Optimized Damping in Cylinder Drives Using the Meter-out Orifice - Design and Experimental Verification

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Abstract

This paper analyses the damping of a flow controlled cylinder with a mass load and an outlet orifice. By using linear models, a mathematical expression for the damping is derived. It is shown that the volumes on each side of the piston 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; both smaller and larger orifice areas give low damping independently of the size of the volumes. In this paper a design is proposed of the outlet orifice area that optimizes the damping of the system. Experimental results which confirm the theoretical expectations are also presented. The conclusions are that without an outlet orifice, the hydraulic system will not contribute with any damping at all. Furthermore, large dead volumes in the cylinder will increase the damping, but at the expense of the system's efficiency.

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 and closed centre. Open centre valves are often associated with constant flow systems. All pump flow goes through the open centre channel while in closed position. Closed centre valves are commonly used in systems where the pump is actively controlled, for example in constant pressure and load sensing systems /1/. While in neutral position, no flow passes through the valve.

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. A common solution to this problem is to equip the valves with pressure compensators /2/, /3/. The pressure drop across all the control 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. This is because a change in load pressure will not cause a change in load flow. Except for secondary effects such as leakage, stiffness of the compensator spring and friction in the cylinder, only the meter-out orifice of the directional valve will provide damping to the system. Damping is especially important in applications where large inertia loads are to be controlled, for example the swing function of a crane or an excavator.

It is possible to obtain damping on the meter-in side even though the valve is pressure compensated. For example, by a load pressure feedback affecting the main spool position /4/ or by manipulation the force balance of the pressure compensator /5/. An overview about active damping in mobile machines can be found in /6/.

This paper analyses the damping in systems using closed centre spool valves and pressure compensators placed upstream of the directional valve. No flow-pressure dependency on the inlet side is assumed. The pump controller can be of any type; it does not affect the analysis. Experimental results, which confirm the theoretical expectations, are also presented. The most important contribution of this paper is a mathematical analysis describing the damping of a flow controlled cylinder with an outlet orifice. A design of the outlet orifice to optimize the damping of the complete system is also proposed.

2. Mathematical Model

In a conventional pressure compensator, the reduced pressure acts on one end of the compensator and the load pressure together with a preloaded spring on the other, see **figure 1**. Linearization around a working point give equations (1)-(3). The dynamics of the compensator spool and the volume between the compensator and the directional valve are considerably faster than the rest of the system and are therefore ignored.

$$Q_s = K_{q_{PC}} X_{PC} + K_{c_{PC}} (P_p - P_m) \tag{1}$$

$$Q_s = K_{c_a} (P_m - P_a) \tag{2}$$

$$(P_a - P_m) A_{PC} = k_{PC} X_{PC} \tag{3}$$

The transfer function from pressure difference, $P_p - P_a$, to flow into the cylinder, Q_s, can be derived according to equation (4).

$$\frac{Q_s}{P_p - P_a} = \frac{K_{c_a} K_{c_{PC}}}{K_{c_a} + K_{c_{PC}} + \frac{K_{q_{PC}} A_{PC}}{k_{PC}}} \tag{4}$$

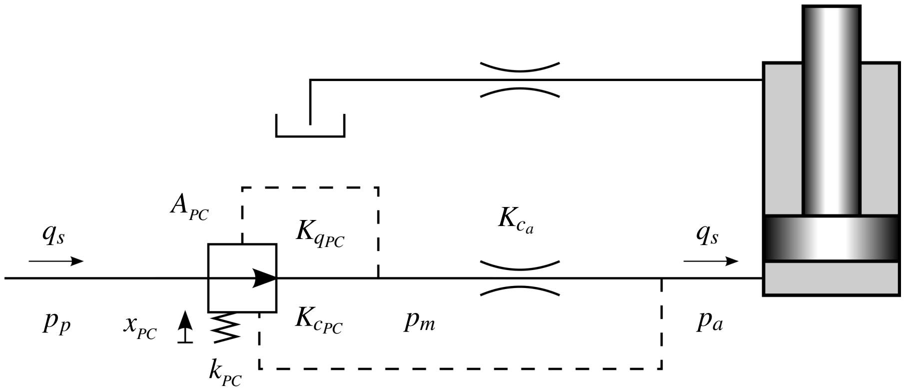


Figure 1: The meter-in part of the valve consists of a pressure compensator placed upstream of the directional valve.

If an ideal pressure compensator is assumed, which means that $k_{PC} \to 0$, $\frac{Q_s}{P_p - P_a} \to 0$ and the valve will have no flow-pressure dependency. In this paper, an ideal pressure compensator is assumed, which means that the meter-in part of the valve can be considered to be a perfect flow source. The system analyzed in this paper consists of a flow controlled cylinder with a mass load and a meter-out orifice, see **figure 2**. Equations (5)-(8) describe the linearized system. The viscous friction in the cylinder has been ignored to simplify the analysis.

