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FLOW VS. PRESSURE CONTROL SERVOVALVES IN FORCE-BASED POSITION CONTROL APPLICATIONS

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ABSTRACT

In teleoperation applications torque controlled joints are often required. If hydraulic actuator is driven by flow control servovalve, a force-based position control requires tuning of inner loop force controller and force or pressure sensor. An alternative for flow control servovalves are pressure servovalves which are seldom used in industrial applications. Advantage of this technology should be that tuning of inner force-control loop should be avoided and also more importantly addition of pressure and/or force feedback sensors should not required. Sensors are often the most likely component to fail while working in hazardous environments. Our aim is to compare three kinds of inner loop force controls. First, we use pressure servovalve in position control application without inner loop pressure or force sensors. In remaining two cases we use force control in the inner control loop with flow and pressure control servovalve. Three controller alternatives are compared and the results are discussed.

KEY WORDS

Pressure servovalve, flow servovalve, force control, position control

NOMENCLATURE

A: Cylinder pressure area
B: System's bulk modulus
V0: System's dead volume
m: Load's mass
b: Load's viscosity friction
k: Load's stiffness

K_q : Flowservo's flowgain
K_P: P-controller gain
k_h : Cylinder's spring constant
K_{PR} : Pressureservo's pressure gain
t : Pressureservo's time constant

X : Cylinder position
X_R : Cylinder position reference
$\dot{X}$: Cylinder velocity
$\dot{X}_R$: Cylinder velocity reference
$\ddot{X}_R$: Cylinder acceleration reference
K_V : Velocity error gain
e : Error
ω_n : Natural frequency
ζ : Damping
QN : Flow

INTRODUCTION

The fundamental difference between electric and hydraulic actuators is that electric actuators are essentially torques sources whereas hydraulic actuators are velocity sources. In other words, with the electric DC-motors actuator output torque is proportional to motor input current. In case of hydraulics control input is proportional to actuator output velocity. This fundamental difference is not usually critical in traditional position control applications. However, in teleoperation applications torque controlled joints are often required. In hydraulics these applications are known as force-based position control applications, see Figure 1, where the principle of control structure is shown. In this case of usual servohydraulic hardware setup where hydraulic actuator (motor or cylinder) is driven by flow control servovalve, a force-based position control requires tuning of inner loop force controller. Inner loop force controller can be simple e.g. PI-type controller or more complex non-linear controller. Force feedback can either be based on pressure or force (load-cell) feedback.

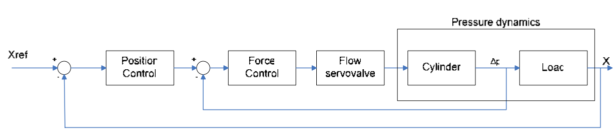

Figure 1 Force based position controller utilizing flow control servovalve and inner force control loop

An alternative for flow control servovalves are pressure control servovalves which are seldom used in industrial applications anymore most likely because of their high price. Pressure servovalves give pressure difference, which is proportional to input signal. Conventional pressure servovalves operate based on hydro-mechanical pressure feedback principle and therefore do not require pressure feedback sensor. Therefore, the big advantage of this old almost forgotten technology should be that tuning of inner force-control loop should be avoided and also more importantly addition of pressure and/or force feedback sensors is not required. Sensors are often the most likely component to fail while working in hazardous environment. Still another advantage of pressure servovalves is the high bandwidth of valve response. For example, a commercial pressure servovalve blocked port pressure control bandwidth is close to 200 Hz [3]. This kind of pressure control bandwidth is very difficult to achieve with a controller combination of a servovalve and pressure feedback transducers. These design factors mentioned are of the major concern in application where high reliability, bandwidth and robustness are required. In the Figure 2 position controller utilizing pressure servovalve is shown. In this kind of control scheme no force/pressure feedback transducer is required and thus inner loop force controller is removed.

