HYDRAULIC FORCE CONTROL UNDER DISTURBANCES REALIZED AS PRESSURE CONTROL

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ABSTRACT

Hydraulic cylinders are typical actuators in force control applications. The friction force is the most serious disturbance of a force control, when the piston does not move. When the piston is moving the volume flow and changes in the controlled volume are disturbances. In this study the force control is realized as the piston side pressure control of a hydraulic cylinder. Three different ways to control the pressure are compared. The 3-way valve is used in this study. The influence of the piston movement and spring force on the performance of the force control is studied experimentally. The theoretical background of force and pressure controls is presented. Experimental results are shown and different solution compared. Two different ways to modify the conventional PI controller are presented and tuned to fulfill the system requirements.

KEYWORDS

Force control, Pressure control, Gain schedule,

NOMENCLATURE

\begin{tabular}{lll}
A1, A & Piston area & [m^2] \\
A2 & Rod side area & [m^2] \\
B & Effective bulk modulus & [Pa] \\
b & Viscous friction coefficient & [Ns/m] \\
F & Force & [N] \\
F_C & Coulomb friction & [N] \\
F_S & Static friction & [N] \\
F_p & Friction force & [N] \\
G & Gain & [m^3/Ns] \\
k & Spring constant & [N/m] \\
Kc & Valve flow/pressure-gain & [m^3/sPa] \\
m & Mass & [kg] \\
p1 & Piston side pressure & [Pa] \\
p2 & Rod side pressure & [Pa] \\
p10 & Piston side start pressure & [Pa] \\
s & Laplace operator & [s^{-1}] \\
V10 & Piston side pipe line volume & [m^3] \\
v_{ref} & Friction reference velocity & [m/s] \\
v & Piston velocity & [m/s] \\
y0 & Piston start position & [m] \\
\end{tabular}
INTRODUCTION

In most hydraulic applications the only practical cylinder is a single rod double acting cylinder. The piston movement is an essential part of most potential hydraulic force control applications, as in mobile machines and paper industries as well as in milling, car, steel and in general heavy material handling applications. The movement of the piston means that a certain volume flow is required through the control valve. The characteristics of a control valve are an important factor in a control loop [1]. The natural feedback of a force servo system is the actual force applied to the load. The practical installation of a force transducer for control purposes is a very demanding task. Force transducers are also quite expensive. Pressure transducers offer a simple and economical way to realize the force feedback.

In this study the hydraulic differential cylinder force control is realized as the pressure control in the piston side applying the supply pressure to the rod side. It is easy to realize pressure control in a constant volume with no load flow. However, traditional pressure control valves have some dynamic problems in spite of their fairly good steady state performance [2]. The tuning of these valves to application changes is almost impossible. Valves with electrical pressure feedback can be re-tuned case by case [3]. The load flow and controlled volume vary in many practical applications, too. The problem with commercial pressure control valves is that valve manufacturers inform very poorly especially dynamic characteristics of valves. In this paper pressure control is tuned in all three cases with the nominal volume. The load flow, the changes of the flow direction, and controlled volume are used to study the performance of the pressure control methods.

The biggest drawback in using pressure transducers to force feedback is that the friction forces in a cylinder remain outside of the control loop. This means an error in the actual force applied to the load. The steady state and dynamic performance of a control valve are quite essential for the quality of a force control [1], [4] and [5]. The force control accuracy and robustness have been improved in the following ways. The cylinder friction force is modeled based on measurements with different load masses and velocities and it is applied to reduce the influence of friction forces. Two ways to modify controller gains are applied in order to reduce the influence of load flow and volume changes on force control dynamics.

DIFFERENT PRESSURE CONTROL SCHEMAS

A hydro-mechanical pressure control valve and a 3-way direction control valve with the PI-controller as well as a hydro-mechanical pressure control valve with an outer pressure control loop are tuned with the 0.3 liters volume (steel pipe). The measured pressure error step responses are shown in Fig.1 the electro-mechanical pressure control valve, in Fig.2 the electro-mechanical pressure control valve with outer pressure control loop, and in Fig.3 the 3-way valve.
All responses are from 4 MPa to 8 MPa and vice versa. The response of the 3-way valve is noticeably faster and more stable than the responses of other cases. The steady state characteristics of the 3-way valve are also better than the characteristics of two others. When the volume is changed from the 1 m pipe to the 5 m hose the all dynamic responses of the electro-mechanical valve with outer pressure control loop are unstable with the tuning based on the case of the 1 m pipe. The pressure error responses of the electro-mechanical valve and the 3-way valve are shown in Fig. 4 and Fig. 5. The response of the electro-mechanical valve is still stable, but the settling time is quite long and there are strong oscillations. The response of the 3-way valve is still quite good with the short settling time and the acceptable overshoots. When a step-wise changing load flow is applied through the valves, only the responses of the 3-way valve are acceptable.

On these basis the 3-way direction control valve is used in the final force control assembly.

