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Novel concept for stabilising a hydraulic circuit containing counterbalance valve and pressure compensated flow supply

Jesper K. Sørensen, Michael R. Hansen and Morten K. Ebbesen

Faculty of Engineering and Science, Department of Engineering Sciences, University of Agder, Grimstad, Norway

ABSTRACT

In this paper, a novel concept for stabilising a hydraulic system containing a counterbalance valve and a pressure compensated flow supply is presented. The concept utilizes a secondary circuit where a low-pass filtered value of the load pressure is generated and fed back to the compensator of the flow supply valve. The novel concept has been investigated theoretically and experimentally. A linear model has been developed to verify the improved stability conditions. The novel concept has been implemented on a single boom actuated by a cylinder. The results show that the pressure oscillations in an otherwise unstable system can be suppressed with the novel concept. This happens without any compromise on the load independence of the flow supply but with some limitations on response time.

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Oscillations; counterbalance valve; pressure compensated valve; instabilities in hydraulic systems; load-holding application

1. Introduction

For safety reasons, hydraulic load carrying applications are required by law to contain a load holding protection device. The most widely used device is the counterbalance valve (CBV). It is multi-functional and provides leak tight load holding, load holding at hose/pump failure as well as shock absorption, overload protection, and cavitation prevention at load lowering.

However, it is well known that a series connection of a pressure compensator (CV), a directional control valve (DCV) and a CBV tends to introduce instability in a system, see Miyakawa (1978), Persson *et al.* (1989), Handroos *et al.* (1993), Zähe (1995) and Hansen and Andersen (2010). This is mainly a problem when the controlled actuator is subjected to a negative load, i.e. a load that tends to drive the actuator as a pump, because this will require the CBV to throttle the return flow, see Figure 1. This system will be referred to as the base circuit.

It is a major challenge within hydraulic system design to find solutions that offer stable handling of negative loads together with pressure compensated metering-in flow. Typically, practical solutions will compromise either the load independency, the response time or the level of oscillations (Nordhammer *et al.* 2012). The consequences of the oscillatory nature of such systems are reduced safety, reduced productivity as well as added fatigue load on both the mechanical and hydraulic system. The severity of oscillations is affected by a wide variety of parameters some of which are hard to predict or change: external load on the actuator, the damping and hysteresis of the CBV, the operator input as well as volumes and restrictions in the hydraulic lines.

The efforts to minimize the oscillatory nature of the base circuit can be divided into three groups:

- Parameters variations (pilot area ratio of CBV, pilot line orifices, etc.) on the circuit using same main components.
- Feedback control of DCV.
- Replace either the CBV or the pressure compensated DCV with alternatives.

The parameters most influential on the stability are the damping of the system and the pilot ratio of the CBV (Hansen and Andersen 2001). A more stable operation when lowering the load can be achieved by reducing the pilot ratio of the CBV. This will come at the expense of a higher pressure level and thereby cause a higher energy consumption – especially with small external loads.

Another approach is to add damping when designing the hydraulic circuit. Many ways of doing this have been tried over the years; adding volumes, adding orifices, adding logic valves, etc. Especially, interest has been focused on the pilot line to the CBV. By manipulating the pressure towards the CBV pilot port in different ways; add delays, create a difference in the path back and forth, positive effects can be achieved. Different commercial solutions are available, as for example the ones shown in Figure 2, from Bosch Rexroth (Bosch 2015) and NEM Hydraulic (NEM 2014). Common for

CONTACT Jesper K. Sørensen jesper.k.sorensen@uia.no

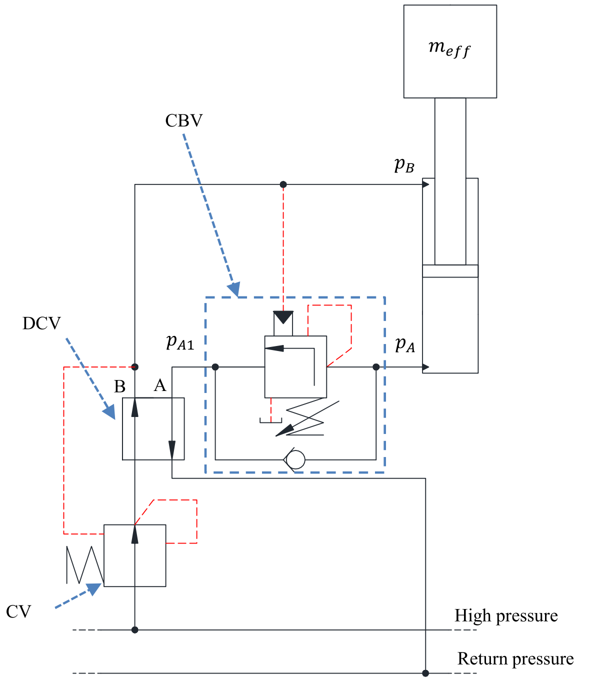

Figure 1. The base circuit consisting of a pressure compensator (CV), a directional control valve (DCV), and a counterbalance valve (CBV).

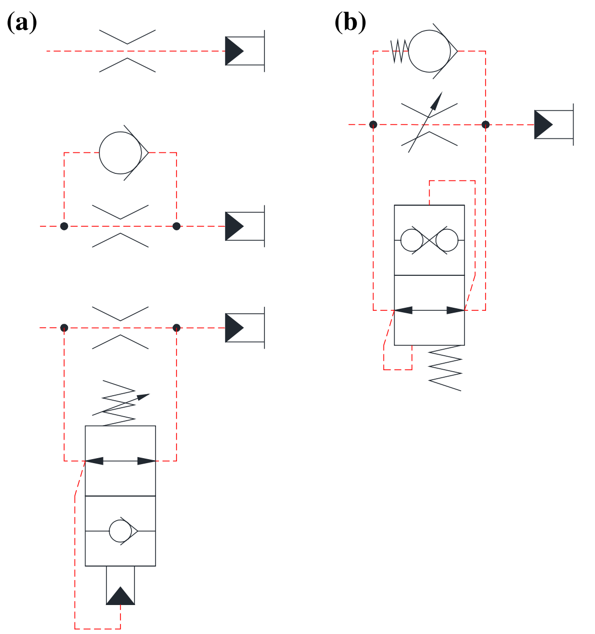

Figure 2. Examples of current solutions for damping the pilot line of the CBV. (a) is from Bosch Rexroth (Bosch 2015) and (b) is from NEM Hydraulic (NEM 2014).

these is that they are application and condition sensitive, especially towards temperature variations.

