LOAD HOLDING VALVES WITH INTEGRATED FLOW SENSORS

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ABSTRACT

Counterbalance valves are widely used in mobile hydraulics, they preload the return line of motors and cylinders to ensure the load doesn't move when the proportional control valve is in open center position. They also preload the return line when the cylinder moves to lower a load. In that condition the circuit can be unstable and restrictive counterbalance valves are needed. They stabilize the circuit but require higher inlet pressures. The paper describes a way to stabilize the circuit without adding pressure losses. A flow sensor can be built into the counterbalance. The signal can be used for a closed loop control. A simulation shows that the controller can improve the stability.

An alternative to the flow sensor in the counterbalance valve is a velocity sensor on the cylinder. Again, a controller can use the signal: it closes the loop on the velocity of the cylinder. The paper compares the closed loop controls.

Keywords: Counterbalance, Flowmeter, Stability, Load lowering

1. FUNCTION OF COUNTERBALANCE VALVES

Counterbalance valves are hydromechanical poppet type valves. A common design is a screw-in cartridge with 3 ports. See fig. 1.

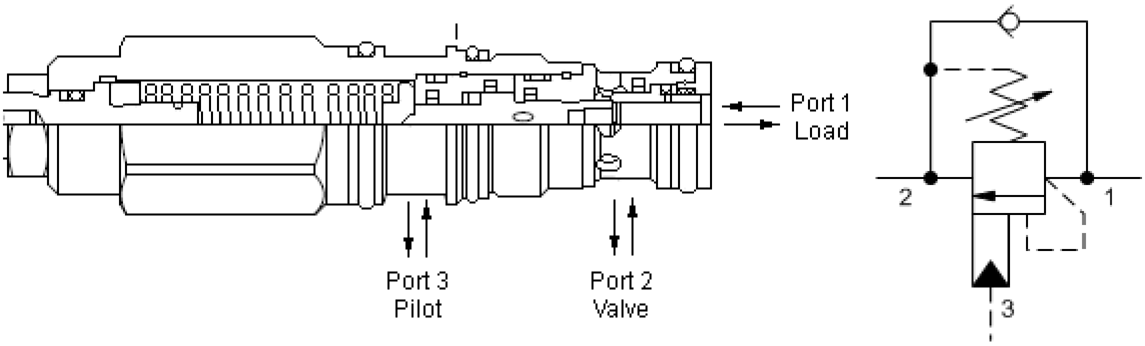


Figure 1: Counterbalance valve as screw-in cartridge

The counterbalance valve acts like a relief valve between port 1 and 2 but has an additional third port. Pilot pressure on port 3 helps to open the valve. The valve is usually built into the hydraulic line downstream of a cylinder that supports an external force if that force pushes or pulls the in the direction of the intended movement (negative load). Fig.2 on the left (A) shows a typical circuit: The cylinder doesn't move if the proportional valve is in the centre position and the counterbalance is set high enough. A recommended setting is 30% above the highest expected load induced pressure F/A

$$\mathrm{set} = 1.3 * \mathrm{F/A} \tag{1}$$

The setting is the pressure on port 1 that starts to open the valve against an adjustable spring. Since a pilot pressure on port 3 also helps to open the valve, it is important to set the valve with no pilot pressure on port 3. When the proportional valve in the circuit opens, the pressure p_A increases. It works on port 3 of the counterbalance valve. It also works on the cylinder and increases the pressure p_B between cylinder and counterbalance valve. The pressures p_A and p_B that open the counterbalance valve can be calculated from two equations: The force balance across the cylinder:

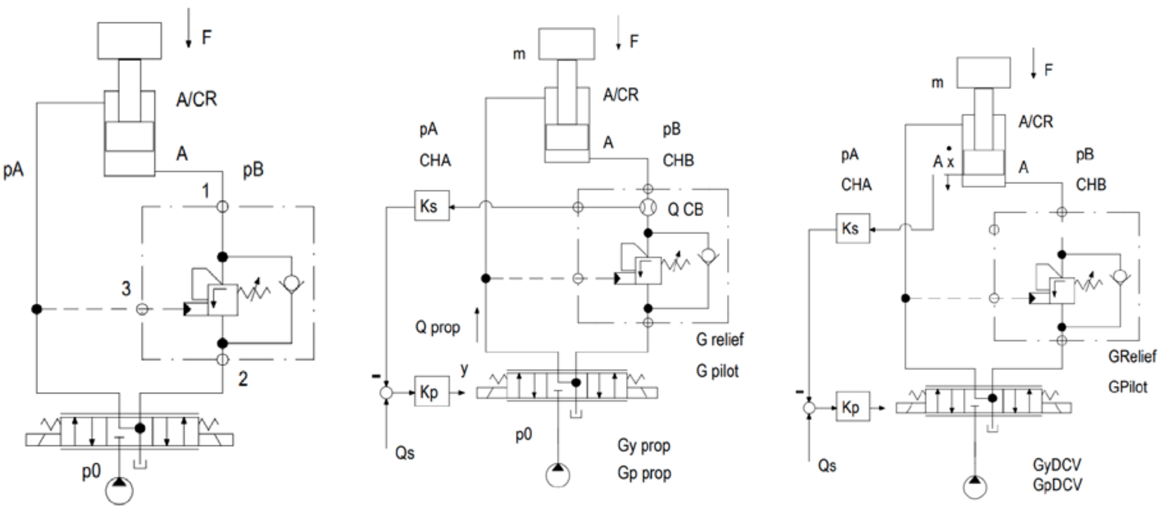
$$p_B = \frac{F}{A} + p_A/\mathrm{CR} \tag{2}$$

and the force balance for the counterbalance valve when it starts to open:

$$\mathrm{set} = p_B + p_A * PR \tag{3}$$

From the equations (2) and (3) the pressure p_A at that the movement starts can be calculated:

$$p_A = \frac{set - F/A}{\frac{1}{CR} + PR} \tag{4}$$



A: Standard circuit B: Circuit with flow feedback C: Circuit with cylinder speed feedback

Figure 2: Circuits with load reactive counterbalance valves

The pressure $p_A$in equation (4) must be positive to avoid cavitation. A low pressure is desirable because the inlet pressure p_A determines the required pressure to move the cylinder.
The difficulty in selecting a counterbalance valve is in finding the one that is stable (low pilot ratio, restrictive) but also doesn’t generate excessive pressure losses.