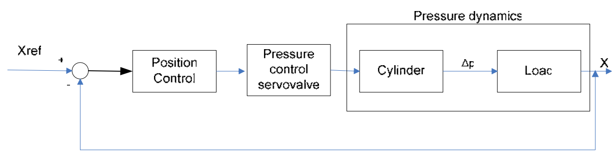

Figure 2 Force based position controller utilizing pressure control servovalve

In this study, we compare two alternatives, namely flow and pressure control servovalves in teleoperation application requiring inner loop force control. Our aim is to compare three kinds of force controls. First, we use pressure servovalve in position control application without inner loop pressure or force sensors. In remaining two cases we use force control in the inner control loop with flow and pressure control servovalve. The use of pressure control servovalve with pressure sensor has the advantage that in case of sensor failure force control still functions. Thus the operator should be able to complete the given task with somewhat decreased force feedback sensitivity. With this last option, if pressure control servovalve is operating under pressure sensor feedback, the force feedback accuracy should be as good as with flow control servovalve and pressure feedback combination. Therefore, pressure control servovalve should have the best robustness but also equally good force sensitivity as compared to flow control servovalve. Three controller alternatives are compared and the results are discussed.

THEORETICAL BACKGROUND OF FORCE CONTROL

Flow servovalve's force control loop can be described by following block diagram [1]:

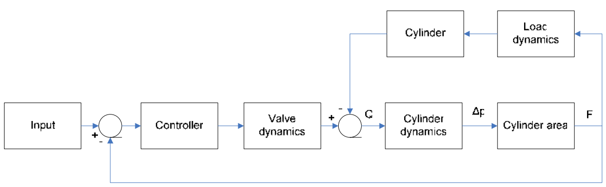

Figure 3 Flow servovalve's force control block diagram

The valve flow gain is K_q and controller gain is K_P

Cylinder pressure dynamics model can be described as an actuator transfer function from flow to pressure:

$$G_A(s) = \frac{2AB}{sV_0} \tag{1}$$

The feedback path of the system consists of the load transfer function from force to position and of the cylinder transfer function from position to flow:

$$G_L(s) = \frac{1}{ms^2 + bs + k} \tag{2}$$

$$G_P(s) = As \tag{3}$$

Collecting the transfer functions (1, 2, 3) together and describing the gains according to Figure 3 yields:

$$G(s) = 2K_q K_p \frac{AB(ms^2 + bs + k)}{s(V_0 ms^2 + V_0 bs + V_0 k + 2A^2 B)} \tag{4}$$

The hydraulic spring constant is defined as:

$$k_h = \frac{2BA^2}{V_0} \tag{5}$$

Applying equation (5) to (4) yields the open loop transfer function from voltage to force:

$$G(s) = k_h K_P \frac{K_q}{A} \frac{(ms^2 + bs + k)}{s(ms^2 + bs + k + k_h)} \tag{6}$$

From transfer function (6) can be seen that the poles of the load become the zeros of the open loop transfer function of the system. The actuator transfer function will not affect the zeros, which means that they can not be changed with PID-type controller as shown in [2]. As indicated in [2], these lightly damped open-loop zeros limit the bandwidth of the closed-loop force system.

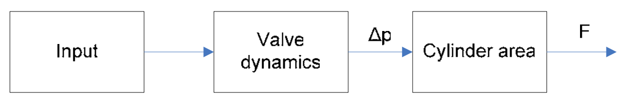

Figure 4 Pressure servovalve's force control block diagram

On the other hand, in the case of pressure control servovalve, the gain from input current to pressure difference is K_{PR}. Fast valve dynamics can assumed to obey first order model with a time constant τ. Cylinder area is A and the model from valve input current to output force is:

$$G(s) = \frac{K_{PR}}{ts + 1} A \tag{7}$$

As we can see, there are no zeros in open loop, so bandwidth is bigger and force control should be also faster. According this theory, flow servovalve should be controlled with some other controller than PID-type, if we want to get as fast response as pressure control servovalve has.

CONTROLLERS TUNING PRINCIPLES

As described above pressure control servovalve doesn't require inner loop force/pressure controller. One of the test control cases was pressure control servovalve with inner-loop control with pressure feedback. Feedforward control must be used in this case. Pressure servovalve needs some input current so that it would keep pressure difference. Anyway P-type controller gives signal only if there is some error.