A different approach is to actively compensate for the oscillations by applying closed-loop control strategies that involve the input signal to the DCV and some kind of pressure feedback (Hansen and Andersen 2010, Cristofori *et al.* 2012, Ritelli and Vacca 2014). The most important limitation in these strategies is the bandwidth limitation in typical DCVs. Alternatively, the pressure compensator (CV) can be removed and the DCV replaced by a servo valve which is a proven and reliable method for motion control. The weakness here are in the investment costs and the difficulties in handling disturbances in the supply pressure caused by neighbouring circuits. Separate meter-in separate meter-out techniques are also a possibility. These utilise two or more actively controlled valves that opens up for the combination of flow and pressure control in the inlet and outlet ports. However these systems are, inherently, less reliable and their performance is sensitive to parameter variations (Pedersen *et al.* 2010).

The author has investigated the use of a DCV with compensated supply pressure, see Sørensen *et al.* (2014) and Sørensen *et al.* (2015). This is a commercially available alternative that is characterised by low cost but also load dependent flow. Another example is described in Nordhammer *et al.* (2012), where the main throttling ability is moved from the CBV to the return orifice of the DCV, thereby eliminating the oscillations. However, this is not a viable solution if the minimum load is 60% or less of the maximum load, which strongly minimizes the applicability.

All of the approaches have certain drawbacks as compared to the base circuit. In this paper, a novel concept is presented. It has the same steady state characteristics as the base circuit only without the corresponding oscillatory nature.

2. Novel concept

In Figure 3 a hydraulic diagram of the proposed novel concept is shown, patent pending (Hansen and Sørensen 2015). It is shown in a situation where the actuator is subjected to a negative load, i.e. a lowering motion of some gravitational payload.

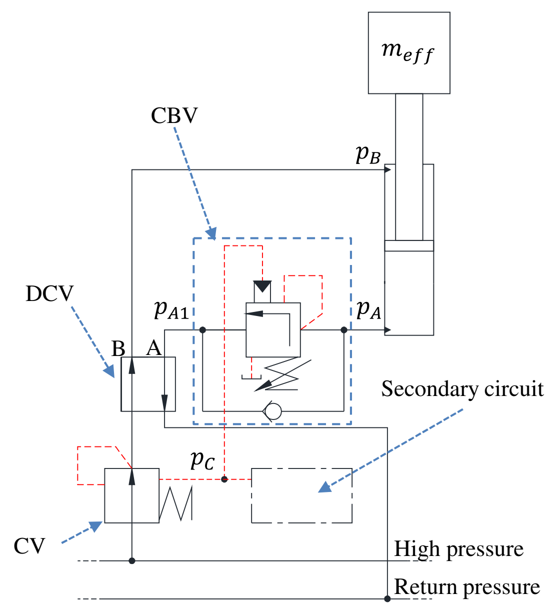

Figure 3. Hydraulic diagram of novel concept. Both the CBV and the CV are connected to the secondary circuit.

When compared to the base circuit in Figure 1, it can be seen that the pilot connections of both the CV and CBV are supplied by the secondary circuit rather than by the B-port pressure. The underlying idea is to suppress oscillations in the system by generating the steady state value of the B-port pressure in the secondary circuit.

The novel concept also encompasses solutions where the secondary circuit only is connected to the CV or the CBV. In fact, the solution where only the CV is connected to the secondary circuit, see Figure 4, is preferred from a reliability point of view because the CBV and the related safety functions are activated as usual.

The concept can, because the system works passively from the operators point of view, be combined with any closed loop control strategies on the DCV. In this paper, the concept presented is with a linear actuator, but the method will also work for rotational actuators in circuits with CBVs.

In Figure 4 the secondary circuit is shown in more detail. It is realised with an orifice and a proportional pressure relief valve (PPRV) in series. The intermediate pressure, p_C, will be referred to as the compensator pressure and it is connected to the CV.

The overall target is that p_C shall be the steady state value of p_B thereby suppressing oscillations of the compensator and, subsequently, in the entire system. For that purpose, a control strategy is suggested that requires the measurement of p_B, a low-pass filtering yielding a reference value for the compensator pressure, $p_{C,ref}$ and a measurement of said pressure, p_C. This allows for a closed loop control where the pressure setting of the PPRV is adjusted by means of a voltage signal, U, in order to continuously meet the reference value of the compensator pressure.

A block diagram of the used control strategy is shown in Figure 5.

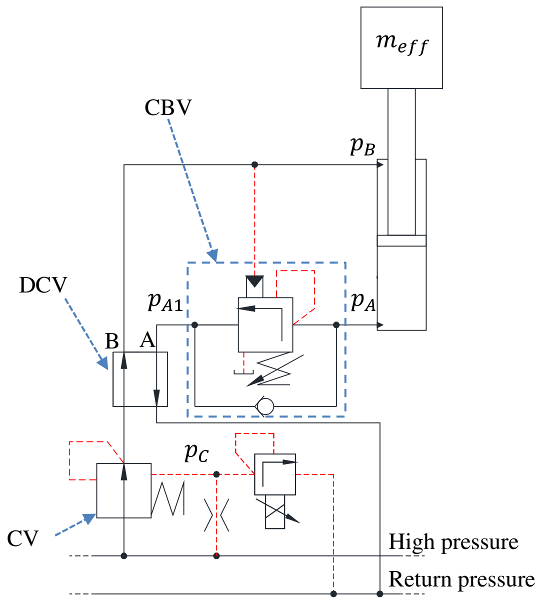

Figure 4. Hydraulic diagram of novel concept where only the CV is connected to the secondary circuit. The secondary circuit is shown in more detail.