This paper shows a counterbalance valve with a built-in flowmeter. The goal is using the flow signal in a simple closed loop (fig. 2B) to improve the performance and achieve stability with a low inlet pressure. The paper compares the stability of that circuit to a circuit with a feedback of the cylinder speed (fig. 2 C).

2. COUNTERBALANCE VALVE WITH INTEGRATED TURBINE AS FLOW SENSOR

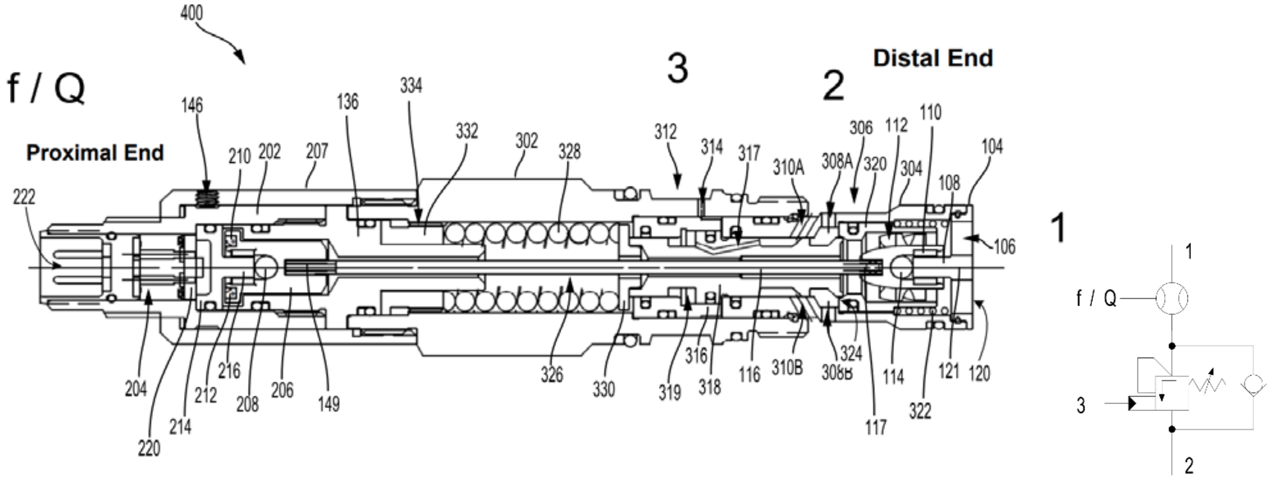


Figure 3: Cross section of an adjustable counterbalance valve with integrated turbine as flowmeter (Pat. Pending)

Fig. 3 shows a cross section of a counterbalance with the usual porting and function. The load is on port 1. Pilot pressure on port 3 moves the poppet (317) against a spring to open a flow path from 1 to 2. A turbine (112) near port 1 starts to spin when the valve opens. A long shaft (326) connects the turbine (320) to a wheel (206) with permanent magnets (219). A hall sensor on a circuit board (220) detects the speed of the wheel that is proportional to the flow. A digital signal is available through an M12 connector (222) at the proximal end of the counterbalance valve.

The valve has not been built as shown. But a very similar turbine in a different valve showed the correlation between the measured rpm of the turbine and the flow measured through a fixed displacement gear flow meter. The turbine reacts within about 10 milliseconds to changing flows.

Customers are interested in flowmeters if they can replace expensive sensors on long telescopic cylinders, but they are skeptical about the benefits of the integrated flow sensor with respect to stability. A common practice is using encoders to measure the position of work platforms. The signal is required for the correct positioning of the platform, but the circuit is still known for low damping. So, the question came up, whether measuring the flow instead of the movement of the cylinder could be more useful to improve the performance.

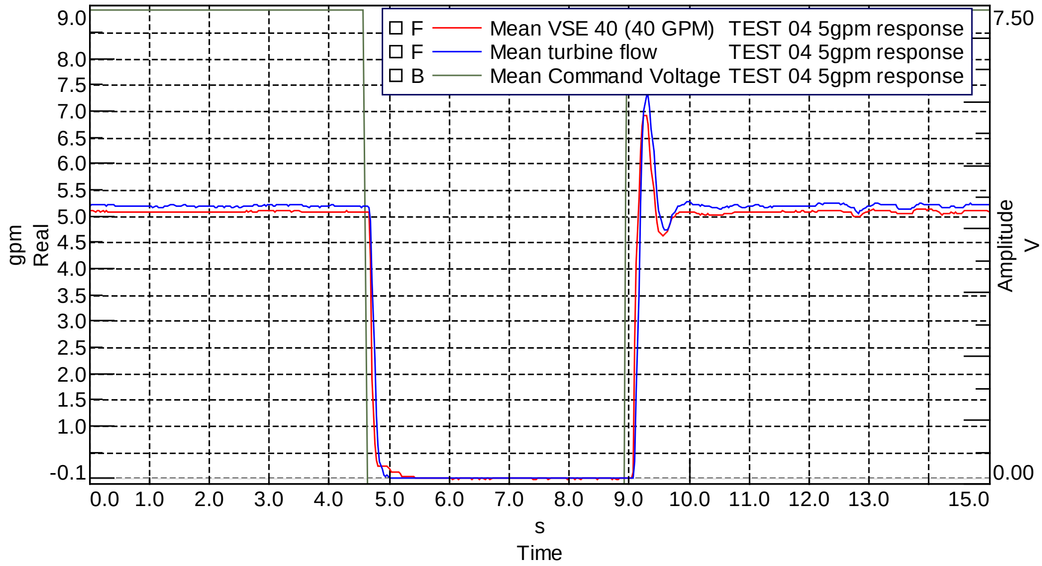


Figure 4: Flow measured with a gear flow meter (red) and with a turbine (blue), built into a cartridge.