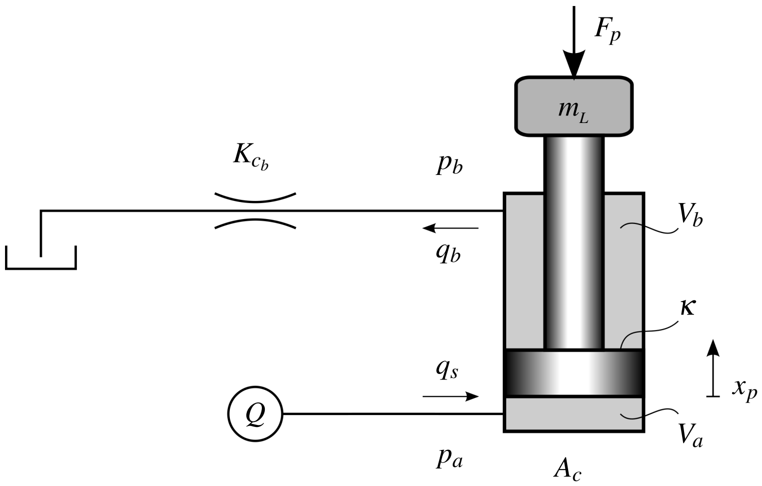


Figure 2: The system analyzed in this paper consists of a flow controlled cylinder with a mass load and a meter-out orifice. The meter-in part of the valve is considered to be a perfect flow source.

$$Q_s - A_c s X_p = \frac{V_a}{\beta_e} s P_a \tag{5}$$

$$m_L s^2 X_p = A_c P_a - \kappa A_c P_b - F_p \tag{6}$$

$$\kappa A_c s X_p - Q_b = \frac{V_b}{\beta_e} s P_b \tag{7}$$

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

By introducing the parameters in equations (9)-(11), a cubic polynomial is derived according to equation (12).

$$\gamma = 1 + \kappa^2 \frac{V_a}{V_b} \tag{9}$$

$$\omega_o = A_c \sqrt{\frac{\beta_e}{m_L V_a}} \tag{10}$$

$$\omega_b = \frac{\beta_e K_{c_b}}{V_b} \tag{11}$$

$$\left(\frac{s^3}{\omega_o^2 \omega_b} + \frac{s^2}{\omega_o^2} + \frac{\gamma s}{\omega_b} + 1 \right) s X_p = \left(1 + \frac{s}{\omega_b} \right) \left(\frac{1}{A_c} Q_s - \frac{V_a}{A_c^2 \beta_e} s F_p \right) \tag{12}$$

This expression can be factorized. The interesting case is when we have a resonance and a break frequency /7/, see equation (13).

$$\left(1 + \frac{s}{\omega_n} \right) \left(\frac{s^2}{\omega_h^2} + \frac{2\delta_h s}{\omega_h} + 1 \right) = \frac{s^3}{\omega_o^2 \omega_b} + \frac{s^2}{\omega_o^2} + \frac{\gamma s}{\omega_b} + 1 \tag{13}$$

ω_n, ω_h and δ_h can be identified according to equations (14)-(16).

$$\frac{\omega_b}{\omega_o} = \frac{\omega_b}{\omega_n} \sqrt{\frac{\gamma - \frac{\omega_b}{\omega_n}}{\frac{\omega_b}{\omega_n} - 1}} \tag{14}$$

$$\frac{\omega_h}{\omega_o} = \sqrt{\frac{\omega_b}{\omega_n}} \tag{15}$$

$$\delta_h = \frac{1}{2} \sqrt{1 + \gamma - \frac{\omega_b}{\omega_n} - \gamma \frac{\omega_n}{\omega_b}} \tag{16}$$

The maximum damping is found when

$$\frac{\omega_b}{\omega_n} = \sqrt{\gamma} \tag{17}$$

$$\frac{\omega_b}{\omega_0} = \gamma^{3/4} \tag{18}$$

which results in

$$\delta_{h_{max}} = \frac{1}{2} \left(\sqrt{\gamma} - 1 \right) \tag{19}$$

Equation (19) shows that the maximum damping of the system depends only on the value of γ, which includes the volume on each side of the cylinder and the cylinder area ratio according to equation (9). The maximum damping is thus dependent only on geometrical parameters. With equations (10), (11) and (18), the optimal expression for the flow-pressure coefficient of the meter-out orifice, K_{c_b}, can be derived. In this paper, the optimal expression for the flow-pressure coefficient is the K_{c_b} value which gives the highest damping of the system.

$$K_{c_{b_{opt}}} = \kappa A_c \sqrt{\frac{V_b}{\beta_e m_L (\gamma - 1)}} \gamma^{3/4} \tag{20}$$

According to equation (20), the optimal value of K_{c_b} will change during a cylinder stroke due to the change in cylinder volumes. Section 3. describes how the damping of the system is affected when the piston moves. A design of the shape of the meter-out orifice to make K_{c_b} valve-stroke independent is also proposed. Simulation results are shown in section 4. and experimental verifications are made in section 5..