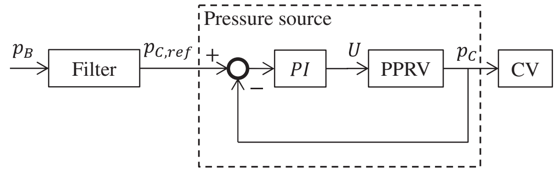

Figure 5. The proposed control strategy.

Clearly, a hydraulic filter circuit connecting p_B with p_C could be considered as an alternative, however, the concept of a closed loop control minimizes the uncertainties on parameters associated with small orifices and liquid dynamics.

In this setup, a PPRV instead of a proportional pressure reducing valve is chosen. The main reason for this is to minimise the pressure drop across the proportional unit because the compensator pressure typically will be 10–20% of the supply pressure.

3. Considered system

In order to examine the novel concept in more detail both linear stability analyses and experiments have been carried out. These investigations have been conducted on the same setup in the mechatronics laboratory at the University of Agder, see Figure 6. The setup consists of a hydraulically actuated boom and a control system.

The hydraulic diagram of the setup corresponds to the one in Figure 4.

The novel concept has been implemented using commercially available components. The DCV and CV are embedded in a pressure compensated 4/3-way DCV group from Danfoss (Model: PVG32). It has an electro-hydraulic actuation with linear flow vs. input signal characteristics with a maximum value of $Q_{max,DCV} = 25\frac{L}{min}$. The 4-port CBV is from Sun Hydraulics (Model: CWCA) with a 3:1 pilot area ratio and a rated flow of $Q_{r,CBV} = 60\frac{L}{min}$. The PPRV is from Bosch Rexroth (Model: DBETE) and has a crack pressure that varies linearly with the voltage input. At maximum signal, $U_{max} = 10$ V, the valve cracks open at $p_{C@0L/min} = 280$ bar and has a rated value at $p_{C@0.8L/min} = 315$ bar.

In Table 1 are listed some other important data for the experimental setup.

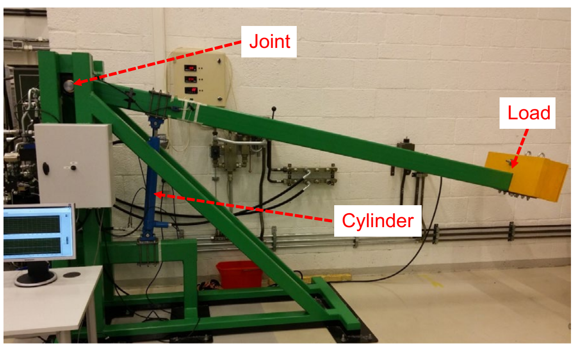

Figure 6. Hydraulic arm test setup.

Table 1. Important data of the experimental setup.

Parameter	Value
Distance from main hinge to mass centre of boom + load	$L = 3570$ mm
Mass of boom + load	$m = 410$ kg
Cylinder stroke	$H_c = 500$ mm
Cylinder piston diameter	$D_p = 65$ mm
Cylinder rod diameter	$D_r = 35$ mm
Cylinder area ratio	$\mu_C = \frac{D_p^2}{D_p^2 - D_r^2} = 1.41$
Supply pressure	$p_S = 180$ bar
Eigen frequency of mechanical hydraulic system (cylinder mid stroke)	$f_{mh} = 2.9$ Hz
Fixed orifice, pressure drop at a flow $Q = 2\frac{L}{\min}$	$\Delta p\|_{@2L/\min} = 220$ bar
PPRV, pressure drop at a flow $Q = 2\frac{L}{\min}$ and fully open	$\Delta p\|_{@2L/\min} = 6.1$ bar
CBV pilot ratio	$\alpha_p = 3$

Table 2. Compensator pressure range for secondary circuit connected to the CV and disconnected.

p_S [bar]	$p_{C,min}$, PPRV fully open [bar]	$p_{C,max}$, PPRV closed [bar]
180	3	75
180*	5*	180*

*disconnected from CV.

A real-time I/O system is used to control the hydraulic valves on the crane with a loop time of 10 ms. The control system can record sensor information from all the position and pressure sensors mounted on the crane.

The primary circuit is simply activated in a regular way by continuously supplying the DCV with an input signal.

The purpose of the controller on the secondary circuit is to keep the compensator pressure p_C, in accordance with the reference pressure $p_{C,ref}$ in Figure 5. The filter box uses the actual p_B value as input and returns $p_{C,ref}$. The choice of filter frequency should of course reflect both the dominant lowest eigen frequency of the mechanical hydraulic system as well as the demand for a certain response time of the system. The role of the low-pass filter is to remove oscillations, however if it is chosen overly conservative then the system reacts too slowly. Therefore, some logic has been added so that the compensator reference pressure, $p_{C,ref}$, never goes below a certain minimum value, p_{min}, see Equation (1).

$$p_{C,ref} = \begin{cases} p_{min} & p_B < p_{min} \\ p_{B,LPF} & p_B \geq p_{min} \end{cases} \tag{1}$$

where $p_{B,LPF}$ is the low-pass filtered value of p_B.

$$\dot{p}_{B,LPF} = \frac{1}{\tau}\left(p_B - p_{B,LPF}\right) \tag{2}$$

The PI-controller has the classical outlook

$$U = K_P \cdot \left(p_{C,ref} - p_C\right) + \int K_I\left(p_{C,ref} - p_C\right)\mathrm{d}t \tag{3}$$

where saturation and corresponding anti-windup measures (integrated effort not accumulated at saturation) are implemented so that $0 \leq U \leq 10$ V.

Basically, only four parameters need to be set: p_{min}, τ, K_P and K_I.

More advanced control strategies can be implemented if required from a performance point of view. As an example, the use of a feed-forward technique could have been investigated, however the scope of this paper is to present the novel concept rather than pursuing an optimal controller.

The experimental work showed some clear limitations in the chosen implementation of the secondary pressure circuit, which influences the achievable compensator pressure range listed in Table 2.

In the experimental work done in this paper the p_C value did not rise above 55 bar, hence, no attempt was made to increase $p_{C,max}$.

The performance obtained indicates that the circuit in reality looks like the one shown in Figure 7 and that it is somewhat component (CV) dependent.

