Design of Pressure Control for Optimal Damping in Individual Metering Systems

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Abstract

Modern oil-hydraulic systems for moving heavy payloads are designed for optimised motion, but also for minimal energy loss. Individual metering technique, using separate control of the two actuator chambers, offers some advantages. A common strategy when moving the load is to control the incoming oil flow to obtain a desired speed, and the pressure at the downstream side for good efficiency. In this work analysis and design of PI (proportional-integral) pressure control is done. The adjustment of the control parameters of this loop is usually uncritical. In the worst case, the damping of the mechanical system is the only contribution. It is shown in this work, that pressure control can increase the damping of load oscillations. The influence of the P and I parameters to the system properties is investigated using the poles of the transfer function of the system. It is shown, that there is a point, where the damping factor of the system has its maximum value, and a design method for this optimisation is given. The problem ends up in a system of two equations of fourth order. A method is shown how to reduce the problem to solving one third-order equation, which is done numerically. Finally, the results are verified using simulation.

Keywords: hydraulic actuator; pressure control; individual metering

1 Introduction

Hydraulic actuators are often equipped with proportional valves. Oil flows for both chambers are controlled with orifices located on a single valve spool. The distribution of pressures and flows is designed with the geometric shape. Individual metering technique uses separate control for meter-in and meter-out. This gives an additional degree of freedom and enables several strategies for control, especially for the increasing demand of energy efficiency [1].

A standard method for individual metering is to control the oil flow to the first cylinder chamber in order to maintain a defined velocity, and concurrently to control the pressure in the tank side chamber low enough for small power loss, but high enough to have a margin to cavitation [1], [2]. Such strategies may be implemented in the embedded controller of a valve [3]. It is important, but difficult to know the system parameters, in particular the oil properties [4]. For example, additional sensors together with a model for the viscosity can help to improve the control results [5].

Beside the energy efficiency the damping of vibrations and oscillations can be an important design goal [6] [7]. With electronic control, acceleration or pressure feedback are preferred method [8]. A thorough guide how to design pressure feedback can be found in [9], an application example is shown in [10].

A completely different approach to reduce oscillations is to avoid those that are induced with command signals. The basic working principle is to filter out frequencies that could excite resonances in the physical system. An early work on this topic is found in [11]. These techniques are usually subsumed under the terms *input shaping*, *signal shaping* or *preshaping*. Successful implementation was done in electrical servo drives and in hydraulic systems as well [12]. A possible implementation in an individual metering system can be found in [13]. With a few exceptions, such filters are not often implemented in industrial controllers. The reason could be, that the