3. Damping

A conventional spool valve will have a constant K_{c_b} value during a cylinder stroke provided that the flow is constant. According to equation (20), the optimal value of K_{c_b} will change during a stroke because of changed cylinder volumes. The maximum damping of the system will be low when the cylinder is at its lower position (when V_a is small), see equations (19) and (9). It is therefore appropriate to choose $K_{c_b} = K_{c_{b_{opt}}}$ for the maximum value of V_b. The damping then becomes higher for other positions of the piston, see **figure 3**. It can also be seen that a larger area ratio increases the damping of the system.

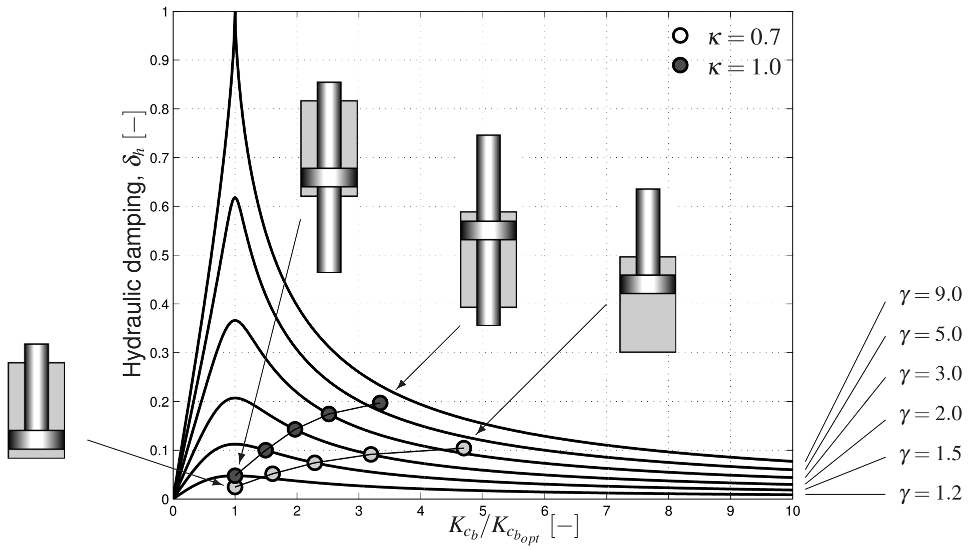


Figure 3: System damping during a cylinder stroke. The valve is designed so that $K_{c_b} = K_{c_{b_{opt}}}$ for the maximum value of V_b. When the cylinder moves, γ increases and the damping gets higher. At the same time however, $K_{c_{b_{opt}}}$ decreases, which means that the valve no longer has an optimal K_{c_b} value. A bigger area ratio increases the damping of the system. Each damping curve corresponds to a constant value of γ, specified on the right-hand side. A dead volume of 20% on both sides of the piston is assumed.

3.1. Shape of the Outlet Orifice

When using a conventional spool valve, the inlet and outlet orifices are coupled. So far, it has been assumed that the directional valve is designed so that $K_{c_b} = K_{c_{b_{opt}}}$ for the maximum value of V_b. For this assumption to be valid during all flow conditions, K_{c_b} has to be valve-stroke independent. Equations (21) and (22) describe the flow and the flow-pressure coefficient for the meter-out orifice and equation (23) the static relationship between the flow in and out of the cylinder.

$$q_b = C_q A_b \sqrt{\frac{2}{\rho} p_b} \tag{21}$$

$$K_{c_b} = \frac{\partial q_b}{\partial p_b} = \frac{C_q A_b}{\sqrt{2\rho p_b}} \tag{22}$$

$$q_b = \kappa q_s \tag{23}$$

Assume that the shape of the inlet and outlet orifices can be described by exponential functions according to equations (24) and (25). The inlet orifice is assumed to be ideally pressure compensated and the flow is therefore directly proportional to the opening area of the valve.

