APPROXIMATE VELOCITY FEEDBACK USING HYDRAULIC VALVE FLOW MODELLING

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ABSTRACT

In this paper the cross sectional area of a hydraulic servo valve is modelled as a non-linear function of the spool position. The actual leakage flow between the work port and the tank as well as between the work port and the supply port is measured. The leakage flow is modelled based on an idea of a separate pressure drop factor for the flow through the notches and the flow in the gap. The actual leakage flow to the tank is subtracted from the measured flow and the actual leakage to the work port is added to it. The discharge coefficient in the theoretical turbulent flow equation is fitted to the remaining flow curve and estimated volume flow is used as a velocity feedback in a hydraulic cylinder velocity servo.

KEYWORDS

Velocity feedback, valve model, flow measurement,

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tr>
<td>A_i</td>
<td>Area i</td>
<td>m^2</td>
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<tr>
<td>A</td>
<td>Constant</td>
<td>kg/sm^2</td>
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<tr>
<td>B</td>
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<td>d/dt</td>
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<td>F_i</td>
<td>Force i</td>
<td>N</td>
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<tr>
<td>MToff</td>
<td>piston-to-tank notch offset</td>
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<td>MToff</td>
<td>supply-to-piston notch offset</td>
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<tr>
<td>pi</td>
<td>Pressure i</td>
<td>Pa</td>
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<td>U</td>
<td>Valve control voltage</td>
<td>V</td>
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<tr>
<td>V_i</td>
<td>Volume i</td>
<td>m^3</td>
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<tr>
<td>v</td>
<td>velocity</td>
<td>m/s</td>
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<tr>
<td>VPoff</td>
<td>supply-to-rod notch offset</td>
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<tr>
<td>VToff</td>
<td>rod-to-tank notch offset</td>
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INTRODUCTION

A velocity feedback is often used to improve the dynamic behaviour of servo systems, especially in hydraulic position servo systems. In these applications the quality and accuracy of the velocity measurements are not so critical. The requirements are much tighter in actual velocity servo systems. In rotary motion applications velocity measurements can be quite easily realized. However, linear motions are very common in hydraulics. In general the velocity measurement is somewhat complicated in linear motions, especially in long linear motions. In harsh environments the specifications of linear velocity and position sensors are very demanding. In hydraulics it is quite common to install a position sensor inside a piston rod. In these cases the problem is that the resolution of the position sensor is quite poor. It causes the quality of the velocity estimated from the position signal to also be quite poor.

In the measurement of a position with an analog sensor the voltage range is typically 10 V. With A 12 bit AD- converter this means that in a stroke of 1 m the resolution of 2.44 mV corresponds to the resolution of 0.244 mm. With a 100 Hz sampling frequency this means that the lowest velocity that can be measured with the accuracy of 5% is 244 mm/s.

As the authors have previously concluded, the velocity measurement in velocity servo applications can be implemented as an orifice flow meter [2,4] while the control notch of a servo valve acts as a variable orifice. The volume flow through the valve control notch is calculated from the relative spool position and pressure difference over the notch. That way 1:100 flow rates can be achieved with no additional pressure loss and a very little extra space and with no extra cost. In previous studies the absolute error of piston velocity measurements was approximately 15% at 5% velocity. To further improve the accuracy it is necessary to know the valve leakage behaviour in detail. In this case the piston maximum velocity outward is 1.3 m/s, so the velocity of 73 mm/s means 5.6 % resolution in an analog measurement. The variable orifice flow measurement has the potential to achieve 1% resolution, relative accuracy of 1% and absolute accuracy of 10%.

Since the valve dimensions depend on manufacturing tolerances and are not exactly known, any constants in accurate laminar flow equations are replaced with one single constant and values for these constants are determined by the measurements. Mathematical description is very simple and it should basically be valid for any same kind of valve when a few calibration measurements are carried out.

VALVE FLOW MODEL

When a commercial hydraulic valve is used, the dimensions of a spool and valve are not exactly known. A valve may appear symmetric but valve control notches always have individual offsets even when a valve has basically zero overlaps. The eccentricity of a spool is also an unknown variable as well as a backlash. The basic turbulent flow equation poorly matches the actual flow in the ±10% flow rate range. The mismatch is smaller at higher flow rates but some variations in the effective notch area still exist. As [5] shows, the discharge coefficient varies as a function of the spool position. The measuring and modelling of the leakage flow and other non-linearities of the valve is one way to improve the velocity estimation.

The leakage in each valve control notch was measured separately. The work port was connected to the supply pressure and the supply and the tank ports were connected to the atmosphere pressure. In each measurement the spool positions were varied from −10 to +10 percent with 0.5% intervals. While measuring one work port, the other work port was plugged. In the leakage range the pressure drop in the gap between the spool and housing was divided in two parts according to [1]. Leakage flow model does not seem to follow the square interdependence in sudden change as well as Ellman found out in his study on the water medium.

The valve spool manufacture can be assumed to achieve similar edges on each spool, and thus a leakage flow model curve fitting one edge should at least be closely valid for another edge, too. This was not the case with the kind of interdependency that Ellman used. When the pressure loss in a sudden change in cross section was modelled to depend linearly on flow, the curve fitting constant for one edge was equally valid for another. The leakage curves (spool in center position) provided by the valve manufacturer back up this theory too. When the control notch is just about to open, the pressure drop in the notch reaches its minimum value that does not depend on the spool position. When the spool is moved to close the notch, the pressure drop over the notch increases according to the laminar flow equation, which yields to the Eq. (1).

$$Q_{L_i} = \frac{\Delta P_i}{A + Bc_i}$$

(1)

The subscript i refers to the leakage of the specific control notch. The factors A and B are tuned to match the measured and modelled leakage flows. The leakage flows from the work port to the supply port as well as
from the work port to the tank port were measured. The results are shown in the same plot, Fig.(1). Fig. (2) shows the modelled and measured leakages of port B. Port A is modelled the same way.