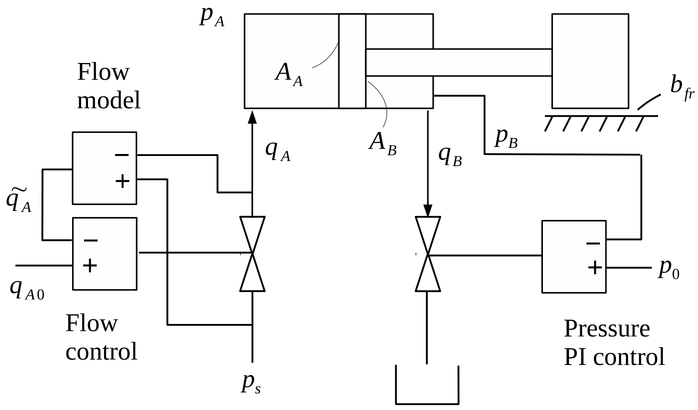

Figure 1: Scheme of the hydraulic system

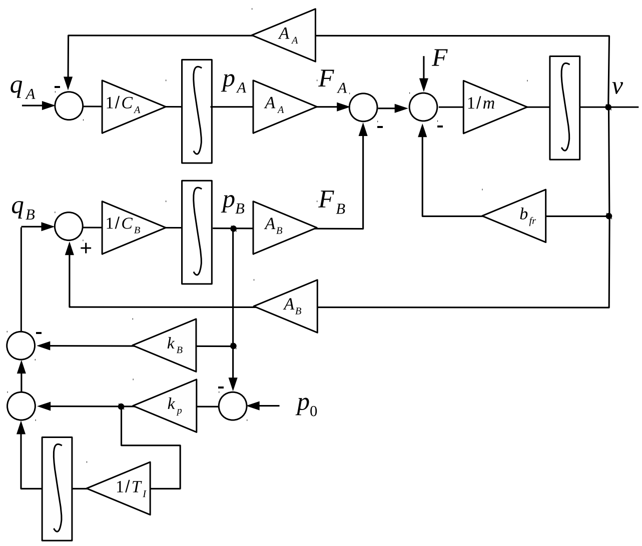

Figure 2: Block diagram of the system

dimensioning needs the work of experts. Additionally, the design requires good knowledge about the physical parameters. Furthermore, the performance is prone to changes of the parameters, like drift during operation or non-linear behaviour.

The pressure information from the downstream chamber can be used for two purposes: For active damping and for level control with respect to energy efficiency. The actual work takes a look on both aspects of pressure feedback. In the following it is shown that a PI (proportional, integral) pressure feedback loop can have an essential damping contribution to an oscillating load. A design procedure to optimise the controller is presented. Finally, it is shown by simulation, that beside the damping effect, the controller still is doing well what it is originally intended for, to keep a desired pressure level.

2 Modelling

2.1 Description of the System

Figure 1 shows a scheme for independent metering, where the oil flow into the upstream cylinder chamber is regulated with the purpose to obtain a desired motion speed. The pressure difference over an orifice is measured to calculate the flow. This flow model can be improved using oil temperature and viscosity data [5]. With the help of a control loop, a desired value for the oil flow and further for the motion speed is obtained. For the second chamber on the downstream side, a pressure controller is used to keep a constant pressure.

2.2 Dynamic Model

To model the dynamic behaviour, a linear system is used, fig. 2 shows the block diagram. The payload is modelled with a mass and linear friction, the cylinder has two chambers with compliant oil, and the electronic pressure feedback loop is a PI controller. Of course all elements in the model interact with each other, so does the flow controller. The assumption of a constant input flow, which is not affected by system dynamics, must be justified first.

The dominating elements are the mass and the compliance of the oil in the cylinders, acting as a spring. For heavy machines, the resonant frequency will be quite low, maybe up to ten Hertz. With control system, the motion is controlled up to this frequency. Usually, higher frequencies are not a goal for control, since this would require too much hydraulic power, and would lead to wear of the mechanical components.

The pressures inside the cylinders are subjected to the dynamics of the mechanical system and hence cannot change fast, even if the sensors can measure up to some kHz. Only the supply pressure can change fast according to the pump dynamics, this property is important for the flow controller.

In the model, fig. 2, flow is regarded as a constant input without any dynamics. Pressure sensors are quite fast, having upper border frequencies of several kHz. In the case of a valve with embedded electronic control the sensors are mounted inside the valve close to the metering orifices [3]. Flow control, also integrated in the valve, is done by measuring the pressure difference. The control loop finally actuates the valve spool to change the orifice. Since the characteristic frequency of the spool motion is at least ten times higher than the resonant frequency of the mechanical system, the dynamics of flow control can be neglected, and flow may be assumed to be constant at the desired level. Under this conditions, the model in fig. 2 will be a good approximation of the actual system.

The results of this work are only valid in combination with an accurate and fast flow control, like provided by the previously mentioned configuration. We did not consider other solutions like mechanical load balancing valves.

The block diagram fig. 2 contains also non-linear elements. Friction does not influence the results much, if load forces are high. The oil flow q_B depending on the pressure p_B is modelled with two factors: The feedback gain k_p that is resulting from the design procedure and the factor k_B, which models the flow change by the pressure through the orifice of the valve. This contributes an additional damping. In order to find the factor k_B for a linear model, a linearisation about the working point has to be done first. Assuming a steady motion of v_0, the oil flow on the B-side will be

$$q_{B0} = A_B \cdot v_0. \tag{1}$$

With an opening factor η and the nominal data of the valve, denoted by the index N, the oil flow is