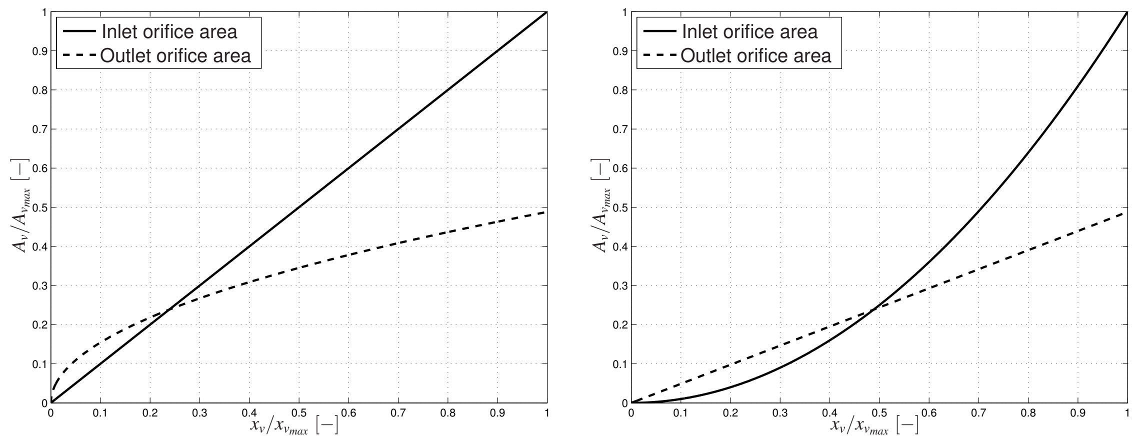
$$q_s = C_1 x_v^n \tag{24}$$

$$A_b = C_2 x_v^m \tag{25}$$

Equations (21)-(25) give an expression for K_{c_b}, including the valve opening position for both the inlet and outlet orifices.

$$K_{c_b} = \frac{C_q^2 C_2^2 x_v^{2m-n}}{\rho \kappa C_1} \tag{26}$$

According to equation (26), the valve should be designed so that $2m = n$ in order to obtain a constant K_{c_b} value independent of the valve opening position. By doing so, the K_{c_b} value will be valve-stroke independent. The special case when the flow into the cylinder is directly proportional to the valve opening position ($n = 1$) gives that the outlet orifice



(a) The inlet area is directly proportional to the valve opening position ($n = 1$) and the outlet area is proportional to the square root of the opening position ($m = 1/2$).

(b) The inlet area is proportional to the valve opening position in square ($n = 2$) and the outlet area is directly proportional to the opening position ($m = 1$).

Figure 4: The shape of the outlet orifice is designed so that K_{c_b} is valve-stroke independent.

area should be proportional to the square root of the inlet area ($m = 1/2$). In **figure 4**, two different shapes are presented. In reality, the outlet orifice is also dimensioned to handle, for example, an over-running load.

3.2. Loss Analysis

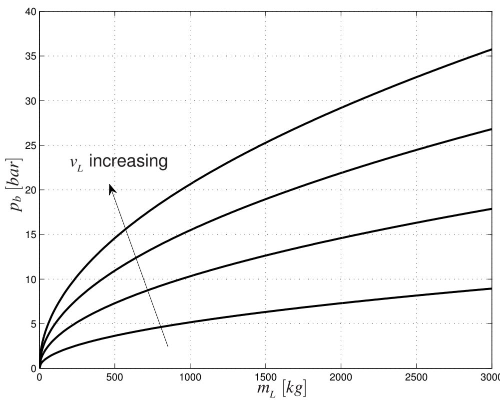
Assume that the valve is designed so that $K_{c_b} = K_{c_{b_{opt}}}$ for the maximum value of V_b, as in figure 3. Also, assume that the shape of the outlet orifice is designed so that K_{c_b} is valve-stroke independent as in figure 4. A mechanical gear ratio between the cylinder and the load, like in a crane, is also assumed. An expression for the pressure drop across the outlet orifice, p_b, can then be derived using equations (9) and (20)-(22). The total system efficiency can be derived using the output power and the power losses, which are defined according to equations (28) and (29). Only the power losses across the outlet orifice are considered.

$$p_b = \frac{\kappa v_L}{2V_{b_{max}}\gamma_{min}^{3/4}}\sqrt{\beta_e m_L V_{a_{min}}} \tag{27}$$

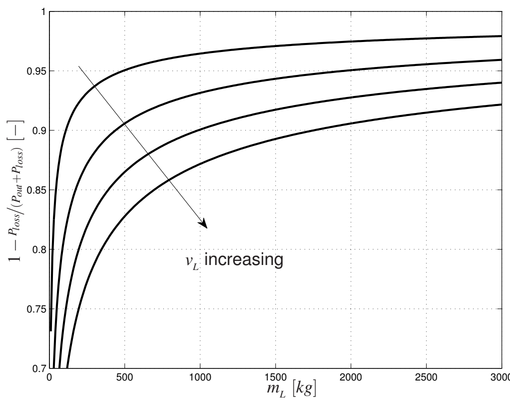
$$P_{out} = m_L g v_L \tag{28}$$

$$P_{loss} = p_b \kappa q_s = p_b \kappa \frac{A_c v_L}{U} \tag{29}$$

For realistic parameter values, the pressure drop across the meter-out part of the valve becomes rather small for low velocities, see **figure 5a**. For high velocities, however, the pressure drop is high and the total system efficiency decreases, see **figure 5b**. Even though a high load mass requires a high pressure drop, the efficiency is still rather high.