3. LINEARISED MODEL OF A LOAD LOWERING CIRCUIT WITH COUNTERBALANCE VALVES AND FLOW SIGNAL FEEDBACK

In a previous paper [1] a mathematical, linear model was presented for a circuit with counterbalance valves. The paper also showed counterbalance valves with adaptive pilot ratios that have a low pilot ratio only in operating points where stability is critical. The overall efficiency of circuits with adaptive pilot ratios is higher. The adaptive counterbalance valves have been introduced into the market.

This paper describes another approach to improve load lowering circuits: The goal is finding out whether the signal from a flowmeter near the or built into the counterbalance could be used to improve the stability of the circuit. The goal is again to reduce the inlet pressure $p_A$when the load is being lowered and maintain stability. As in the previous paper, a state-space representation is chosen to calculate the stability of the circuit.

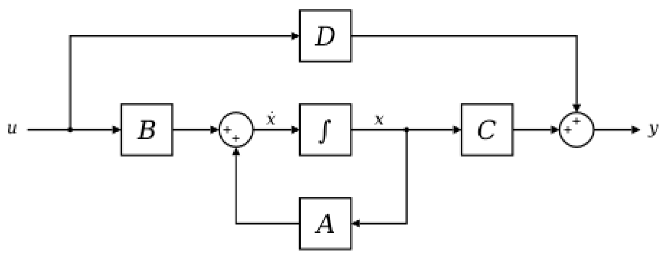


Figure 5: State-space representation

Figure 5 shows the general format: ‘A’ is the system matrix, ‘B’ the input matrix and ‘C’ the output matrix. We want to model the hydraulic system shown in fig. 2 B. The feedthrough matrix D is zero. Figure 2B describes the same circuit as in the previous paper [1] except for a sensor for the flow across the counterbalance valve. So, we can reuse the model for all components except the additional sensor. Please refer to that paper for the complete modelling of the circuit. The block diagram in figure 6 shows the circuit including the new sensor and the controller. The linearized flow Q_{CB} through the counterbalance valve is a function of pilot pressure p_A and load pressure p_B :

$$Q_{CB} = G_{pilot} * p_A + G_{relief} * p_B \tag{5}$$

The assumption is that a sensor measures that flow without delay. The block diagram shows the sensor with a factor K_S . Changing that value from 0 to 1 activates the sensor. The parameter K_P describes the controller for the proportional valve. The simple proportional controller opens the proportional valve without delay. The variable y is the stroke the proportional valve:

$$\mathrm{y} = K_p * (Q_s - Q) \tag{6}$$

The linearized flow across the proportional valve is a function of stroke y and the pressure differential $p_O - p_A$:

$$Q_{Prop} = Gy_{prop} * y + Gp_{prop}(p_0 - p_A) \tag{7}$$

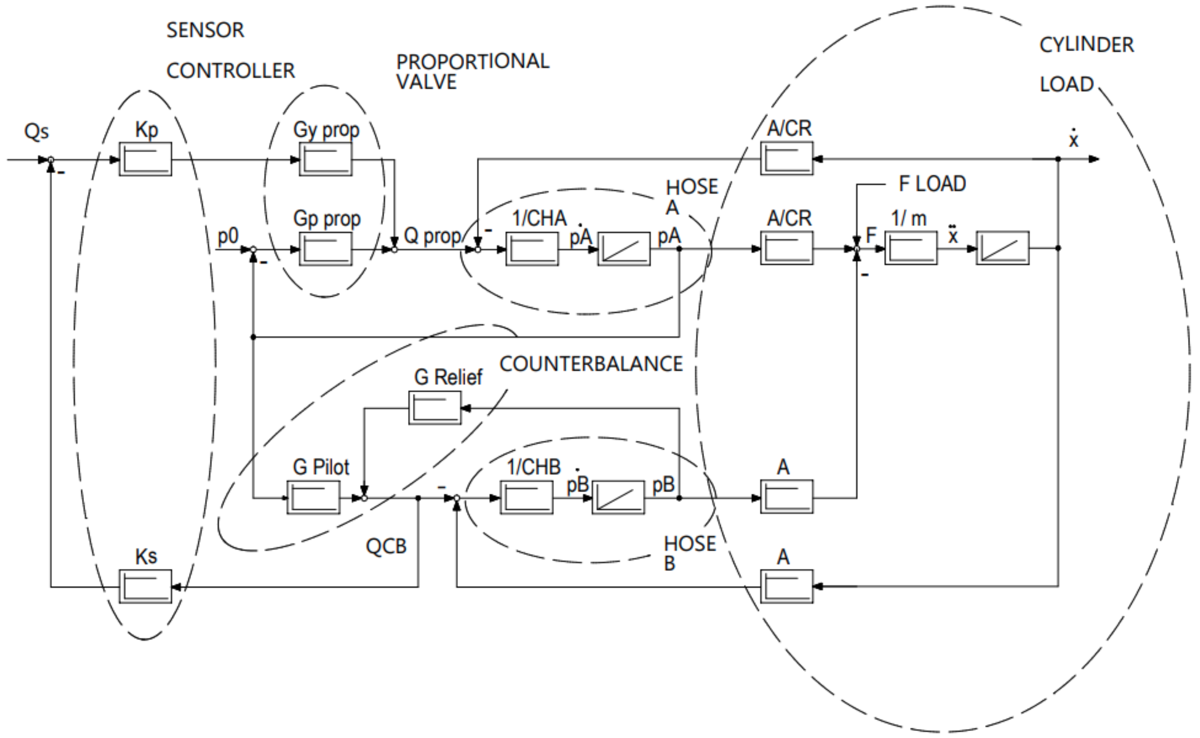


Figure 6: Block-diagram for the circuit in figure 2B.