For flow control servovalves typically a P- or PI-type of force/pressure controller is required for inner loop force control. In this study, we used P-type controller due to faster response. As described above pressure control servovalve doesn't necessarily require inner loop force/pressure controller. However, we also try in third test case pressure control servovalve with inner-loop pressure feedback. Inner loop pressure controller is a P-controller in parallel with pressure demand.

Upper-loop is also a simple PD-type position controller with desired acceleration as a new input to the system is described by following equation:

$$F = m(\ddot{X}_R + K_V(\dot{X}_R - \dot{X}) + K_P(X_R - X)) \qquad (8)$$

The final closed-loop system becomes under this choose of feedback law:

$$\ddot{e} + K_V\dot{e} + K_P e = 0 \qquad (9)$$

The error equation (9) can be transformed to the s-domain, which results the following:

$$s^2 + 2\omega_n\zeta s + \omega_n^2 = 0 \qquad (10)$$

$$s^2 + K_V s + K_P = 0 \qquad (11)$$

, where

$$K_V = 2\omega_n\zeta s \qquad (12)$$

$$K_P = \omega_n^2 \qquad (13)$$

Where second equation corresponds to well known second order characteristic equation with damping ζ and natural frequency ω_n. In the servocontrol the damping ratio of $\zeta = 1.0$ is highly desirable because it corresponds to situation of the fastest possible non-oscillatory response of the system to a step input. With the choose of $\zeta = 1.0$, simple relation holds for feedback gains in:

$$K_V = 2\sqrt{K_P} \qquad (14)$$

MEASURED RESULTS

In order to compare the performance of three different inner loop force/pressure controller in force based position, experiments were carried out in 1 DOF linear testbed. The specifications for the experimental system are as follows

- Cylinder 40/28-710, load mass was 250 kg.
- Flow control servovalve: Bosch-Rexroth closed loop proportional valve NG6, QN = 40 l/min at delta_p = 35 bar per notch. Natural frequency about 100 Hz, when input amplitude is +/- 5 %.
- Pressure control servovalve: Moog series 15-010. Rated flow capacity of 55 l/min at 105 bar per notch. Block port natural frequency is about 250 Hz.
- Pressure sensors: Kistler K-line 300bar
- Supply pressure is 70bar and maximum flow is enough so that valves doesn't saturate

Position, velocity and acceleration motion profiles for point to point motions are required to be smooth functions of time. Therefore, a quintic polynomial is used for desired position profile in measurements. Amplitude of the motion profile was 3.5 cm and a rise was time 0.5 seconds.

In Figures 5 and 6, the results with pressure control servovalve without inner feedback loop force controller are shown. In experimental system, stability limit for upper-loop position controller was found to be 6500 and a value used for it was 6000.