(a) The pressure drop across the outlet orifice as a function of the load mass for different load velocities.



(b) The total efficiency of the system as a function of the load mass for different load velocities. Only the power losses at the outlet orifice are considered.

Figure 5: Losses in the system when the outlet orifice is designed according to figure 3. The relationship between the inlet and outlet orifices is as shown in figure 4. The parameters used in the figure are $A_c = 0.01\ m^2$, $V_{a_{min}} = 1 \cdot 10^{-3}\ m^3$, $V_{b_{max}} = 5 \cdot 10^{-3}\ m^3$, $U = 10$, $v_L = [0.25\ 0.5\ 0.75\ 1.0]\ m/s$, $\beta_e = 1 \cdot 10^9\ Pa$, $\kappa = 0.7$.

A high mechanical gear ratio decreases the power losses according to equation (29). The maximum values of the system pressure and the cylinder area ultimately set the design limit for the gear ratio. Larger dead volumes in the cylinder will increase the pressure losses according to equation (27).

4. Simulation Results

The next generation of the simulation program Hopsan is used to perform the simulations /8/, /9/. The model consists of a flow controlled cylinder with a mass load and an outlet orifice. After one second, a step is made in the inlet flow and the piston starts to move. The simulation results are shown in **figure 6**. Different designs of the outlet orifice are tested.

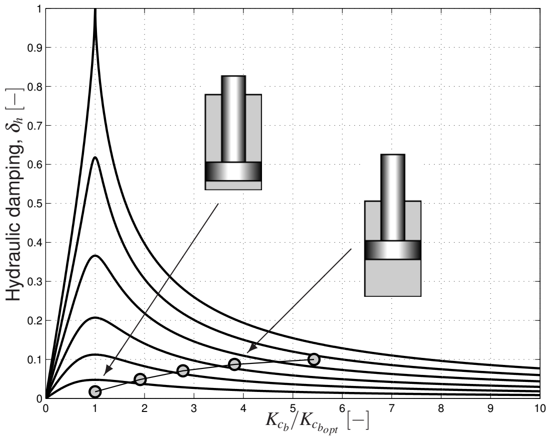
In the first design, the outlet orifice is designed to obtain an optimized damping when the piston is at its lower position, see **figure 6a**. However, even though the damping is as good as it gets, there are a lot of oscillations when the cylinder starts at its lower position, see **figure 6c**. When starting at a higher position, the damping is higher and there are relatively few oscillations according to **figure 6e**, even though the damping is not optimized for that position of the piston.

It is also interesting to investigate how a non-optimized design of the outlet orifice will affect the damping of the system. Two different designs are shown in **figure 6b**. In one of them, the outlet orifice area is too small, which means that the damping will be very low at small values of γ, as can be seen in **figure 6d**. The pressure drop across the outlet orifice will also increase, which means higher energy losses. Another scenario is that the outlet orifice area is too large, in which case the damping will not increase much when the piston moves, see figure 6b. This results in a low damping, not only for small values of γ, but also for larger values, according to **figure 6f**.

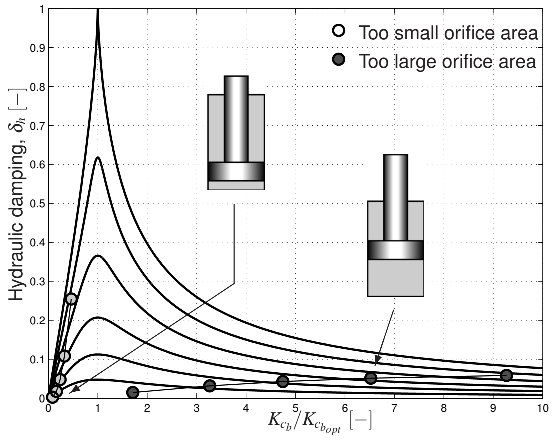
5. Experimental Verification

To verify the mathematical model and the simulation results, measurements have been performed on a real-world application. The test stand consists of a pressure compensated valve on the inlet side, a cylinder with a mass load and a servo valve on the outlet side, see **figure 7**. Different designs of the outlet orifice can be achieved by controlling the opening area of the servo valve. A constant pressure pump 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.

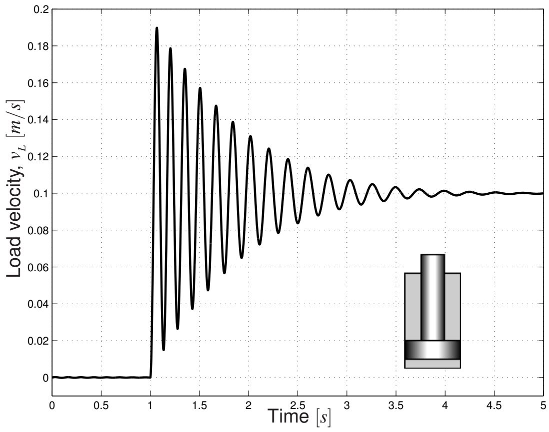
The experimental results are presented in **figure 8**. In **figures 8a** and **8b**, there is a large volume on the inlet side and in **figures 8c** and **8d**, the volume is large on the outlet side.