Energy is stored in three components of the circuit: potential energy is stored in the capacitances CH_A, CH_B of the hoses and the attached volumes of the cylinder with their pressures p_A and p_B , kinetic energy is stored in the masses of the cylinder and the attached load with their velocity $\dot{x}$. So, the state variables of the vector x in figure 5 are p_A , p_B ,and $\dot{x}$. As input variable u in figure 5 we choose the desired flow Q_s. Other input variables could be p_0, and F_{Load} with their input matrices B.

$$\begin{pmatrix} \dot{p}_A \\ \dot{p}_B \\ \ddot{x} \end{pmatrix} = A * \begin{pmatrix} p_A \\ p_B \\ \dot{x} \end{pmatrix} + \begin{pmatrix} \frac{Gy_{prop}*Kp}{C_{HA}} \\ 0 \\ 0 \end{pmatrix} * Q_s + \begin{pmatrix} \frac{Gp_{prop}}{C_{HA}} \\ 0 \\ 0 \end{pmatrix} * p_0 + \begin{pmatrix} 0 \\ 0 \\ \frac{1}{m} \end{pmatrix} * F_{Load} \tag{8}$$

In equation (8) A is the system matrix:

$$A = \begin{pmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ a_{31} & a_{32} & a_{33} \end{pmatrix} =$$

$$\begin{pmatrix} -\frac{Gp_{prop} + Kp * K_s * G_{pilot} * Gy_{prop}}{C_{HA}} & \frac{Kp * K_s * G_{Relief} * Gy_{prop}}{C_{HA}} & -\frac{A}{CR * C_{HA}} \\ -\frac{G_{Pilot}}{C_{HB}} & -\frac{G_{Relief}}{C_{HB}} & \frac{A}{C_{HB}} \\ \frac{A}{CR * m} & -\frac{A}{m} & 0 \end{pmatrix} \tag{9}$$

In this paper we assume that the variables Q_s , p_0, and F_{Load} are constant and independent of each other. That is why they are called input variables and not state variables. So, the paper does not describe instabilities of systems with load sensing systems for instance where the supply pressure p_0 is a function of the load pressure. Also, external forces on the cylinder F_{Load} that change with velocity -cylinder friction for instance- or stroke have an influence on stability but are not considered in this model.

4. STABILTY CRITERION OF THE LOAD LOWERING CIRCUIT WITH FLOW SIGNAL FEEDBACK

Whether the hydraulic system is stable or not depends solely on the system matrix A. The goal is to find out how the feedback of the flow signal and the controller influence the stability. The values of the elements a_{11}, a_{22} , a_{33} on the main diagonal have a positive influence on stability. So, K_p and K_s in the the a_{11} element can contribute to stability in combination with positive values for G_{pilot} and Gy_{prop}. G_{pilot} describes the change in flow across the counterbalance valve per change in pilot pressure, Gy_{prop} describes the change in flow across the proportional valve per stroke. Both gains are usually positive. So, the parameters K_p and K_s in the a_{11} element can have a similar influence as the parameter Gp_{prop}. That gain is the change in flow per change of the pressure differential across the valve.

$$Gp_{prop} = {dQ_{prop}}\Big/{d(p_0 - p_{A)}} \tag{10}$$

An infinitely high Gp_{prop}describes a reducing valve that keeps the pressure constant. That is obviously good for stability since the counterbalance valve would see a constant pilot pressure. The parameters in the elements of the main diagonal have a positive influence on stability if they appear only on the main diagonal. But the controller and sensor are also seen in the a_{12} element of the system matrix. Therefore, the Routh criterion has been used to calculate whether the system is stable. It uses the coefficients of the characteristic equation.

$$a_4 * s^3 + a_3 * s^2 + a_2 * \mathrm{s} + a_0 = 0 \tag{11}$$

That equation can be calculated from the system matrix:

$$\det(A - s * I) = 0 \tag{12}$$

The Routh criterion leads to three inequations. The system is stable if:

$$\begin{aligned} &\frac{A^2 * \left(C_{HA} * C_{HB} * G_{relief} * Gy_{prop} * CR + C_{HB}^2 * G_{pilot} * Gy_{prop}\right) * (K_P * K_S)^2 * G_{Pilot} * Gy_{prop}}{NOM} \\ &+ \frac{A^2 * C_{HA}^2 * G_{relief} * CR^2 * G_{pilot} * Gy_{prop} * K_P * K_S}{NOM} \\ &+ \frac{C_{HA} * C_{HB} * G_{Pilot} * CR}{NOM} \\ &+ \frac{C_{HB}^2 * Gp_{prop} * G_{pilot} * Gy_{pilot} * K_P * K_S}{NOM} + \frac{G_{relief} * Gp_{prop}}{C_{HA} * C_{HB}} > 0 \end{aligned} \tag{13a}$$

With

$$\begin{aligned} NOM = {} & C_{HA} * C_{HB} * CR^2 * m \\ & * \left(C_{HB} * G_{Pilot} * Gy_{prop} * K_P * K_S + C_{HB} * Gp_{prop} + C_{HA} * G_{relief}\right) \end{aligned} \tag{13b}$$

And

$$\frac{G_{Pilot} * Gy_{prop} * K_s * K_P}{C_{HA}} + \frac{Gp_{prop}}{C_{HA}} + \frac{G_{relief}}{C_{HB}} > 0 \tag{14}$$

And

$$\left(G_{relief} - G_{pilot} * CR\right) * \left(1 - K_P * K_S * Gy_{prop} * CR\right) + Gp_{prop} * CR^2 > 0 \tag{15}$$

The equations 13-15 are identical with equations 6-10 in the previous paper [1] for $K_S = 0$. Equations 13 and 14 show that the additional controller has no negative effect on stability. The factor $K_P * K_S$ is additive to parameters that need to be positive to achieve stability. But all three inequations 13-15 need to be fulfilled for the system to be stable. Equation 15 shows:

- The term $\left(G_{relief} - G_{pilot} * CR \right)$ needs to be positive. It means there is an upper limit for G_{pilot}, which describes the change in flow across the counterbalance valve per change in pilot pressure. Counterbalance valves with a with a high pilot ratio tend to be instable in load lowering circuits. Valves with lower pilot ratio and more restrictive valves are more stable but more power is wasted in the circuit to lower a load.