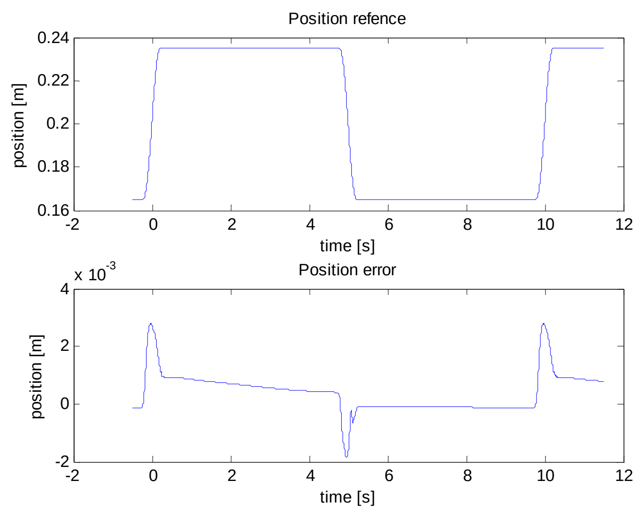

Figure 5 Position control response with pressure control servovalve

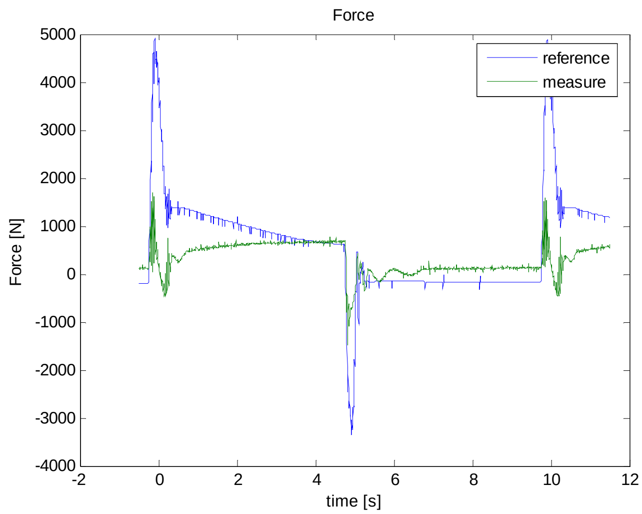

Figure 6 Force control responses with a pressure control servovalve

Next Figures 7 and 8 show results with a pressure control servovalve with the inner-loop force controller. For this experiment position controller gain was reduced to 3000 and inner-loop force control gain was set to 0.22. Stability limit for inner loop control was found to be 0.26.

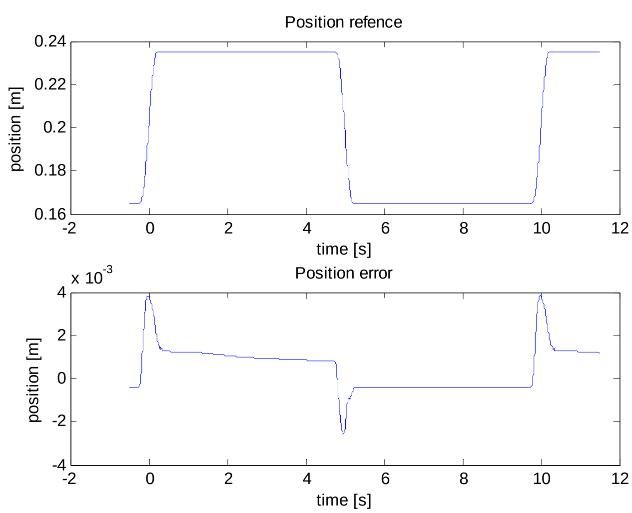

Figure 7 Position control response of a pressure control servovalve

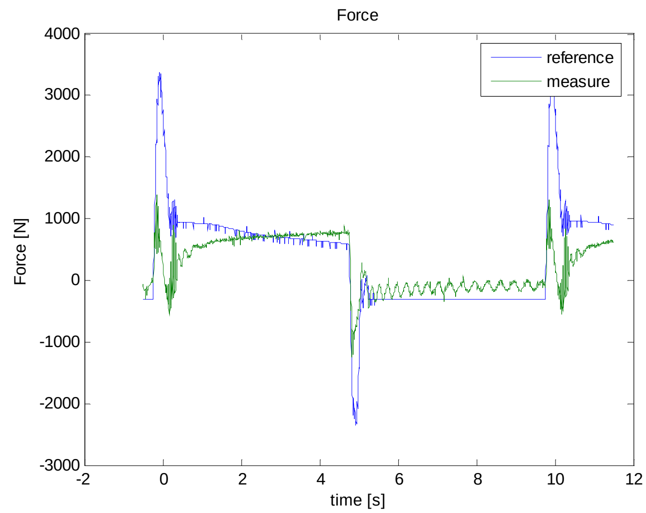

Figure 8 Force control response with a pressure control servovalve

Finally, Figures 9 and 10 show, a flow control servovalve responses, with inner loop force control gain 0.00004 when its stability limit was 0.0001. In this case, outer loop's position controller gain was 28000, when its stability limit was 30000.

