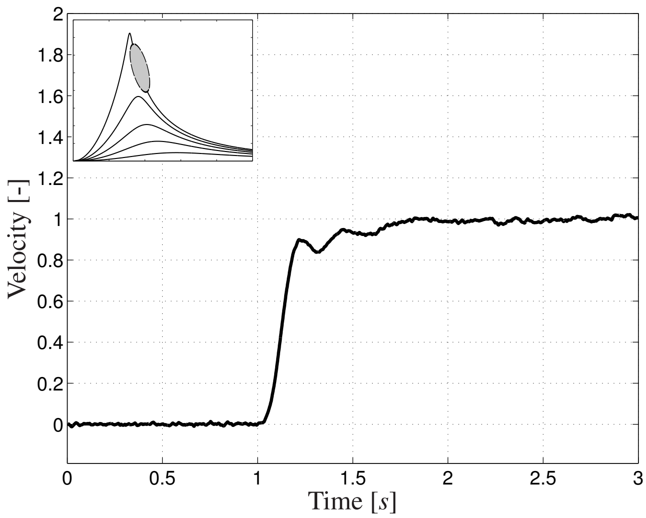

*Figure 15: Test **(a)**: A high damping is obtained when there is a large volume on the inlet side of the cylinder and the outlet orifice is designed close to its optimum.*

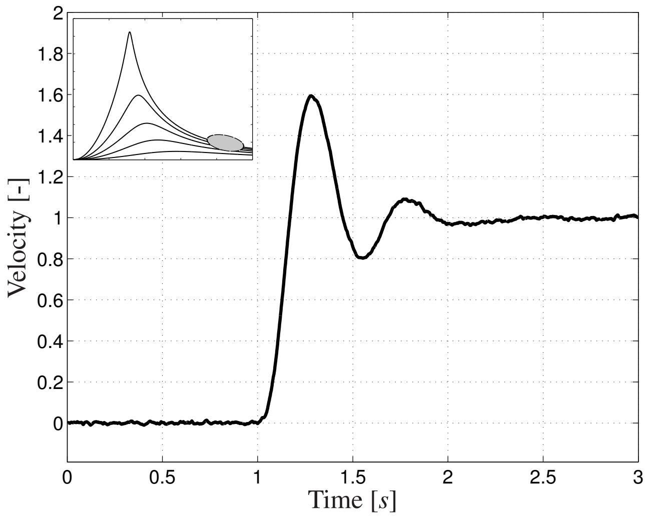

*Figure 16: Test **(b)**: The damping becomes lower when there is a large volume on the inlet side and the outlet orifice area is too large.*

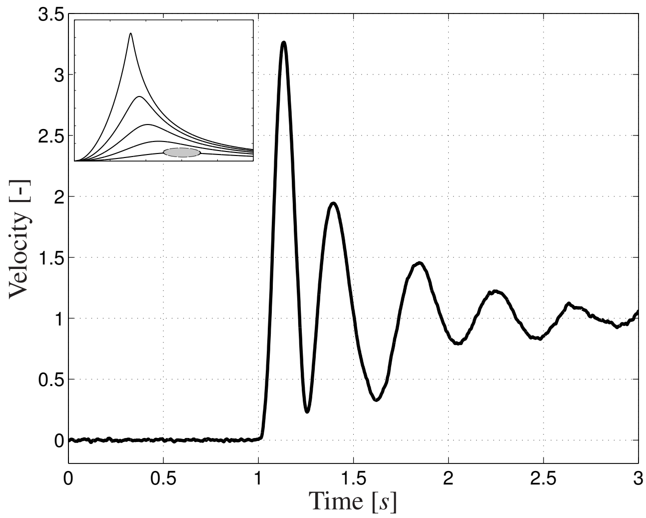

*Figure 17: Test **(c)**: When there is a large volume on the outlet side of the cylinder, the damping becomes low even if the outlet orifice is designed close to its optimum.*