- The term $(1 - K_P * K_S * Gy_{prop} * CR)$ needs to be positive. It means there is an upper limit for the factor K_P of the controller. We can't conclude from equations 12-13 that the flow feedback improves stability under all circumstances.

- The term $Gp_{prop} * CR^2$ needs to be positive. Gp_{prop} is the change in flow across the proportional valve per change in pressure differential. The parameter can describe a reducing valve with a high change in flow and a pressure compensated flow control with a low change in flow. The disadvantage of using a reducing valve in the load lowering circuit is that the velocity of the cylinder is difficult to predict. Sensing the flow and closing the loop allows the application of reducing valves or proportional valves with a low flow gain Gp_{prop}. The flow gain of proportional valves is low at a low pressure-differential across the proportional valve -see [1]. So, less restrictive counterbalance valves can be applied if stability is achieved with a lower flow gain of the proportional valve: that lowers the inlet pressure and saves energy.

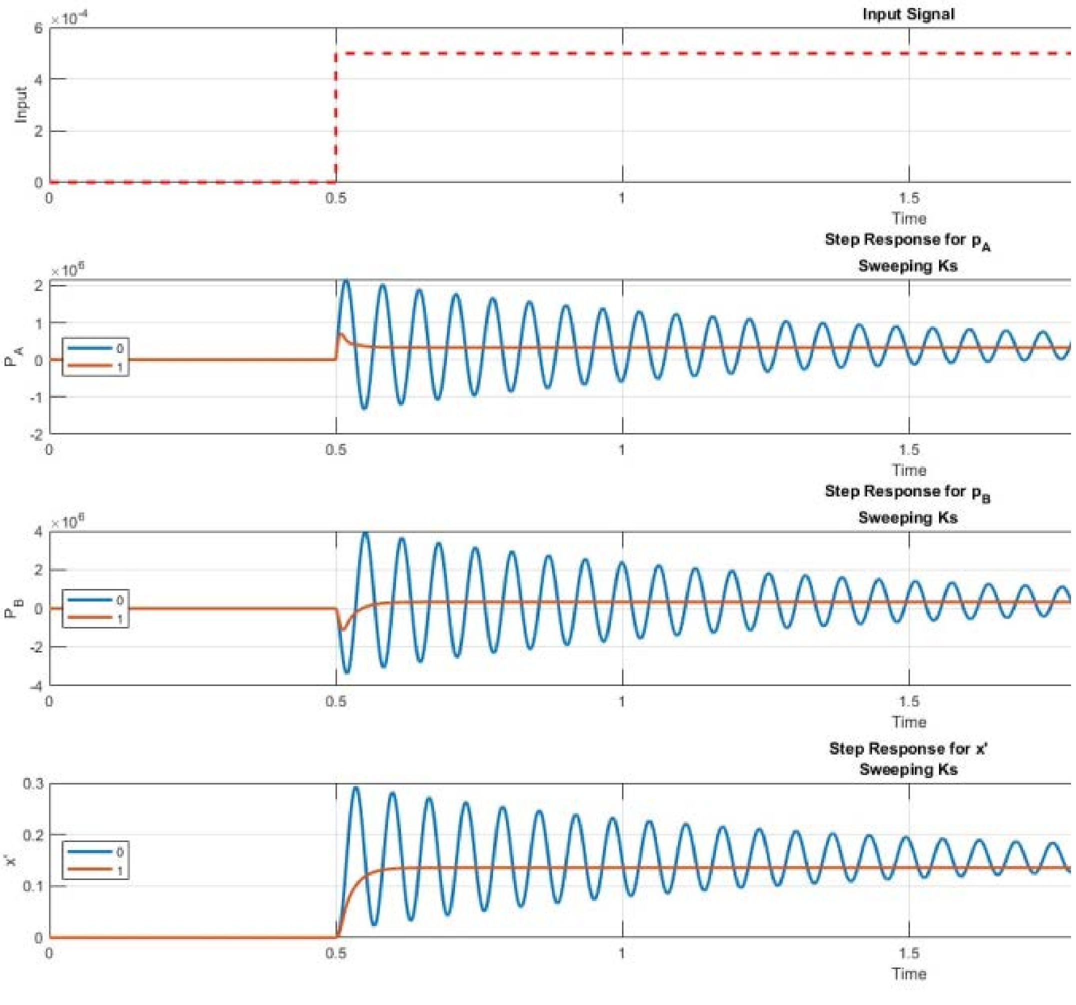


Figure 7: Step response for circuit - see fig. 3 -with ($K_S = 1$) and without ($K_S = 0$) flow feedback. $A = .001256\, m^2$, $\boldsymbol{C_{HA} = 1e^{-12}\, m^3/Pa}$, $C_{HB} = 7.59e^{-13}\, m^3/s * Pa$, $CR = 1$, $Gp_{prop} = 1.33\, e^{-11}\, m^3/(s * Pa)$, $G_{pilot} = 3.5\, e^{-10}\, m^3/(s * Pa)$, $G_{relief} = 1.65\, e^{-10}\, m^3/(s * Pa)$, $Gy_{prop} = .133\ m^3/(s * m)$, $K_p = 3\ m * s/(m^3)$, $m = 500\, kg$