$$q_{B0} = \eta \frac{q_N}{\sqrt{p_N}} \sqrt{p_{B0}}. \tag{2}$$

The pressure p_{B0} will be the setvalue of the pressure controller p_0, then the required valve opening can be found with

$$\eta = \frac{q_{B0}}{q_N} \sqrt{\frac{p_N}{p_0}}. \tag{3}$$

For a small change of flow follows

$$\delta q_B = \left. \frac{\partial q_B}{\partial p_B} \right|_{p_B = p_0} \cdot \delta p_B, \tag{4}$$

where

$$\left. \frac{\partial q_B}{\partial p_B} \right|_{p_B = p_0} = -\eta \frac{q_N}{\sqrt{p_N}} \frac{1}{2\sqrt{p_0}}. \tag{5}$$

For a linear model, the factor is found with

$$k_B = \eta \frac{q_N}{\sqrt{p_N}} \frac{1}{2\sqrt{p_0}}. \tag{6}$$

Now the basic differential equations for the system in fig. 2 can be formulated,

$$\dot{p}_A(t) = \frac{1}{C_A} \left(q_A(t) - A_A v(t) \right), \tag{7}$$

$$\begin{aligned} &\dot{p}_B(t) = \\ &\tfrac{1}{C_B} \left(k_p p_0 - (k_B + k_p)\, p_B(t) + \tfrac{k_p}{T_i} \int \left(p_0 - p_B(t) \right) \mathrm{d}t + A_B v(t) \right), \end{aligned} \tag{8}$$

and

$$\dot{v}(t) = \frac{1}{m} \left(F(t) - b_{fr} v(t) + A_A p_A(t) - A_B p_b(t) \right). \tag{9}$$

2.3 Transfer Function

For the present work, a transfer function representation was chosen, where the input is the disturbing force $F(t)$, and the output the velocity $v(t)$ of the cylinder piston. For this purpose, it is required to find the transfer function

$$G(s) = \frac{v(s)}{F(s)}. \tag{10}$$

Applying Laplace transform to eq. (7), (8) and (9) yields

$$p_A = \frac{1}{sC_A}\left(q_A - A_A v\right), \tag{11}$$

$$p_B = \frac{A_B v s + \left(k_p s + \frac{k_p}{T_I}\right) p_0}{s^2 C_B + s\left(k_B + k_p\right) + \frac{k_p}{T_I}}, \tag{12}$$

and

$$v = \frac{F + A_A p_A - A_B p_B}{ms + b}. \tag{13}$$

The pressures in eq. (13) are substituted using eqs. (11) and (12). After setting the inputs of no interest, p_0 and q_A, to zero, the transfer function for the motion speed reacting to a disturbing force input F follows

$$\frac{v(s)}{F(s)} = \frac{1}{mC_B} \cdot \frac{s^3 C_B + s^2(k_B + k_p) + s\frac{k_p}{T_I}}{s^4 + s^3 p + s^2 q + sr + t} \tag{14}$$

with the abbreviations

$$\begin{aligned} p &= \tfrac{k_B + k_p}{C_B} + \tfrac{b}{m} \\ q &= \tfrac{A_B{}^2}{mC_B} + \tfrac{A_A{}^2}{mC_A} + \tfrac{k_p}{C_B T_I} + \tfrac{b}{m} \cdot \tfrac{k_B + k_p}{C_B} \\ r &= \tfrac{b}{m} \cdot \tfrac{k_p}{C_B T_I} + \tfrac{A_A{}^2}{mC_A} \cdot \tfrac{k_B + k_p}{C_B} \\ t &= \tfrac{A_A{}^2}{mC_A} \cdot \tfrac{k_p}{C_B T_I}. \end{aligned} \tag{15}$$

The reaction of the system to a disturbing force, eq. (14), includes also the controller parameters and is of fourth order. To study the behaviour, the poles of the transfer function are to be found. The algebraic solution leads to very complex expressions, hence a numerical approach is used to find the roots of the transfer function.