The offsets of valve control notches were determined from the leakage flow and pressure gain measurements. First the ± limits of the spool positions when the cylinder was still steady were determined. Then the work ports were switched to the different sides of the cylinder and again the spool position limits were determined. This way the static friction force was found and the middle point of the spool position variation range in these measurements was determined to the centre position of the valve. This method resulted in the 0.12 % valve offset. Fig. 3 shows the measured pressure gain.

The following method was used to find out the offsets of each control notch. First the leakages from port P to port A and from port B to port T were determined at -1.3% control point. The leakages were scaled to the 100 bar pressure difference. Then these flows were inserted into the Eq. (1) and the resulting pressure difference was compared to the pressure gain curve at the same control signal. The difference in these two pressure differences was divided equally for both chamber pressures, and corresponding control signal values were calculated from Eq. (1). The difference between these values and the original -1.3% control signal was determined for the control notch offset. The offsets from port P to port B and from port A to port T were determined the same way. Furthermore, the offsets were determined to match the static and dynamic pressure behaviour with the simulation model too. Table 1 shows the results of these determination methods.

<table>
<thead>
<tr>
<th>Table 1. Valve offset determination results</th>
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<tr>
<td>Measurement</td>
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<tr>
<td>MPoff</td>
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<td>MToff</td>
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<tr>
<td>VPoff</td>
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<tr>
<td>VToff</td>
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<tr>
<td>Valve off</td>
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After the offsets of the control notches were found the effective relative spool position was calculated separately for each control notch. The actual flow was found by measuring the cylinder velocity. In the spool position range where the control notch was closed, the measured leakage flow to the tank was subtracted and the leakage flow from the supply was added to the measured flow, Eq. (2).

\[ Q_C = Q_{\text{meas}} + Q_{l,s} - Q_{l,t} \]  

Alternately, when the control notch of the spool was open, the leakage part was set to zero and in the turbulent flow Eq. (3) [3] the effective notch area was replaced with the approximated notch area, Eq. (4).

\[ Q = C_d A \sqrt{\frac{2(p_s - p)}{\rho}} \]  

\[ A_{ef} = C_d A \sqrt{\frac{2}{\rho}} = f(u) \frac{Q}{\sqrt{\Delta p_N}} \]  

Which can be written in more practical form, Eq. (5).

\[ F_{eq} = Q_N \frac{\Delta p}{\sqrt{\Delta p_N}} \]  

This effective notch area function \( f(u) \) was then fitted with third order polynomial to meet Eq. (6).

\[ f(u) = \frac{Q_r}{F_{eq}} \]  

The estimated volume flow is as follows

\[ Q_{est} = F_{eq} \cdot f(u) \]  

In the dynamic state the compressibility flow was subtracted from the estimated volume flow and the velocity estimation was achieved.

\[ v_{est,i} = \frac{Q_{est,i} - dp_i}{dtB} \frac{V_{0,i}}{A_i} \]  

subscript \( i \) refers to a specific cylinder chamber.

The actual valve spool position was measured, thus avoiding the modelling of the flow forces and coil hysteresis. The estimated velocity was calculated from the piston side that had increasing volume. Fig. 4 shows the measured and calculated flow open loop responses with different control inputs, fig. 5 shows the measured and calculated flow step responses.

\[ Q_{est} = F_{eq} \cdot f(u) \]  

In the test installation work port pressures, tank pressure and supply pressure were measured. In the calculations the pressure drop from the supply pressure to the corresponding cylinder chamber pressure was used. It was necessary to calibrate the pressure sensors, as the calculated pressure force is quite sensitive to the actual cylinder chamber pressures.
Since the flow evaluation accuracy was poor in the valve control signal range that is less than 1%, some instability at low velocities might occur. The friction force Eq. (9) was used to set the velocity to zero and to determine the motion direction.

\[ F_N = |p_1 A_1 - p_2 A_2| - F_s \]  

(9)

In practice this means that with an inertia load of 225 kg the valve opening is about 0.2% which is less than one third of the controller output offset needed to keep the cylinder steady when the velocity reference is zero. In closed loop velocity servo system tests the inertia load of 120 kg was used. The velocity servo was tuned to the lowest natural frequency point. Fig. 6 shows the velocity responses to the step inputs and also the corresponding differentiated velocity.

**SUMMARY**

When the control notches of a servo valve are used as a measuring orifice, very accurate flow measurement can be achieved over a wide working range. The accurate leakage model and offset determination of individual control notches provide many possibilities to apply this proposed volume flow estimation as a velocity feedback in many kinds of control applications. The basic limitations of most flow measurement methods are avoided, but the calibration procedure is more complicated. The main drawbacks are: the leakages and offsets of control notches of a valve need measuring the supply and tank pressures have to be quite constant, and a highly complicated calculation procedure is needed.

The aim of this study was to construct a universal leakage and flow model for single stage servo valves. The model was verified with open loop velocity measurements. The model seems to work well. Some tests have also been done when the proposed velocity estimation has been used in the feedback loop of velocity and position servos. Results are very promising, but some further studies are needed. More tests and applications are needed to find out the usefulness of the proposed velocity estimation method. The following questions have arisen: What are the required measurements to guarantee the needed accuracy of the leakage flows in the valve control notches? What are the number of required pressure sensors and the their calibration requirements? How much do different valves differ from each other in modeling parameters? How valid is the model in different dynamic situations? What kinds of applications suit the proposed method best?

**REFERENCES**