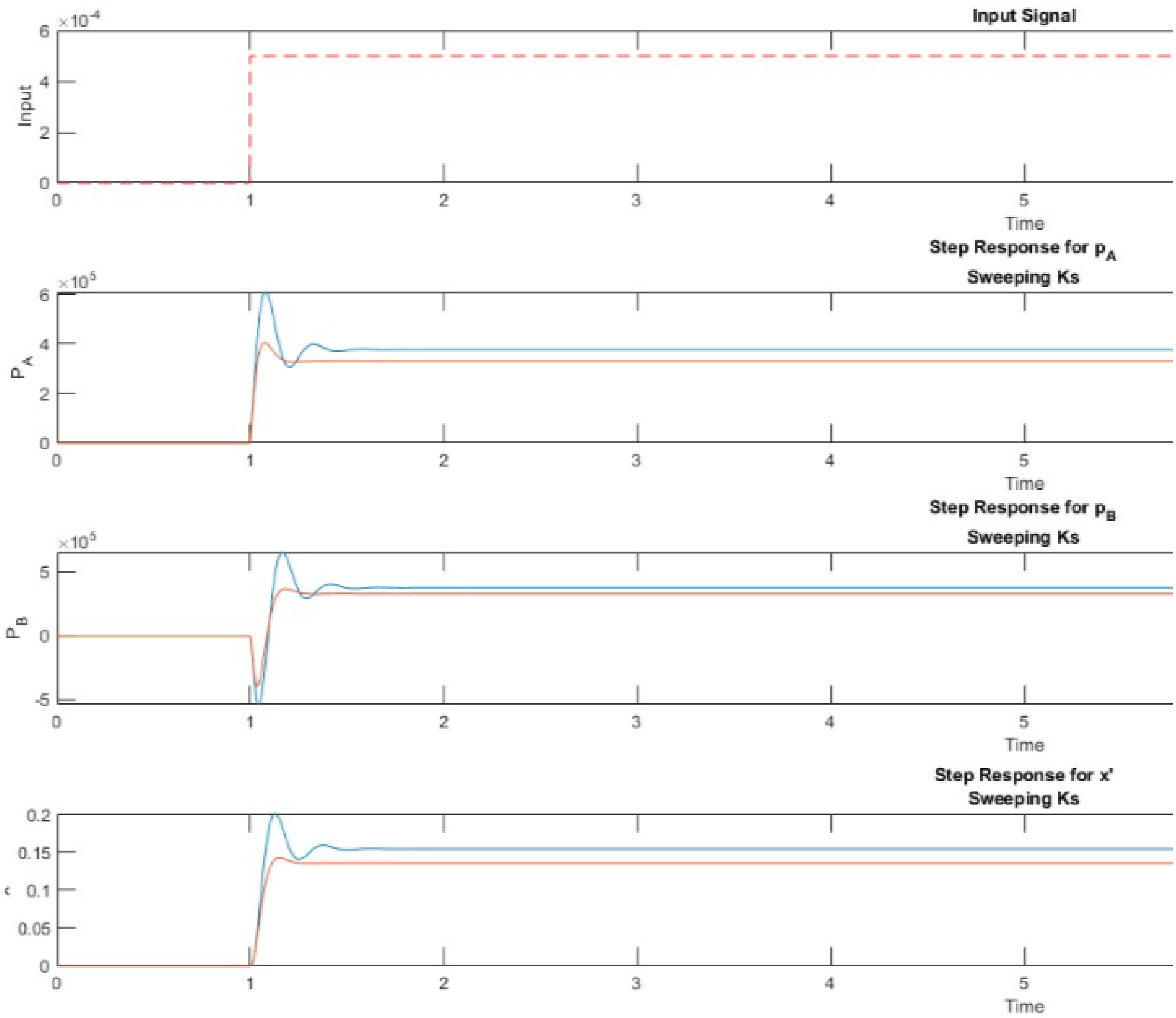


Figure 8: Step response for circuit - see fig. 3 -with ($K_S = 1$) and without ($K_S = 0$) flow feedback. $A = .001256\, m^2$, $\boldsymbol{C_{HA} = 1.5e^{-11}\, m^3/Pa}$, $C_{HB} = 7.59e^{-13}\, m^3/(s * Pa)$, $CR = 1$, $Gp_{prop} = 1.33\, e^{-11}\, m^3/(s * Pa)$, $G_{pilot} = 3.5\, e^{-10}\, m^3/(s * Pa)$, $G_{relief} = 1.65\, e^{-10}\, m^3/(s * Pa)$, $Gy_{prop} = .133\ m^3/(s * m)$, $K_p = 3\ m * s/m^3$, $m = 500\, kg$

Figure 7 shows how the step response to a changing flow command Q_s changes when the feedback signal is introduced. For the blue curves K_S is 0, so, the sensor is not active. For the orange curve K_S is 1. From top to bottom we see the input signal Q_s , pressures p_A, p_B, and the velocity $\dot{x}$ of the cylinder. The parameters are taken form a simulation in a program that has been presented in [1]. The parameters in SI units describe a hydraulic circuit with realistic numbers (cylinder diameter 28 mm, pilot ratio of the counterbalance valve $G_{pilot}/G_{relief} = 2 \ldots$) The intention is not to simulate a real step response with best possible accuracy but to show the influence of the flow feedback under otherwise unchanged conditions.
In the example the damping of the system improves a lot with the additional flow feedback. That was not to be expected: closing a loop usually improves the accuracy, but often it comes at a price: stability is reduced or can only be maintained with a good control algorithm. In this case the performance of the load holding circuit improves with a simple proportional controller.

Figure 8 shows the same comparison of the circuit, but the capacitance C_{HA} is much larger. As a result, the positive influence of the flow feed back is weaker. In systems with long hoses between the proportional valve and the cylinder a controller on that valve can't influence the stability of the circuit much.

6. STABILTY OF THE LOAD LOWERING CIRCUIT WITH CYLINDER SPEED SIGNAL FEEDBACK

It seems natural to measure the speed of the cylinder instead of the flow across the counterbalance valve. The stability has been calculated for that alternative circuit, shown in fig. 1C, figure 8 shows the block diagram.