3 Discussion of the Poles

At first the effect of pure proportional control without integrator is studied. Letting T_i go to infinity in eq. (14) leads to the third order transfer function

$$\frac{v(s)}{F(s)} = \frac{s^2 C_B + s(k_B + k_p)}{s^3 mC_B + s^2\left(m(k_B + k_p) + bC_B\right) + s\left(A_B^2 + A_A^2 \frac{C_B}{C_A} + b(k_B + k_p)\right) + A_A^2 \frac{k_B + k_p}{C_A}} \tag{16}$$

with one pair of conjugate complex poles and one real pole. Figure 3 shows one complex pole, the others are not of interest. It can be seen, that with small values of k_p (weak control) the natural frequency is higher. Both cylinder chambers contribute with their stiffness. With strong control (high values for k_p) the natural frequency goes down, since a perfect constant pressure level in chamber B offers no stiffness.

At a certain value of k_p the real part has a minimum, this is the leftmost point of the graph displayed in fig. 3. This point is the starting point for the next step, adding the I portion of control.

If PI control is applied, again eq. (14) is to be used, which has four poles. One complex conjugate pair is mainly determined by k_p. Introducing an I portion means starting with high values of T_i. The two poles that are introduced with T_i are initially real. With decreasing values of T_i, the real poles walk along the real axis until they meet and split up into conjugate complex poles, shown in fig. 4. The first pair of poles walk towards negative real parts and meet the second pair at a certain point. This point is the goal of the design procedure, where the real part of any pole has its minimum. This means, that all transients have a fast decay. Further decreasing T_i causes one pair going to negative real values, the other pair into positive direction, which means, damping decreases again.

It can be seen that the the loop does not get unstable, but can lose the damping effect. Of course this is valid only with the assumption, that the friction is linear [9]. This can be assumed, when friction force is low compared to the dynamic load forces.

The minimal real part for all poles is obtained, when the pairs meet at the same point. Hence this point is a goal for the design of the control parameters k_p and T_i.

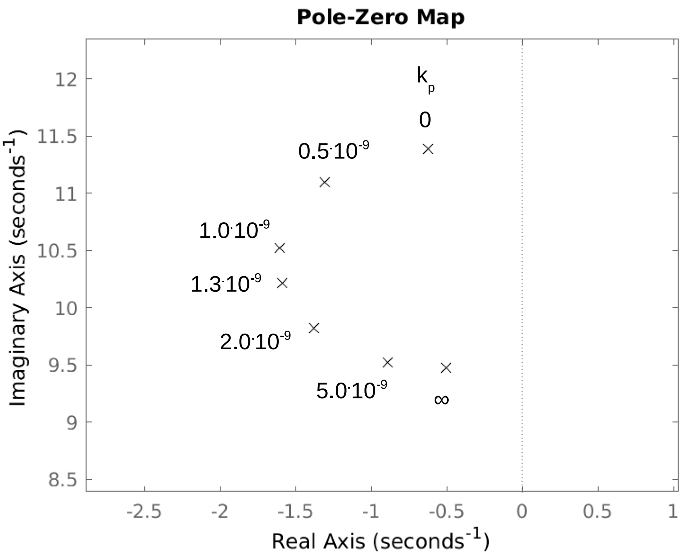

Figure 3: Pole-zero map of P-controlled system, 'x' denote the poles

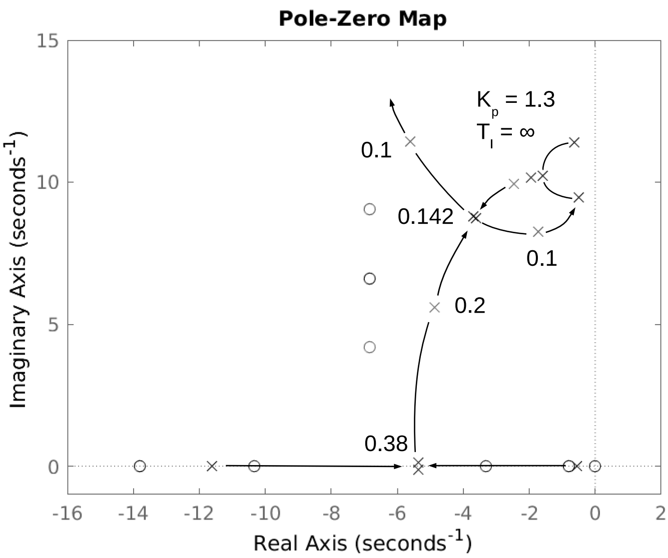

Figure 4: Root locus of PI-controlled system, 'x' denote the poles, 'o' the zeroes