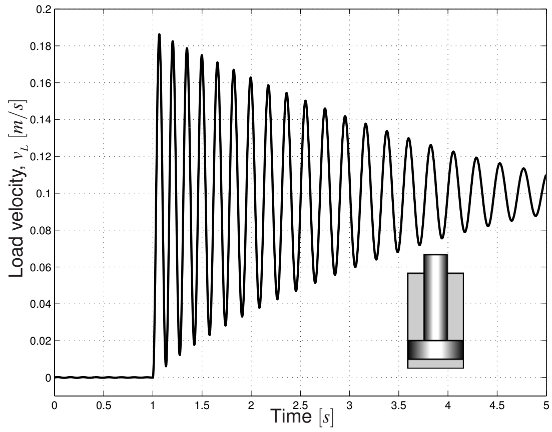
(a) In figures 6c and 6e, the outlet orifice has an optimized damping when the piston is at its lower position.



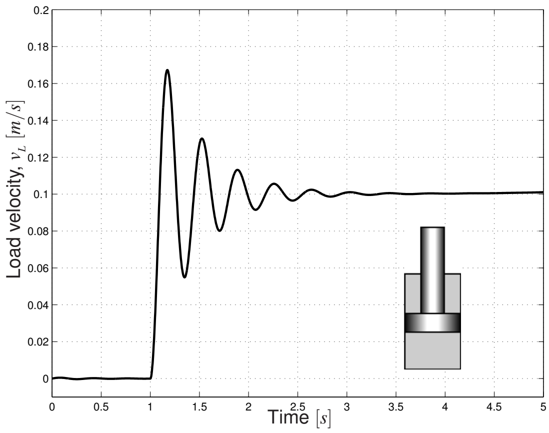
(b) In figure 6d, the outlet orifice area is too small and in figure 6f, it is too large.



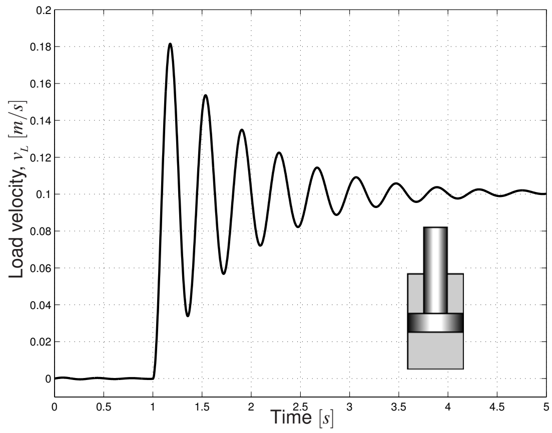
(c) The damping is low when the piston starts at its lower position, even though it is optimized for that position.



(d) There are a lot of oscillations if the piston starts at its lower position and the outlet orifice area is too small.



(e) The damping is relatively good when the piston starts at a higher position, even though it is not optimized for that position.



(f) Even though the piston starts at a higher position, the damping is relatively bad if the outlet orifice area is too large.

Figure 6: Simulation results for different designs of the outlet orifice.

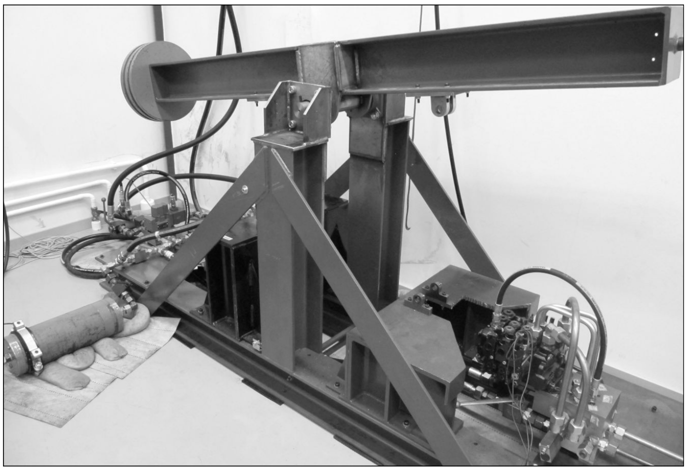
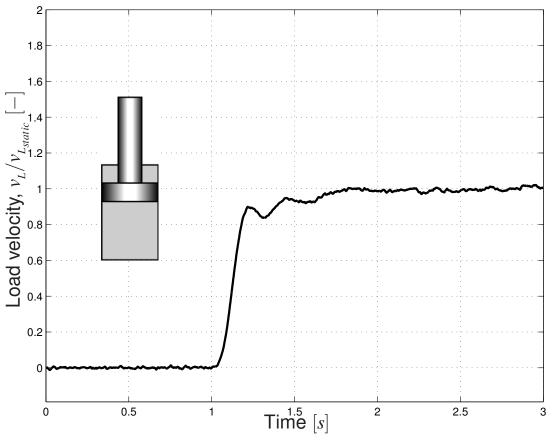
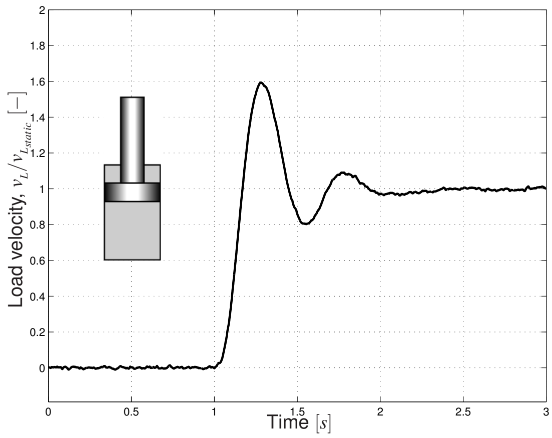