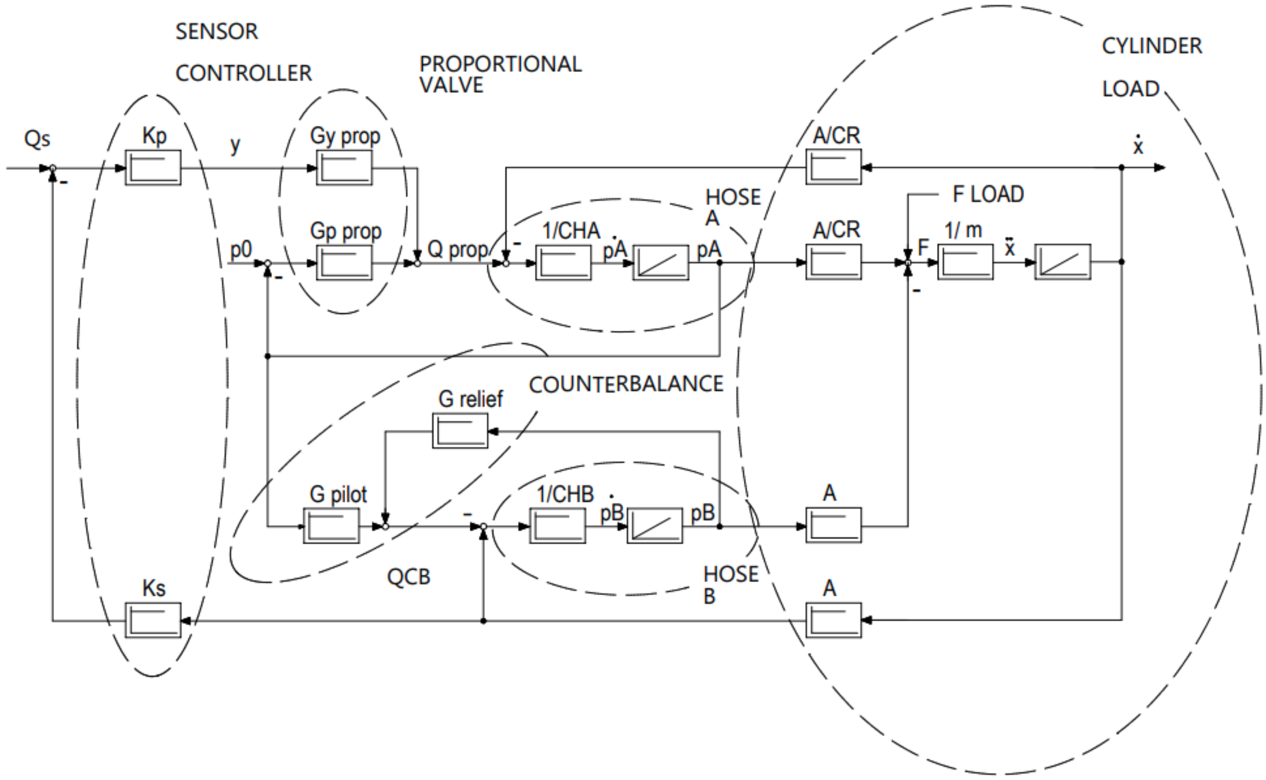


Figure 9: Block diagram of the load holding circuit with feedback of the cylinder velocity $\dot{x}$

The state variables are the same as in the system with flow feedback but the system matrix A is different, the controller and sensor appear in the a_{31} element, not in the a_{11} and a_{21}element:

$$\begin{pmatrix} \dot{p}_A \\ \dot{p}_B \\ \ddot{x} \end{pmatrix} = A * \begin{pmatrix} p_A \\ p_B \\ \dot{x} \end{pmatrix} + \begin{pmatrix} \frac{Gy_{prop}*K_S*K_S}{C_{HA}} \\ 0 \\ 0 \end{pmatrix} * Q_S + \begin{pmatrix} \frac{Gp_{prop}}{C_{HA}} \\ 0 \\ 0 \end{pmatrix} * \ p_0 + \begin{pmatrix} 0 \\ 0 \\ \frac{1}{m} \end{pmatrix} * F_{Load} \qquad (15)$$

With:

$$A = \begin{pmatrix} -\frac{Gp_{prop}}{C_{HA}} & 0 & -\frac{A*\left(\frac{1}{CR}+K_S*K_P*Gy_{prop}\right)}{C_{HA}} \\ -\frac{G_{pilot}}{C_{HB}} & -\frac{G_{relief}}{C_{HB}} & \frac{A}{C_{HB}} \\ \frac{A}{CR*m} & -\frac{A}{m} & 0 \end{pmatrix} \qquad (16)$$

As a result, the stability and the step responses are different. The feedback of the flow across the counterbalance gave better results than the feedback of the cylinder speed. Figure 10 shows a step response in that the system is at least stable with velocity feedback. The capacitance C_{HA} had to be chosen much smaller to avoid instability.