4 Controller Design for Optimal Damping

The controller design problem is to find values for k_p and T_i, such that the real parts of the poles have minimal values. From fig. 3 it can be seen that there exists a solution with two identical conjugate complex pairs of poles

$$s_{01} = s_{02} = a \pm b\mathrm{j}. \tag{17}$$

Then the denominator of a transfer function of fourth order will have the form

$$D(s) = s^4 - 4as^3 + s^2\left(6a^2 + 2b^2\right) - 4as\left(a^2 + b^2\right) + \left(a^2 + b^2\right)^2, \tag{18}$$

which can be used to compare coefficients with the denominator of eq. (14). The comparison yields two non-linear equations for k_p and T_i, one of them is of fourth order. An algebraic solution can be found with a proper tool, but delivers an extremely complicated formula as result. In order to make the work easier, this problem can be solved in two steps. Then the resulting terms are acceptable simple, and experimenting is easier.

By comparing the coefficient of s^3, the real part of the poles can be found as a function of k_p,

$$a = -\frac{1}{4}\left(\frac{k_B + k_p}{C_B} + \frac{b_{fr}}{m}\right). \tag{19}$$

Equation (19) shows, that the proportional gain contributes to the damping in the same way like friction b_{fr}. Both terms have the dimension t^{-1}, they are time constants of a first order system, as it can be seen in the block diagram (fig. 2). Equation (19) further shows, that even a frictionless system with $b_{fr} = 0$ could be stabilised.

A control loop for the pressure is also a kind of pressure feedback, which is often used for damping of oscillations [9]. Unfortunately eq. (19) is only a necessary condition for the optimum, but not sufficient. We cannot increase the gain, and the damping associated with it, arbitrarily.

4.1 Integral Reset Time T_i as a Function of Proportional Gain k_p

Without algebraic solution for the poles, an additional condition for the integral reset time T_i can be found. Comparing the second order coefficient yields

$$q = 6a^2 + 2b^2, \tag{20}$$

the first order coefficient

$$r = -4a\left(a^2 + b^2\right). \tag{21}$$

Eliminating b^2 in eq. (21) using eq. (20) delivers

$$r = 8a^3 - 2aq. \tag{22}$$

Substituting q with the physical parameters from eq. (15) gives a condition for the I part,

$$T_I = -\frac{k_p\left(2am + b_{fr}\right)}{\frac{A_a^2}{C_A}(k_B + k_p) - 8a^3 m C_B + 2a\left(A_B^2 + A_A^2\frac{C_B}{C_A} + b_{fr}(k_B + k_p)\right)}. \tag{23}$$

With a given k_p, a is known from eq. (19), and further, T_i can be found with eq. (23).

4.2 Numerical Approach for Gain k_p

At this point, any value assumed for k_p yields the real part a of a double pair of poles with eq. (19), then eq. (23) finds the corresponding value for T_i. It turns out, that only if k_p with the minimal real part is chosen, the value for T_i is an optimal solution. A numerical solution for the poles of eq. (14) can be found easily with numerical tools, e.g. Matlab™ or with the function *solve* from the *Python* package *sympy*. Given a value for k_p, the value for T_i and the poles are available now. This allows experiments for creating a map of poles, or to find a good pole placement.