**FORCE CONTROL**

The friction forces are the most serious disturbance in the force control when it is realized with a pressure control. The friction forces have to be compensated as well as possible to achieve good force accuracy. Eq. (1) is used to compensate the friction forces.

\[
\sum F_{\mu} = b \cdot v + F_c + (F_s - F_c) e^{-\frac{t}{\tau_{df}}}
\]  

(1)

Because the pressure sensor signal filtering bandwidth is limited by the system dynamics, the cylinder force might oscillate, if the friction forces are fully compensated. In practice this means that at least 20-30 N have to be left outside the compensation. When the maximum cylinder force is 17 kN, the equivalent force error is 0.15%. The actual friction force is estimated from the cylinder pressure measurements using different load masses and load velocities. The coefficients to the Eq.(1) have been chosen based on these measurements. After the friction force compensation, the force equation, Eq. (2), is used to control the actual cylinder force.

\[
F = [p_1 A_1 - p_2 A_2] - \sum F_{\mu}
\]  

(2)

Because the rod side is directly connected to the supply pressure, only the piston side pressure is a controlled variable.
The friction compensation is placed to the feedback loop and the compensation sign is determined based on the sign of the pressure force difference. Since no force sensors are installed to the system, the friction force compensation is verified with a simple test: the controlled net force, according to the Eq. (2), in both directions is set to the 50 N. When the average velocity of the load is the same in both directions the load motion can be stopped by a hand.

When the load is not moving, the dynamics of the cylinder chamber pressure can be simplified to Eq. (3)

\[
P_l = \frac{u}{\frac{u_{\text{max}}}{V} - \frac{s + 1}{B*Kc}}
\]  

(3)

The small force step responses in the different load position are shown in Fig. 6, the load does not move. In this case the friction forces are not compensated.

Figure 6. Force step responses at rear end (left), middle, and front end (right) of cylinder, constant controller gains.

The response dynamics depends noticeably the position of the piston. With bigger force steps, over the friction range, the friction compensation is in use. In the laboratory tests the external load is achieved with the spring which can be installed to the different points of the stroke. This way the same force step responses are measured with different controlled oil volumes. Force step responses are measured both upwards and downwards. Since the influence of the variations in the controlled volume can not be other wise compensated, to further improve the force control dynamic behavior, different controller schemas are used. The relative volume change, especially at the rear end of the cylinder, has significantly bigger influence on the force response than the compressibility flow has.

The simplified model shown schematically in Fig. 7 is used in the preliminary force controller tuning.

CONVENTIONAL PI CONTROLLER

The tight force control tuning at the rear end yields to an oscillatory response at the rod side end of the cylinder. The controller tuning that fulfills the requirements at the other end does not do that at the opposite end. The problem is that in order to achieve fast responses in the whole stroke range different gain combinations are needed at both ends of the stroke. The gain of the integrator should be high and the proportional gain low at the rear end of the cylinder, and vice versa at the rod side end of the cylinder.

CONTROLLER GAINS AS A FUNCTION OF FORCE ERROR

When the position measurement is not available, a quite well behave controller tuning can be achieved with a simple gain scheduling. The basic controller tuning is done in a large volume with smaller force errors. The gain scheduling is realized in the following way. The PI-controller proportional gain is increased and integrator gain is decreased with decreasing force error. Fig. 8 shows the pressure response with the same basic tuning at both ends of the cylinder. The settling times of both responses are quite short. The overshoot at the rod end the cylinder is rather big, but the response stays stable. With a little slower rod end tuning the overshoot can be reduced already remarkably if 0.5 seconds settling time could be accepted.
Pressures in the cylinder ends

Figure 8. Pressure response in both ends of stroke, with gain scheduling

CONTROLLER GAINS AS A FUNCTION OF PISTON POSITION

When the load position measurement is also available, the variations in the system dynamics can be taken into account. An adaptive force controller is an efficient way to improve the pressure control dynamics. A traditional way to increase the speed of responses of a PI-controller is to use a forward loop.

In this case the controller gains are varied according the volume that is determined from the piston position and the DC gain relative to the natural frequency is kept constant all the time.

Basically this controller schema is based on the cylinder dynamic model and the model-based cylinder force control. The actual cylinder position is measured as well as the cylinder chamber pressures. The model is used to estimate the pressure control behavior and the controller gains are chosen according to the model time constant.

Fig. 9 shows the step response of the 2000 N force at the middle and the both ends of the cylinder.

The force control dynamics can be remarkably improved over the whole stroke length when the adaptive controller solution is used. The settling times are short and overshoots are reasonable. The force control can be designed to be very robust against the variations of the load flow as well as in the controlled volume.

Fig. 10 shows the behavior of the force control when the supply pressure decreases and increases step-wise. The supply pressure changes used in this test are very big, but these kinds of variations might occur in mobile hydraulic applications. Variations in the controlled force are very low when compared to the variations in the supply pressure.

SUMMARY

When the cylinder force control is realized as pressure control different solutions are available. As the system requirements increases, the best results can be achieved with a 3-way direction control valve. Even without the position measurement the force control accuracy can be remarkably improved with nonlinear controller gains. With the position measurement and adaptive pressure controller the optimized controller tuning can be found for every operation point. The force control accuracy is limited by the friction compensation and pressure signal quality.
REFERENCES


2. Parker Hydraulics; pressure control valve Catalogue of VMY10


5. Tunay, I., Rodin, E.Y., Beck A.A., Modelling and Robust Control design for aircraft brake hydraulics, IEEE Transactions on control systems technology, Vol. 9, no 2, March 2001