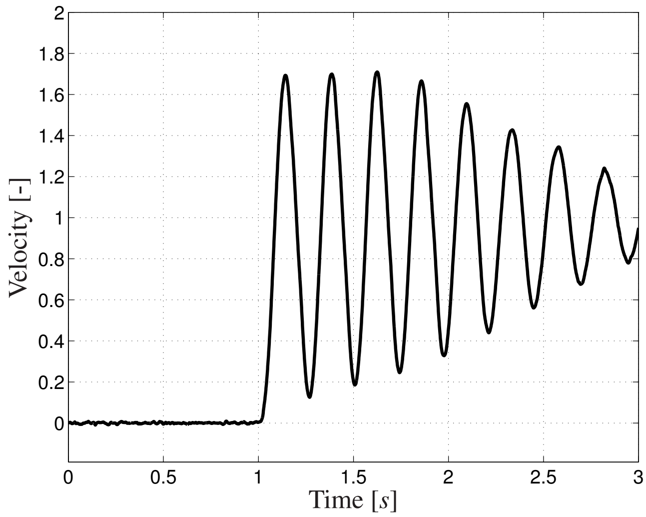

*Figure 18: Test **(d)**: Without an outlet orifice, only secondary effects such as friction and leakage will contribute to the damping.*

Consider load changes The size of the load will affect the optimal orifice area. For a rotational load, the orifice should be designed optimally for the minimum load. For a linear load, the orifice should be designed optimally somewhere between the minimum and maximum load.

Consider the losses When improving the damping, it is important to consider the losses. The final valve design will always be a compromise between high damping and low losses.

7 Conclusions

A hydraulic system consisting of a pressure compensated valve, a load and an outlet orifice has been studied in this paper. Without the outlet orifice, the hydraulic system will not contribute with any damping at all. Also with an outlet orifice, the damping might become low. However, it is possible to obtain a high damping contribution from the outlet orifice in the directional valve if the system is designed correctly.

To obtain a high damping, the outlet orifice area should be designed properly. There exists an optimal orifice dimension; both smaller and larger orifice areas give lower damping. It is also important to consider the volumes on each side of the load. The damping will be higher if the volume on the inlet side of the load is large compared to the outlet volume. It is possible to obtain a high damping in one flow direction by having a large inlet volume. However, the damping will be low for the other flow direction since the outlet volume is large in that case.

It is important to consider the losses while improving the damping. Some combinations of system parameters might give a high damping but also unrealistically high power losses. This is a well-defined optimization problem which will give different optimal system designs depending on the penalty factors for the damping and the energy efficiency.

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Nomenclature

The quantities used in this paper are listed in the table. Capital letters are used for linearized and Laplace transformed variables.

Quantity	Description	Unity
A_b	Outlet orifice opening area	m^2
$A_{b_{opt}}$	Outlet orifice opening area which gives the highest damping	m^2
A_c	Cylinder area	m^2
C_q	Flow coefficient	-
D_m	Hydraulic motor displacement	m^3/rad
F_p	External force	N
J_t	Rotational inertia load	kg m^2
K_{c_b}	Flow-pressure coefficient for the outlet orifice	m^3/Pa s
$K_{c_{b_{opt}}}$	K_{c_b} which gives the highest damping	m^3/Pa s
m_L	Load mass	kg
p_a	Pressure on the inlet side of the load	Pa
p_b	Pressure on the outlet side of the load	Pa
q_a	Flow into the load	m^3/s
q_b	Flow out of the load	m^3/s
T_m	External torque	Nm
U	Mechanical gear ratio	-
V_a	Volume on the inlet side of the load	m^3
V_b	Volume on the outlet side of the load	m^3
x_p	Piston position	m
β_e	Bulk modulus	Pa
γ	Parameter	-
δ_h	Damping	-
$\delta_{h_{max}}$	Maximum damping	-
θ_m	Rotational angle	rad
κ	Cylinder area ratio	-
ρ	Density	kg/m^3