To find the solution faster, an iterative approach can be used. The optimal solution are two pairs of poles with the same real parts and imaginary parts. As criterion, the sum of the *Manhattan* distances of the poles from the point with the mean real and mean imaginary parts of the poles is used,

$$\sigma = \sum_{i=1}^{4}\left(\left|a_i - \bar{a}\right| + \left|b_i - \bar{b}\right|\right) \tag{24}$$

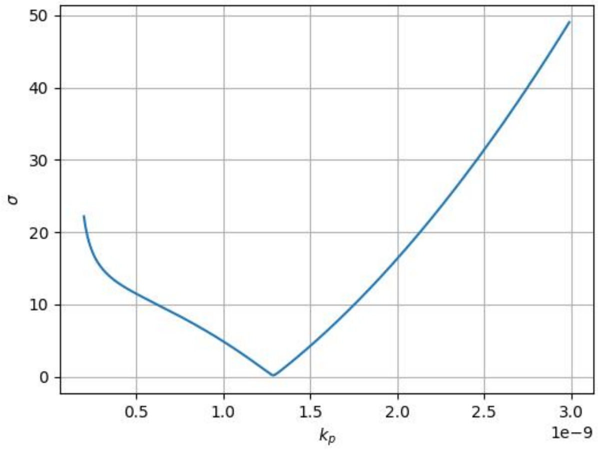

Figure 5: Manhattan distances of the poles with minimum of zero at the optimal solution

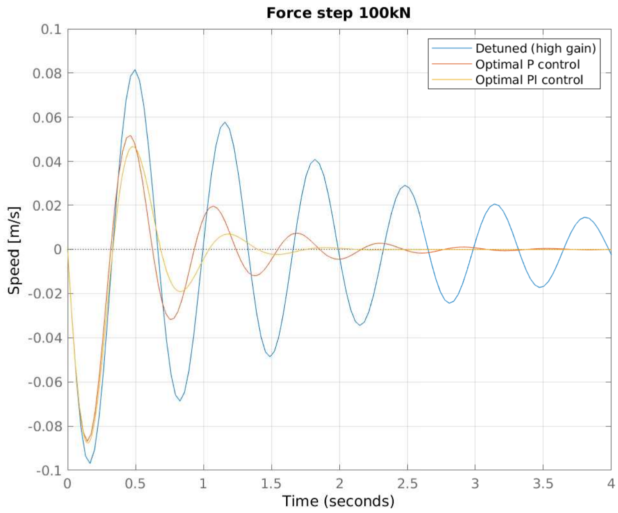

Figure 6: Step response of transfer function

with

$$\bar{a} = \sum_{i=1}^{4} a_i, \quad \bar{b} = \sum_{i=1}^{4} b_i. \tag{25}$$

In order to prove, that eq. (24) is a suitable measure, the values of σ over k_p are displayed in fig. 5. The graph shows clearly a minimum at zero, which means, that there is a point, where all poles meet, and that the Manhattan distance can be used to find this optimum with a numerical procedure. This can be done in *Python* with the *minimize_scalar* function of the *scipy.optimize* package in *Scipy*, or with Matlab™.

5 Simulation Results

For the verification of the proposed design procedure an example hydraulic system was assumed. The mass m is chosen with 100 000 kg, the cylinder areas are $A_A = 0.03\,\text{m}^2$ and $A_B = 0.02\,\text{m}^2$, the chamber capacitances are $C_A = C_B = 10^{-10}\,\text{m}^3\text{Pa}^{-1}$ and the friction is assumed to be viscous with $b_{fr} = 100\,\text{kNsm}^{-1}$. A steady speed v_0 was chosen with 41 mm/s, which causes as steady oil flow q_B of 50 l/min. From this follows the factor k_B with $8 \cdot 10^{-11}\,\text{m}^3\text{Pa}^{-1}\text{s}^{-1}$. A disturbing force step is applied with $F = 100\,\text{kN}$. For the optimal pressure control gain k_p a value of $1.3 \cdot 10^{-9}\,\text{m}^3\text{Pa}^{-1}\text{s}^{-1}$ was found, and for the integrating part a T_i of 0.142 s.

The first experiment is to simulate the linear model according to eq. (14). Figure 6 shows the optimal solution with high damping, a result with proportional control only, and a detuned solution with very high controller gain k_p.