Measurements revealed that the internal leakage of the load sensing system of the proportional valve group has the same order of magnitude as the fixed orifice.

An estimate of the internal leakage yielded a pressure drop at $Q = 2\frac{L}{\min}$ of $\Delta p|_{@}2L/\min = 160$ bar, i.e. slightly less resistance as compared to the fixed orifice, see Table 1. Internal leakage must be expected across the CV and other places in the load sensing system of the valve group, and this must be taken into account when choosing a PPRV and a fixed orifice. Basically, the fixed orifice shall not be too small relative to internal leakage since this will limit the maximum obtainable compensator pressure. Simultaneously the fixed orifice shall not be too large relative to the PPRV since this will limits the minimum obtainable compensator pressure. Therefore, the secondary circuit components shall be designed based on expected minimum and maximum steady state values of p_B and an estimate of the internal leakage of the valve group. Alternatively, a proportional pressure reducing valve could be connected to the high pressure line and in series with the internal leakage yielding a $p_{C,max}$ close to p_S.

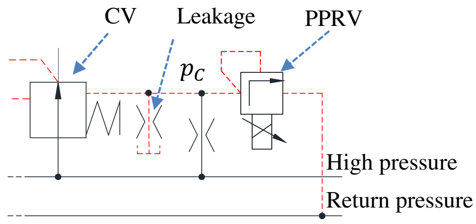

Figure 7. Actual secondary circuit including leakage in the load sensing system of the proportional valve group.

4. Linear stability analysis

A linearised stability analysis can help characterise the novel concept. In Figure 8 is shown a simplified circuit of the considered system including the core components and parameters.

The pressure drop across the metering-in orifice is

$$p_\Delta = p_C - p_B + p_{cl} \tag{4}$$

where p_{CL} is the set pressure of the compensator spring that relates to the fully closed position. The box, FB, is the first order low-pass filter that has been used:

$$p_C = \frac{1}{\tau \cdot s + 1} \cdot p_B \tag{5}$$

The extreme values of the time constant τ of the filter yields:

$$\tau = 0 \qquad p_\Delta = p_{cl} \tag{6}$$

$$\tau = \infty \qquad p_\Delta = p_S - p_{cl} - p_B - p_{C0} \tag{7}$$

Which reduces the system to either the base circuit, see Figure 1, without any filtering or a system with constant compensator pressure, $p_C = p_{C0}$.

The governing equations for the system are:

$$m_{eff} \cdot \dot{v}_C = p_B \cdot A_B - p_A \cdot \mu_C \cdot A_B + F_{cyl,static} \tag{8}$$

$$C_A \cdot \dot{p}_A = \mu_C \cdot A_B \cdot v_c - Q_{CBV} \tag{9}$$

$$C_B \cdot \dot{p}_B = Q_{DCV} - A_B \cdot v_c \tag{10}$$

$$\tau \cdot \dot{p}_C = p_B - p_C \tag{11}$$

$$Q_{DCV} = k_{v,DCV} \cdot u_{DCV} \cdot \sqrt{p_\Delta} \tag{12}$$

$$Q_{CBV} = k_{v,CBV} \cdot u_{CBV} \cdot \sqrt{p_A} \tag{13}$$

$$u_{CBV} = \frac{\alpha_p p_B + p_A - pcr}{\Delta p_{op,CBV}} \tag{14}$$

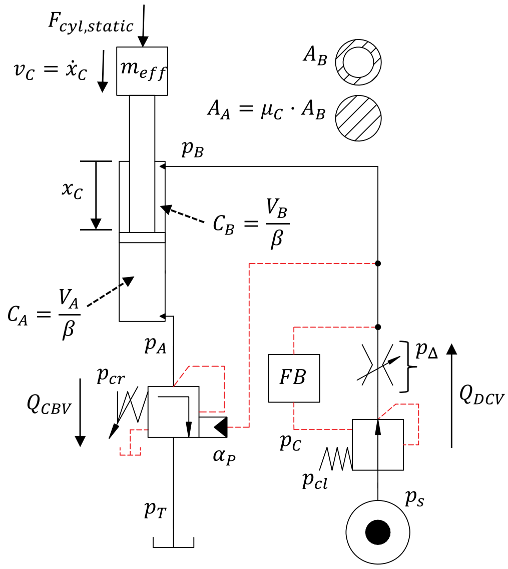

Figure 8. Simplified circuit.

In the above, a number of assumptions have been made including: no backpressure, no friction or damping in the cylinder and in the valves, no valve dynamics, no flow forces, and a linear discharge characteristic of the CBV. The value $\Delta p_{op,CBV}$ is the extra pressure required to fully open the CBV.

Linearising and denoting any parameter variation relative to the steady state solution with a tilde yields the following expressions in the Laplace domain:

$$m_{eff} \cdot s \cdot \tilde{v}_C = A_B \cdot \tilde{p}_B - \mu_C \cdot A_B \cdot \tilde{p}_A \tag{15}$$

$$C_A \cdot s \cdot \tilde{p}_A = \mu_C \cdot A_B \cdot \tilde{v}_C - \tilde{Q}_{CBV} \tag{16}$$

$$C_B \cdot s \cdot \tilde{p}_B = \tilde{Q}_{DCV} - A_B \cdot \tilde{v}_C \tag{17}$$

$$\tau \cdot s \cdot \tilde{p}_C = \tilde{p}_B - \tilde{p}_C \tag{18}$$

$$\tilde{Q}_{DCV} = k_{qu,DCV} \cdot \tilde{u}_{DCV} + k_{qp,DCV} \cdot \left(\tilde{p}_C - \tilde{p}_B\right) \tag{19}$$

$$\tilde{Q}_{CBV} = k_{qu,CBV} \cdot \tilde{u}_{CBV} + k_{qp,CBV} \cdot \tilde{p}_A \tag{20}$$

$$\tilde{u}_{CBV} = \frac{\alpha_p \tilde{p}_B + \tilde{p}_A}{\Delta p_{op,CBV}} \tag{21}$$