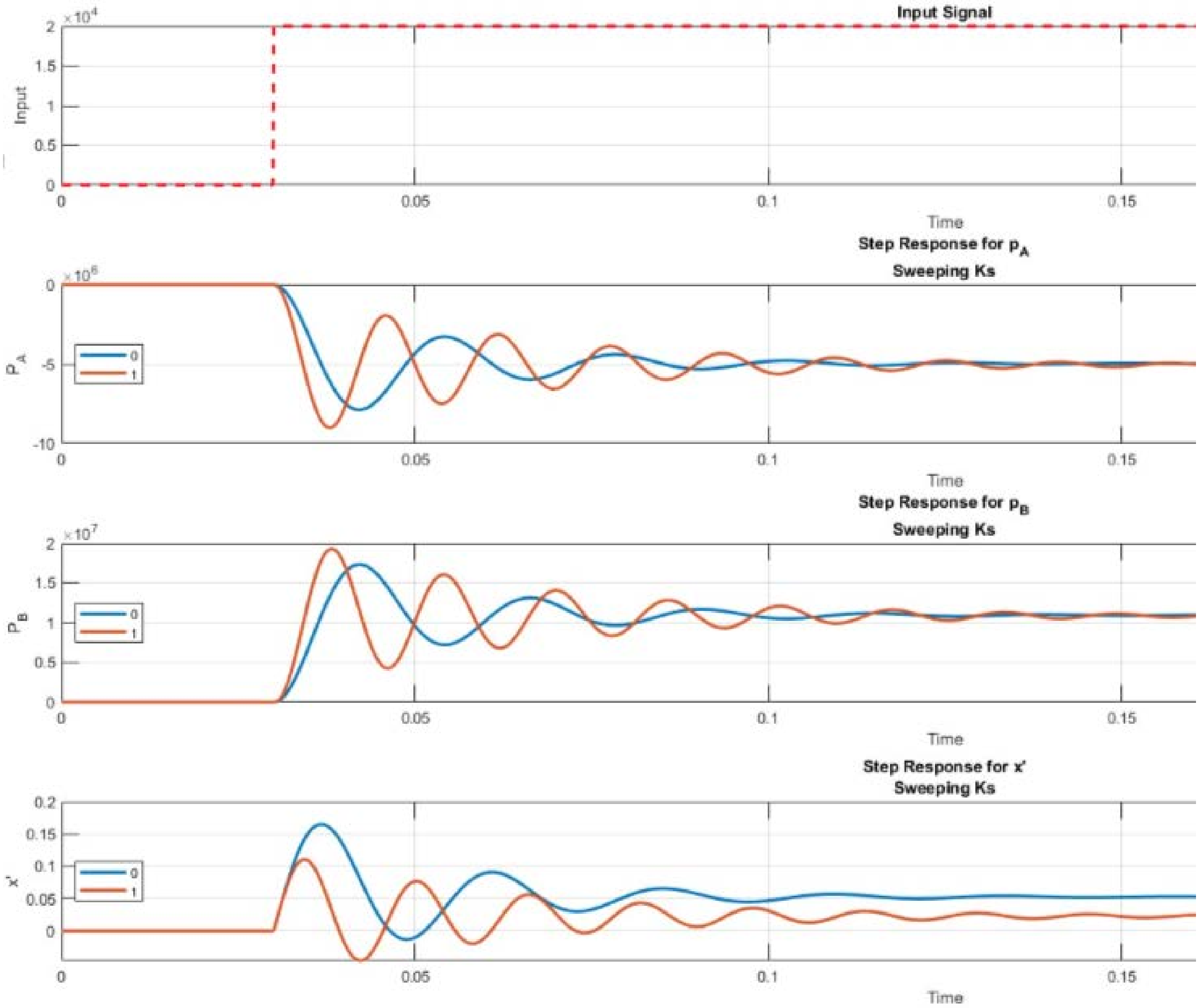


Figure 10: Step response for circuit - see fig. 8 -with ($K_S = 1$) and without ($K_S = 0$) cylinder speed feedback.

$A = .001256\ m^2$, $\boldsymbol{C_{HA} = 1.5e^{-13}\ m^3/Pa}$, $C_{HB} = 7.59e^{-14}\ m^3/(s*Pa)$, $CR = 1$, $Gp_{prop} = 1.33\ e^{-11}\ m^3/(s*Pa)$, $G_{pilot} = 3.5\ e^{-10}\ m^3/(s*Pa)$, $G_{relief} = 1.65\ e^{-10}\ m^3/(s*Pa)$, $Gy_{prop} = .133\ m^3/(s*m)$, $K_p = 10\ m*s/m^3$, $m = 500\ kg$

The system matrix in equation 16 (cylinder speed feedback) compared to the matrix in equation 9 is an indicator for the poor results of the velocity feedback. With cylinder velocity feedback there is no additional term on the main diagonal. The velocity changes after pressures p_A and p_B changed. The high inertia of the attached load delays the acceleration of the cylinder. So, when the counterbalance valve opens too far the cylinder velocity reacts with a delay. A turbine built into or near the counterbalance valve gives an earlier signal when the counterbalance valve opens too far. Therefore, a controller that uses that signal can stabilize the system more effectively.

SUMMARY

A counterbalance valve can be built with an integrated turbine as a flowmeter. A controller for a proportional valve can use that signal in a load lowering circuit.
The circuit is more stable and more accurate with the flow signal used for a simple proportional controller but there is an upper limit for the gain of the controller.
A reducing valve is more stable in load lowering circuit but is rarely used since the speed of the cylinder is difficult to predict. That problem can be solved with a flow feedback signal. The controller opens the reducing valve until the desired flow passes through the counterbalance valve.
Measuring the flow and using that signal for a closed loop gives better results with respect to stability than measuring the velocity of the cylinder and using that for a closed loop control since the cylinder with the attached mass is a slower flow meter than the turbine in or near the counterbalance valve.

NOMENCLATURE

A	Area	m^2
C_{HA}, C_{HB}	Capacitance	m^3/Pa
CR	Cylinder ratio	1
G_{relief}	Counterbalance: delta flow per delta load pressure	m^3/(s*Pa)
G_{pilot}	Counterbalance: delta flow per delta pilot pressure	m^3/(s*Pa)
Gy_{prop}	Proportional valve: delta flow per delta stroke	m^3/(s*m)
Gp_{prop}	Proportional valve: delta flow per delta pressure differential	m^3/(s*Pa)
K_S	Flow sensor factor	1
K_P	Prop controller factor	m*s/m^3
p_A, p_B	Pressure in Volume A, B	Pa/m^2
PR	Counterbalance pilot ratio (effective area for pilot pressure per effective area for load pressure)	1

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