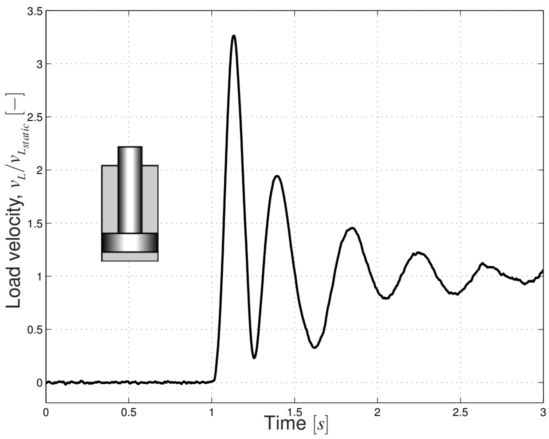
Figure 7: The experimental test stand. The pressure compensated valve can be seen at the lower right and one of the volumes to the left.



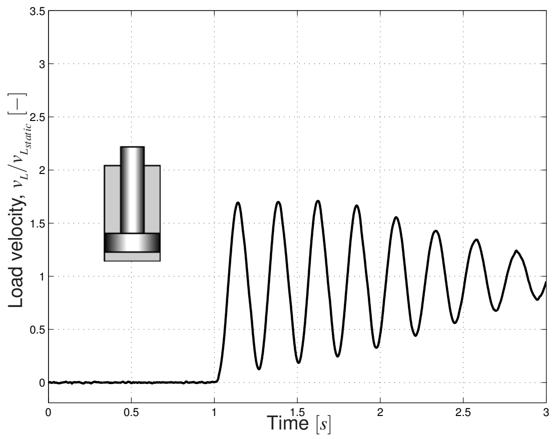
(a) The damping is good when V_a is large and the outlet orifice is is not too large.



(b) If the outlet orifice area is too large, the damping is lower. V_a is large.



(c) In case of a large V_b, the damping is low no matter the size of the outlet orifice area. This plot shows the optimized damping.



(d) This is an example of when V_b is large and the outlet orifice area is not optimal. The damping is very low.

Figure 8: Experimental results for different designs of the outlet orifice.

The damping is much better in case of a large inlet volume, which is expected since the value of γ is high. However, as can be seen in figure 8b, a too large outlet orifice area results in a low damping even though γ is high. When the outlet volume is large the damping will be low. Figure 8c shows the case when the damping is optimized. Nevertheless, the system is oscillative. If γ is low and the outlet orifice area is poorly designed, almost no damping is obtained. This is the case in figure 8d.

6. Discussion

The analysis made in this paper is valid for several hydraulic systems frequently used in mobile machines. The requirement is that the inlet side of the valve can be considered to be a constant flow source. This is true for all systems using pressure compensators, which include constant pressure and load sensing, but also, for example, independent metering systems /10/. Valveless systems using one dedicated pump for each function are also covered /11/. It is, however, important to bear in mind that secondary effects such as leakage and friction are ignored in this analysis.

The actuator in the studied system layout could be replaced by a hydraulic motor with an inertia load. In that case, the volumes on both sides of the motor will always be constant. The outlet orifice area could therefore be designed to obtain a desired damping without considering changed dynamics in the system.

7. Conclusions

A flow controlled cylinder with a mass load and an outlet orifice has been studied in this paper. Without the outlet orifice, the hydraulic system will not contribute with any damping. In that case, only secondary effects such as leakage and cylinder friction will provide damping to the system. This is the case for example in purely displacement controlled systems using a dedicated pump for each function.