In Equations (19) and (20) the valve gains are:

$$k_{qu,DCV} = k_{v,DCV} \cdot \sqrt{p_\Delta^{(ss)}} \tag{22}$$

$$k_{\mathrm{qp,DCV}} = \frac{k_{\mathrm{v,DCV}} \cdot u_{\mathrm{DCV}}^{(\mathrm{ss})}}{2 \cdot \sqrt{p_{\Delta}^{(\mathrm{ss})}}} \quad (23)$$

$$k_{\mathrm{qu,CBV}} = k_{\mathrm{v,CBV}} \cdot \sqrt{p_{\mathrm{A}}^{(\mathrm{ss})}} \quad (24)$$

$$k_{\mathrm{qp,CBV}} = \frac{k_{\mathrm{v,CBV}} \cdot u_{\mathrm{CBV}}}{2 \cdot \sqrt{p_{\mathrm{A}}^{(\mathrm{ss})}}} \quad (25)$$

Specifically, Equation (19) is of interest since it reveals the influence of the filtering. With no filtering, $\tau = 0$, the DCV flow is:

$$\tilde{Q}_{\mathrm{DCV}} = k_{\mathrm{qu,DCV}} \cdot \tilde{u}_{\mathrm{DCV}} \quad (26)$$

which corresponds to the base circuit without any damping of the inlet flow. With maximum filtration, $\tau = \infty$, it becomes

$$\tilde{Q}_{\mathrm{DCV}} = k_{\mathrm{qu,DCV}} \cdot \tilde{u}_{\mathrm{DCV}} - k_{\mathrm{qp,DCV}} \cdot \tilde{p}_{\mathrm{B}} \quad (27)$$

where the negative term automatically introduces damping in the system. In practice, a finite time constant is present, which leads to the flow being described by Equation (19). It should be noted that when the steady state opening of the DCV becomes zero, $\tilde{u}_{\mathrm{DCV}} = 0$, then Equation (27) becomes (26). In practice, there will be some leakage in the zero position so that $k_{\mathrm{qp,DCV}}$ never becomes exactly zero as discussed by (Merritt 1967) on servo valves in neutral position, however, the novel concept will have less impact for small valve openings.

From Equations (15–21) the closed loop transfer function for valve opening $\tilde{u}_{\mathrm{DCV}} = 0$ as input and the cylinder displacement flow $A_{\mathrm{B}} \cdot \tilde{v}_{\mathrm{c}}$ as output can be derived. The transfer function is of the fourth order and the denominator is expressed by:

$$D(s) = D_4(\tau) \cdot s^4 + D_3(\tau) \cdot s^3 + D_2(\tau) \cdot s^2 + D_1(\tau) \cdot s + D_0 \quad (28)$$

where the coefficients are:

$$D_4 = C_{\mathrm{A}} \cdot C_{\mathrm{B}} \cdot m_{\mathrm{eff}} \cdot \tau \quad (29)$$

$$D_3 = \left(C_{\mathrm{A}} \cdot k_{\mathrm{qp,DCV}} + C_{\mathrm{B}} \cdot k_{\mathrm{qo,CBV}} + C_{\mathrm{B}} \cdot k_{\mathrm{qp,CBV}}\right) \cdot m_{\mathrm{eff}} \cdot \tau + C_{\mathrm{A}} \cdot C_{\mathrm{B}} \cdot m_{\mathrm{eff}} \quad (30)$$

$$D_2 = \left(A_{\mathrm{B}}^2 \cdot C_{\mathrm{A}} + A_{\mathrm{B}}^2 \cdot C_{\mathrm{B}} \cdot \mu_C^2 + k_{\mathrm{qo,CBV}} \cdot k_{\mathrm{qp,DCV}} \cdot m_{\mathrm{eff}} + k_{\mathrm{qp,CBV}} \cdot k_{\mathrm{qp,DCV}} \cdot m_{\mathrm{eff}}\right) \cdot \tau + (k_{\mathrm{qo,CBV}} + k_{\mathrm{qp,CBV}}) \cdot C_{\mathrm{B}} \cdot m_{\mathrm{eff}} \quad (31)$$

$$D_1 = \left(k_{\mathrm{qo,CBV}} + k_{\mathrm{qp,CBV}} + k_{\mathrm{qp,DCV}} \cdot \mu_{\mathrm{C}}^2 + \alpha_{\mathrm{p}} \cdot k_{\mathrm{qo,CBV}} \cdot \mu_{\mathrm{C}}\right) \cdot A_{\mathrm{B}}^2 \cdot \tau + (C_{\mathrm{B}} \cdot \mu_{\mathrm{C}}^2 + C_{\mathrm{A}}) \cdot A_{\mathrm{B}}^2 \quad (32)$$

$$D_0 = \left(\left(1 + \alpha_{\mathrm{p}} \cdot \mu_{\mathrm{C}}\right) \cdot k_{\mathrm{qo,CBV}} + k_{\mathrm{qp,CBV}}\right) \cdot A_{\mathrm{B}}^2 \quad (33)$$

In the above equations the term

$$k_{\mathrm{qo,CBV}} = \frac{k_{\mathrm{qu,CBV}}}{\Delta p_{\mathrm{op,CBV}}} = \frac{k_{\mathrm{v,CBV}} \cdot \sqrt{p_{\mathrm{A}}^{(ss)}}}{\Delta p_{\mathrm{op,CBV}}} \quad (34)$$

has been introduced in order to simplify the notation.

The Routh-Hurwitz stability criteria states that the system is stable if the following three inequalities are fulfilled:

$$D_i > 0, \qquad \text{for } i = 1 \ldots 4 \quad (35)$$

$$D_3 \cdot D_2 - D_4 \cdot D_1 > 0 \quad (36)$$

$$D_3 \cdot D_2 \cdot D_1 - D_4 \cdot D_1^2 - D_3^2 \cdot D_0 > 0 \quad (37)$$

Inequality in Equation (35) is always satisfied. The other two inequalities must be investigated numerically because the expressions are not easily interpreted. The cylinder position is chosen as almost fully extracted with $x_{\mathrm{C}} = 0.05$ m, or 90% extracted, avoiding singularities caused by zero capacitance in the B-volume. It is well known that a combination of the fully extracted position and small openings of the CBV is critical with respect to stability (Table 3).