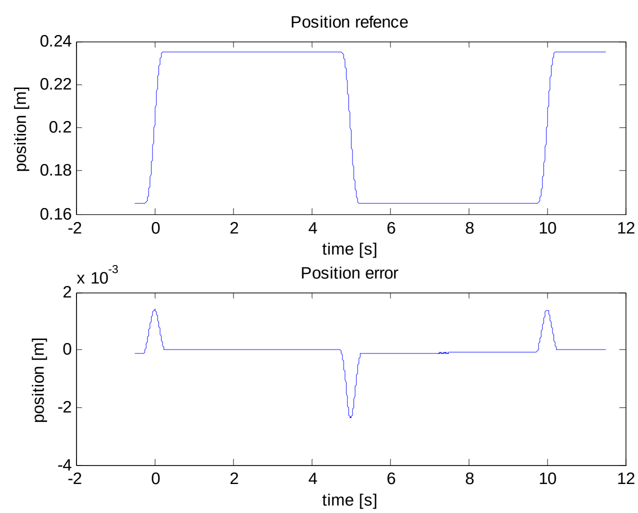

Figure 9 Position control response of a flow servovalve

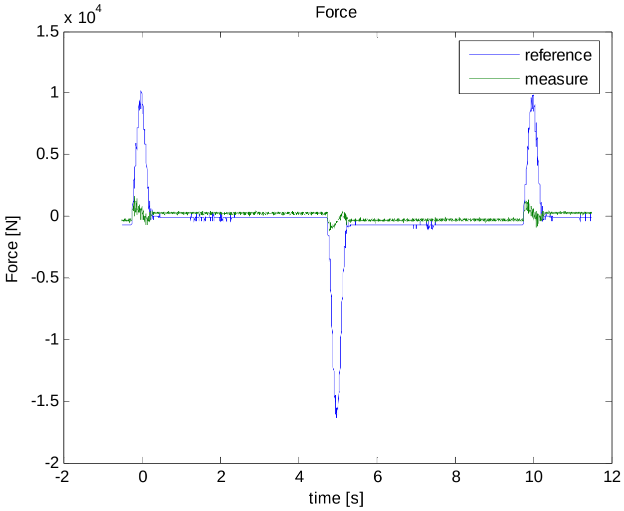

Figure 10 Force response of a flow servovalve

As we can see from measured results, flow servovalve's position tracking error is the lowest. Also static position accuracy of the flow servovalve is very good and it is equal to both motion directions. In both control cases with pressure control servovalve, the static accuracy depends on motion directions. Most likely, a symmetric cylinder is a cause of that. For the pressure control servovalve, as can be seen from the Figures 5 and 7, the motion settling time is very long and depends on motion direction. In typical manipulator application even with symmetric actuators (e.g. vane actuators), pressure valve offsets can be a problem due to a different load force direction. As expected, without a contact with the environment phase of the motion, inner-loop force controller doesn't improve the performance of the pressure control servovalve. Therefore, series of additional tests have to be carried out in the future with a realistic combination of both free space and contact phase motions.

Comparing force tracking capabilities, it can be noticed that flow servovalve is quite poor in dynamic force tracking but the static force accuracy is good. With pressure control servovalve, the settling time of force response is very slow while valve is correcting small position error as can be seen from Figure 6 and 8.

CONCLUSIONS

Pressure servovalve works quite well in force-based position control, and they can be considered for teleoperation applications in harsh environments. Pressure control servovalves are easy to tune by few tuning rules. Flow servovalves also provides good force-based position control results. However, with flow control servovalves, significant amount of effort has to put into careful selection of pressure transducers, sensor signal conditioning and to robust tuning of the inner-loop force controller's etc.

The best properties of the pressure control servovalves are that they are easy to tune and have very robust design. The concern about using pressure servovalves are their price and availability in the near future. A pressure control servovalve used in is this study is manufactured in accordance with MIL-speciation and therefore should be reliable and proven technology. However, some of the other valve manufacturers have already discontinued their pressure control servovalve program.

In near future, more tests are needed to be carried out in a rotary test joint subject to non-constant gravity force and contact with the compliant environment.

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