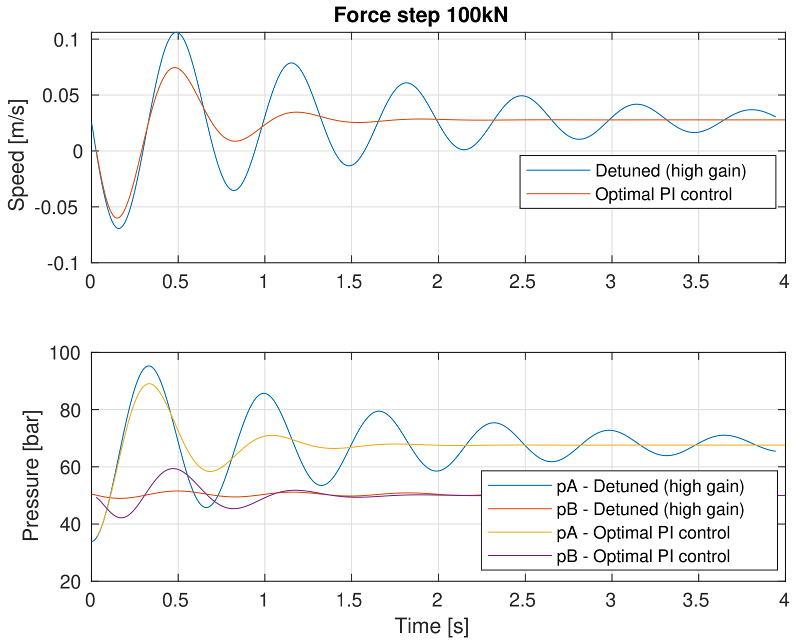

Figure 7: Step response of the simulated non-linear system

A second experiment is done according to the system in fig. 2 in Matlab®/Simulink® with a non-linear system. A constant oil flow is maintained with a fast proportional valve and a flow controller. For the downstream pressure controller also such a valve is used. The set value p_0 is 50 bar, the power supply pressure is 200 bar. The force step is delayed to allow the system to settle before the experiment. Figure 7 shows the speeds and the cylinder pressures over time.

6 Summary and Conclusions

In this work pressure feedback control in hydraulic actuators for motion control is investigated. Such loops are often used in individual metering systems to stabilise the pressure on the downstream side, while a flow controller governs the motion speed. Another well-known technique is the use of a hydro-mechanical pressure relief valve. This would be equivalent to an electronic P control with a very high gain and has a low damping contribution. Alternatively, electronic pressure feedback with a high-pass filter maybe applied to bring additional damping into the system. Since a control loop is also a kind of feedback, and the PI controller can be regarded as filter, the configuration can expose damping behaviour too. It turns out that the parameters can be optimised with respect to this property, and a design procedure for this is presented. A simulation experiment shows that this works, while the pressure stabilising feature still is in function.

The design is strictly connected with the physical system parameters and therefore is sensible against changes, for example drift during the operation. In comparison to this, the classical pressure feedback using a separate filter is not so sensible to parameter errors and can provide higher damping ratios. But it is shown, that there is an essential damping contribution by the PI controller, when the parameters are adjusted accordingly.

Nomenclature

Designation	Denotation	Unit
q_A	Constant (controlled) input flow	m^3/s
q_B	Downstream flow determined by pressure controller	m^3/s
q_{B0}	Steady motion flow to chamber B	m^3/s
q_N	Nominal oil flow of valve	m^3/s
p_N	Nominal pressure of valve	Pa, bar
p_A, p_B	Pressures in chamber A and B	Pa, bar
p_0	Set value for pressure control	Pa, bar
p_s	Power supply pressure	Pa, bar
A_A, A_B	Piston areas	m^2
C_A, C_B	Hydraulic capacitances	m^3/Pa
k_p	Proportional control gain	m^3/Pa s
k_B	Flow change due to pressure change (linearised)	m^3/Pa s
T_i	Integral reset time	s
v	Velocity of piston and load	m/s
v_0	Steady motion velocity of piston and load	m/s
F	Disturbing force applied to load	N
F_A, F_B	Force contributions of cylinder chambers	N
m	Mass of the load	kg
b_{fr}	Viscous friction coefficient	Ns/m
η	Valve opening ratio, 0..1	

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