Without maximum filtering, see Equation (27), the minimum steady state opening that is stable is $u_{\mathrm{DCV}}^{(\mathrm{ss})} = 0.033$. Without any filtering, see Equation (26), the system is unstable for any value of $u_{\mathrm{DCV}}^{(\mathrm{ss})}$. In Figure 9 the minimum time constant value that yields a stable system, τ_{min}, is plotted vs. $u_{\mathrm{DCV}}^{(\mathrm{ss})}$. It is clear that the correlation is strongly nonlinear allowing very small time constants until the critical steady state opening is approached.

The linear analysis reveals that the novel concept can improve stability. However, for very small openings of

Table 3. Parameters of linearised analysis.

Parameter	Value
External cylinder load	$F_{cyl,static} = 2.10 \cdot 10^4 N$
Effective mass	$m_{\text{eff}} = 1.50 \cdot 10^4 \text{kg}$
Capacitance of A volume	$C_A = 1.13 \cdot 10^{-4} \frac{L}{\text{bar}}$
Capacitance of B volume	$C_B = 2.45 \cdot 10^{-5} \frac{L}{\text{bar}}$
Directional control valve constant	$k_{v,DCV} = 9.34 \frac{L}{\text{min}\sqrt{\text{bar}}}$
Counterbalance valve constant	$k_{v,CBV} = 15 \frac{L}{\text{min}\sqrt{bar}}$
Counterbalance valve fully open pressure difference	$\Delta p_{op,CBV} = 250$ bar

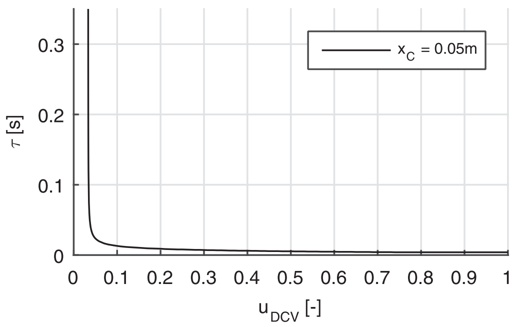

Figure 9. Stability of the linear model. The minimum time constant value that yields a stable system is plotted vs. the dimensionless opening of the directional control valve u_{DCV}.

the DCV and, subsequently, small openings of the CBV the novel concept will become increasingly similar in behaviour to the base circuit that is inherently unstable. The linear analysis has some limitations at the small openings. Leakage in the DCV as well as damping in the compensator and CBV will greatly improve the ability of the novel concept to suppress oscillations even at small flows.

5. Experimental investigation

The investigation is divided in two parts: one for the secondary circuit alone and one for the total system.

5.1. Secondary circuit

In order to be able to evaluate the performance of the novel concept the secondary circuit is analysed first. To do so a reference step input, $p_{C,ref}$, of 40 bar is applied to the secondary circuit alone. With $p_{min} = 5$ bar, the gains were adjusted to: $K_P = 0.01 \frac{V}{\text{bar}}$ and $K_I = 0.04 \frac{V}{\text{bar} \cdot \text{s}}$. The results are shown in Figure 10.

The blue curve shows the reference input. The ability of the closed-loop control system to follow this reference is shown in red. It takes approximately 0.3 s to reach the wanted reference. This curve is obtained without any filtration, i.e. $\tau = 0$ s, The remaining curves, the cyan and magenta ones demonstrate the effect of the low-pass filter in the system, where the curves show the filtered

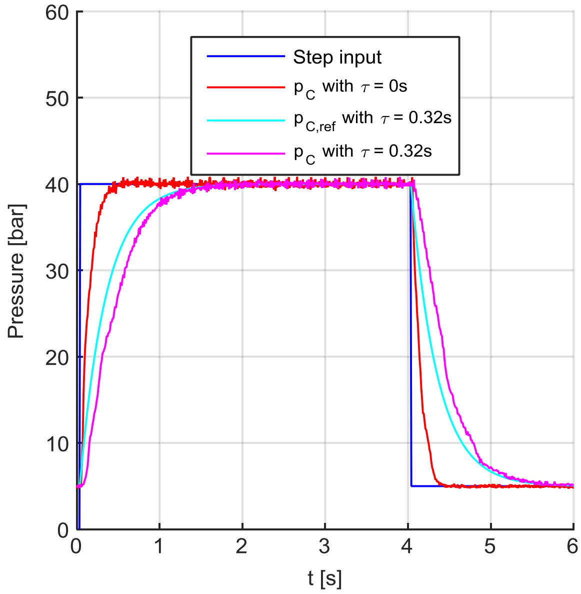

Figure 10. Pressure performance of secondary circuit.

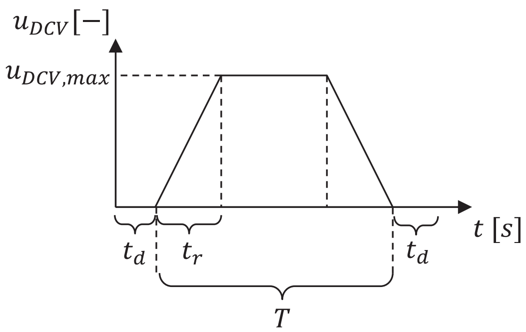

Figure 11. Work cycle – actuation function.

Table 4. Common parameters for all actuation.

Cycle time, T	Ramp time, t_r	Time delay, t_d
8 s	1 s	1 s

reference $p_{C,ref}$ and how p_C follows it. The cut-off frequency is set to approximately 20% of the mechanical hydraulic eigen frequency, i.e. $\tau = 0.32$ s.

5.2. Total system

To achieve a uniform evaluation of the total system a standard actuation of the DCV is used, see Figure 11. Only situations where the cylinder is retracting are investigated.

The actuation is defined by the cycle time, T, a delay time to ensure static conditions, t_d, the ramp time, t_r, and the wanted steady state DCV input $u_{DCV,max}$. The time parameters are similar for all tests, see Table 4.