The maximum damping of the system is only dependent on the volumes on each side of the piston and the cylinder area ratio, which are all geometrical properties. When the inlet cylinder volume is small compared to the outlet, the damping of the system is low regardless of the outlet orifice opening area. Large dead volumes in the cylinder will increase the damping. It is however at the expense of system efficiency.

When the piston moves, the optimal opening area of the outlet orifice changes with the change in cylinder volumes. If the inlet and outlet orifices are decoupled, as in individual metering systems, it would be possible to control the outlet orifice in order to obtain the highest possible damping during the whole stroke. However, when using conventional spool valves, the outlet orifice can only be designed optimally for a specific position of the piston. By choosing the position when the inlet volume is minimized, the highest possible damping will be obtained at the worst case scenario. Nevertheless, the damping is still several times higher for other positions of the piston.

References

/1/ B.R. Andersson. A survey of load-sensing systems. *The* BFPR *Journal*, 13: 103–115, 1980.

/2/ H. Pettersson, P. Krus, A. Jansson, and J.-O. Palmberg. The design of pressure compensators for load sensing hydraulic systems. In UKACC *International Conference on Control'96*, 1996.

/3/ D. Wu, R. Burton, G. Schoenau, and D. Bitner. Analysis of a pressure-compensated flow control valve. *Journal of Dynamic Systems, Measurement, and Control*, 129:203–211, 2007.

/4/ B.R. Andersson. Valves contribution to system damping. In *The 5th Scandinavian International Conference on Fluid Power (SICFP)*, 1997.

/5/ B. Eriksson, J. Larsson, and J.-O. Palmberg. A novel valve concept including the valvistor poppet valve. In *The 10th Scandinavian International Conference on Fluid Power (SICFP)*, 2007.

/6/ R. Rahmfeld and M. Ivantysynova. An overview about active oscillation damping of mobile machine structure. *International Journal of Fluid Power*, 5(2): 5–24, 2004.

/7/ H.E. Merritt. *Hydraulic Control Systems*. John Wiley & Sons, Inc., 1967.

/8/ B. Eriksson, P. Nordin, and P. Krus. Hopsan NG, a C++ implementation using the TLM simulation technique. In *The 51st Conference on Simulation and Modelling (SIMS)*, 2010.

/9/ M. Axin, R. Braun, A. Dell'Amico, B. Eriksson, P. Nordin, K. Pettersson, I. Staack, and P. Krus. Next generation simulation software using transmission line elements. In *Fluid Power and Motion Control (FPMC)*, pages 265–276, 2010.

/10/ B. Eriksson and J.-O. Palmberg. Individual metering fluid power systems: Challenges and opportunities. *Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering*, 225:196–211, 2011.

/11/ R. Rahmfeld and M. Ivantysynova. Displacement controlled linear actuator with differential cylinder – a way to save primary energy in mobile machines. In *Fifth International Conference on Fluid Power Transmission and Control (ICFP)*, volume 1, pages 316–322, 2001.

Nomenclature

A_b	Opening area of the outlet orifice	m^2
A_c	Cylinder area	m^2
A_{PC}	Compensator area exposed to control pressure	m^2
A_v	Opening area of the control valve	m^2
C_q	Flow coefficient	-
C_1	Parameter	m^2/s
C_2	Parameter	m
F_p	Disturbance force	N
k_{PC}	Compensator spring stiffness	N/m
K_{c_a}	Flow - pressure coefficient for the inlet side of the control valve	m^5/Ns
K_{c_b}	Flow - pressure coefficient for the outlet side of the control valve	m^5/Ns
$K_{c_{PC}}$	Flow - pressure coefficient for the compensator	m^5/Ns
$K_{q_{PC}}$	Flow gain coefficient for the compensator	m^2/s
m_L	Load mass	kg
p_a	Pressure on the piston side of the cylinder	Pa
p_b	Pressure on the piston rod side of the cylinder	Pa
P_{loss}	Power losses due to the outlet orifice	W
p_m	Pressure reduced by the compensator	Pa
P_{out}	Output power	W
p_p	Pump pressure	Pa
q_b	Flow out from the cylinder	m^3/s
q_s	Flow into the cylinder	m^3/s
s	Laplace variable	1/s
U	Mechanical gear ratio	-
V_a	Piston chamber volume	m^3
V_b	Piston rod chamber volume	m^3
v_L	Load velocity	m/s
x_{PC}	Compensator spool position	m
x_p	Piston position	m
x_v	Control valve spool position	m
β_e	Effective bulk modulus	Pa
γ	Parameter	-
δ_h	Hydraulic damping	-
κ	Cylinder area ratio	-
ρ	Density	kg/m^3
ω_b	System frequency	rad/s
ω_h	Hydraulic resonance frequency	rad/s
ω_n	Break frequency	rad/s
ω_o	System frequency	rad/s