5.2.1. Demonstration of novel concept

The effect of the novel concept is shown in Figure 12 where the system is subjected to an actuation of

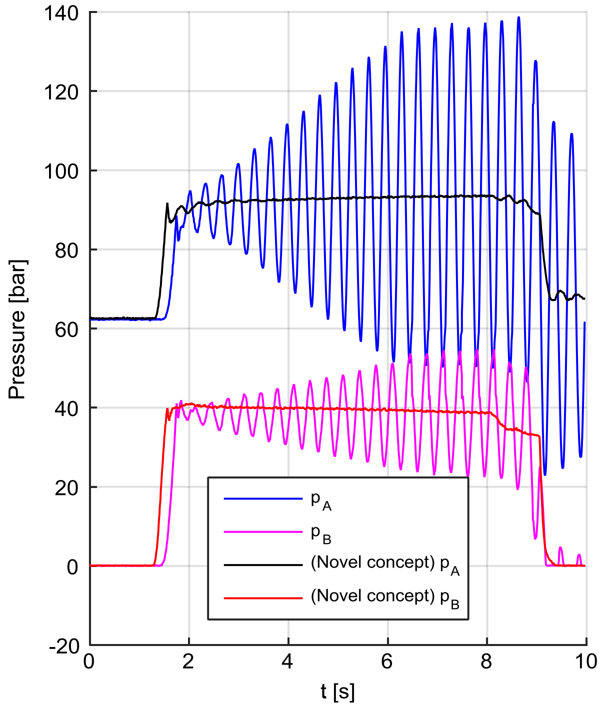

Figure 12. Comparison of pressures between the base circuit and the system with the novel concept implemented ($u_{DCV,max} = 0.15$).

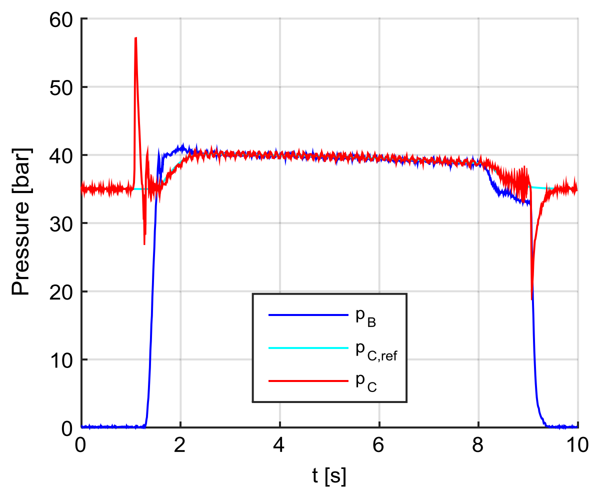

Figure 13. Pressure response of the novel concept's secondary circuit during work cycle ($u_{DCV,max} = 0.15$).

$u_{DCV,max}$ = 0.15. The controller was implemented with p_{min} = 35 bar, and the same gains as in Section 5.1.

The base circuit should be unstable and the novel concept stable according to the linear analysis which is also what can be observed. Further, the ability of the novel concept to suppress oscillations in a real system is demonstrated. The corresponding behaviour of the secondary circuit is shown in Figure 13.

It is clear, that as the DCV is actuated at $t = t_d$ the secondary circuit is subjected to disturbances. The disturbances are probably caused by the motion of the main spool and the related variations in leakage paths as well as the pumping effect of the CV spool. As an example,

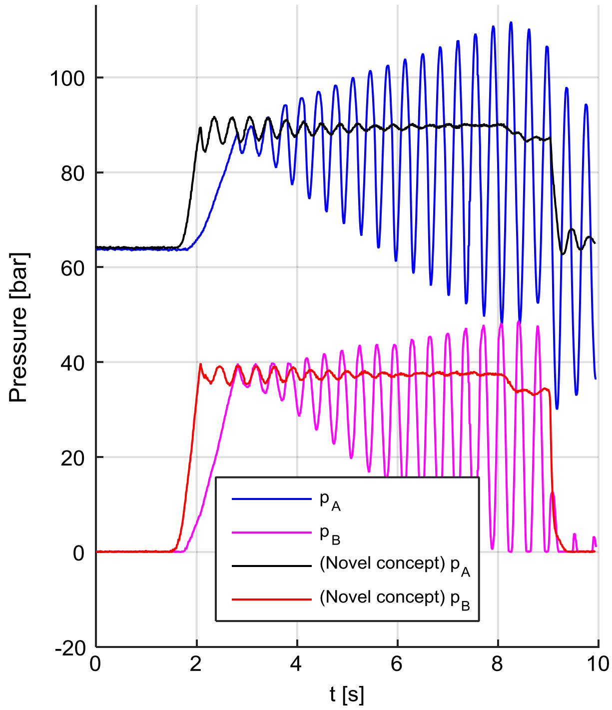

Figure 14. Comparison of pressures between the base circuit and the system with the novel concept implemented ($u_{DCV,max} = 0.05$).

the increase in p_C to approximately 55 bar at t = 1.2 s will remain if the PPRV is not actively controlled. Investigations show that the disturbances are DCV related and should be expected when using a standard valve for this system. However, the PI-controller of the secondary circuit manages these disturbances before the DCV reaches its deadband and the motion of the cylinder is not affected.

5.2.2. Stability for small DCV openings

The stability analysis of the linear model predicted instability for small openings of the DCV. To verify this the work cycle was executed with $u_{DCV,max}$ = 0.05. This was done for the base circuit and the novel concept, see Figure 14.

Experiments show that in reality the novel concept seems able to suppress oscillations for small valve openings as well although a certain variation is observed in both pressures. Obviously, the inherent damping of the system as well as the fact that the cylinder is stroking in and thereby moving towards a more stable position means that the system oscillations are suppressed.

5.2.3. Handling different DCV openings

An important feature of the novel concept is that it shall be able to maintain the load independent flow capability of the DCV. Experiments using the work cycle function for eight different values of $u_{DCV,max}$ for both the base circuit and the novel concept were conducted and the steady state cylinder velocities can be compared.

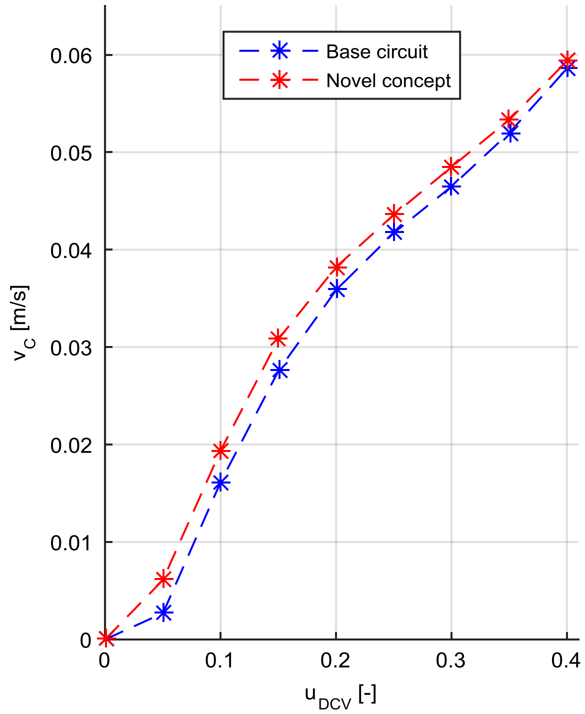

Figure 15. Steady state cylinder velocities for different valve openings for both the base circuit and the novel concept.

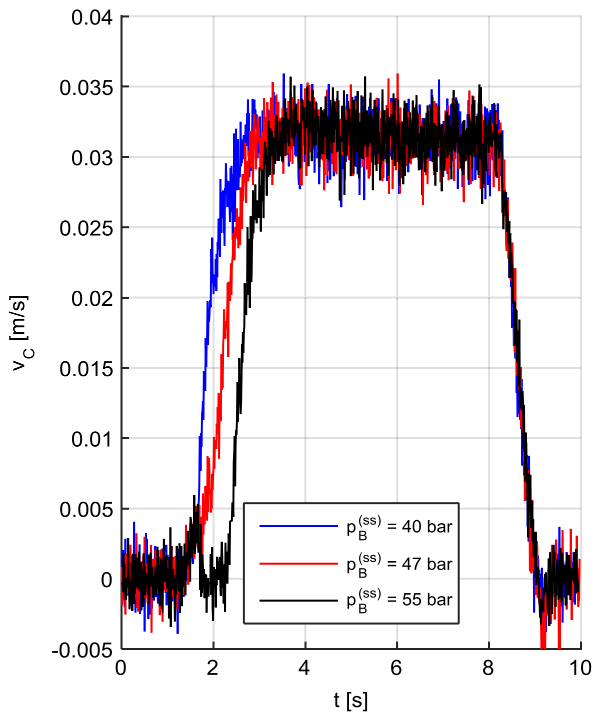

Figure 16. Comparison of cylinder velocities – different pressure levels.

The results of these experiments are shown in Figure 15. The shape of the curves for the two systems are similar indicating that the flow control is intact. It can be seen that the novel concept has a slightly higher flow rate. A possible explanation is that the pressure experienced by the CV in the base circuit, due to small orifice effects in the LS-paths, is actually slightly smaller than the p_B value. Unlike for the novel concept where such a reduction in pilot pressure is automatically compensated.

5.2.4. *Handling load variations*

This section continues with experiments demonstrating the ability of the novel concept to handle load variations. These experiments were conducted using the work cycle with $u_{DCV,max} = 0.15$. Experiments for three different load cases were chosen with required p_B-pressures of 40, 47 and 55 bar corresponding to different loads mounted on the boom.

Figure 16 shows that the steady state speed is load independent when using the novel concept, however, the response time is increased when the difference between the standby pressure and the steady state pressure level is increased. This is caused by the low-pass filter and the effect can probably be reduced by improving the closed loop control of the secondary circuit.

6. Conclusions

A novel concept for suppressing oscillations in hydraulic systems containing a CBV and a pressure compensated flow supply is presented. The concept utilizes a secondary circuit where a low-pass filtered value of the load pressure is generated and fed back to the compensator of the flow supply valve.

Stability of the novel concept has been investigated on a linearised model and has been experimentally verified. The results show a significant reduction of the pressure oscillations when lowering the boom on an otherwise unstable system.

The stability analysis indicates that for DCV openings below 3.3% instability will still be an issue. However, this is not observed experimentally which can be explained by a lack of damping in the linear model.

Experiments comparing the steady state cylinder velocity for different DCV openings and different load pressure levels also show that the load independent flow of the base circuit is kept when using the novel concept. From the operator point of view, the behaviour is left unaltered and hence eliminating the need for additional velocity control of the actuator.

6.1. *Future work*

In the present paper some issues have not been investigated deeply and require more work. Amongst other, this includes the following:

- Deeper analysis of the relation between component properties and system performance utilising the novel concept.
- Further investigation of the consequences of load variations on the boom.

- Investigation of system performance using different pilot ratios of the CBV.
- Try different filtering algorithms in the filter box in order to reduce the response time of the system.

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Notes on contributors

Jesper K. Sørensen graduated in 2010 from Aalborg University with a MSc in Electric Mechanical System Design. He worked two years in the Wind Turbine industry, before beginning as a PhD student in the Mechatronics group at University of Agder, Norway, in 2012. The topic of his research is boom movement in hydraulic cranes, with main focus on the hydraulic actuation.

Michael R. Hansen received his MSc in mechanical engineering from Aalborg University in Denmark in 1989 and his PhD in computer-aided design of mechanical mechanisms from the same institution in 1992. He is currently holding a position as a professor in fluid power in the mechatronics group at the Department of Engineering Sciences at the University of Agder in Norway. His research interests mainly include fluid power, multi-body dynamics and design optimisation.

Morten K. Ebbesen is affiliated with the Department of Engineering Sciences, University of Agder, Norway, as an associate professor in the Mechatronics group. He received his MSc (2003) and PhD (2010) in mechanical engineering from the University of Aalborg, Denmark. His interests are in the field of dynamics, flexible multi-body systems, time domain simulation, hydraulics and optimisation.

ORCID

Morten K. Ebbesen http://orcid.org/0000-0002-6983-8720

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