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PERFORMANCE LIMITATIONS OF A CLASS OF TWO-STAGE ELECTRO-HYDRAULIC FLOW VALVES

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Abstract

By examining the dynamics of a popular type of two-stage electronic proportional valve, this paper addresses its performance limitations, with both cautions in control implementation and suggestions in valve design. While several benefits do exist, there are limitations to the closed loop performance of the valve when it is included in a valve-controlled electro-hydraulic system. These limitations come from the structural feature that the pilot flow not only controls but also contributes to the total flow. Although for steady state performance this design gives a higher flow efficiency, for dynamic performance it results in zeros in the open loop transfer function, which will limit the closed loop bandwidth of a flow control system. A non-linear analytical model of this particular type of valve is derived first. It is then simplified as a reduced order linear model with the inherit system zeros illustrated. Validation of the analysis is obtained by experimental results on a testbed.

Keywords: electro-hydraulic, valve, valvistor, zero, model

1 Introduction

The trend in hydraulic power applications is to improve efficiency and performance. These improvements can often be achieved using feedback controllers in an electro-hydraulic (E/H) framework. However, typical single stage solenoid valves used in these E/H systems often suffer from two problems. Firstly, since the actuation of the valve solely relies on the electro-magnetic force, the dimension and energy consumption of the solenoid increases significantly when designing a valve for a larger flow. Secondly, since flow forces vary as a function of pressure drop, they act as a strong disturbance to the overall control loop. The common solution to these two problems is the introduction of a pilot stage and a position feedback. In a two-stage valve, the main stage is actuated by a hydraulic force instead of an electromagnetic one. The hydraulic force generated from a pressure difference is regulated by the pilot stage, a much smaller electro-hydraulic valve, which reduces manufacturing and operating cost. To improve disturbance rejection, one method is to utilize an inner-loop electronic feedback on valve position. Another is to use a mechanical inner-loop feedback typical on flapper-nozzle type servo valves. However, both options can be economically infeasible due to their high cost. To provide a solution at lower cost, a particular type of two-stage valve developed by Vickers Inc., termed a “Valvistor”, has an embedded internal hydraulic feedback that provides an efficient, flow-force compensated, proportional flow valve. The valve uses a small pilot circuit to drive the larger main flow, similar to the behavior of an electronic transistor; hence the name. Other advantages to this type of valve include a fast response, repeatability, and low hysteresis. As a result of these benefits, the “Valvistor” has seen increased popularity. Evidence of this is its feature as one of the “Reader’s Choice Top 10 Products of 1999” as given in the SAE Off-Highway Engineering Magazine (1999).

Although the “Valvistor” is a popular industrial product, there is relatively little available in the open literature on the modelling and/or analysis of this component. Particularly, while the manufacturer may provide steady state characteristics, little work has been done in the dynamic aspect. In one of the few available works, (Palmberg and Petterson, 2000) presented the concept of a “Valvistor” flow amplifier in a particular flow control system using combination of two or more valves. The focus of (Palmberg and Petterson, 2000) is to maintain a high flow gain and to increase the bandwidth of the valve combination system. Modelling and

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analysis of the traditional servo or proportional flow valve has long been available in the open literature. (Nikiforuk and Ukrainetz et al, 1969; de Pennington, 't Mannetje et al, 1974; Watton, 1989) are only a few of the many available works that describe in detail the process to derive a classical linear model for a commercially available electro-hydraulic flow valve. More recently, Zavarehi et al demonstrated an extension of this modelling approach by taking into account nonlinearities of the system (Zavarevi and Lawrence et al, 1999). In this paper, the linear approach used by researchers such as (Nikiforuk and Ukrainetz et al, 1969; de Pennington and 't Mannetje et al, 1974; Watton, 1989) will be used to model and analyze the "Valvistor" in order to illustrate the inherent performance limitations imposed by system zeros.

2 System Modelling

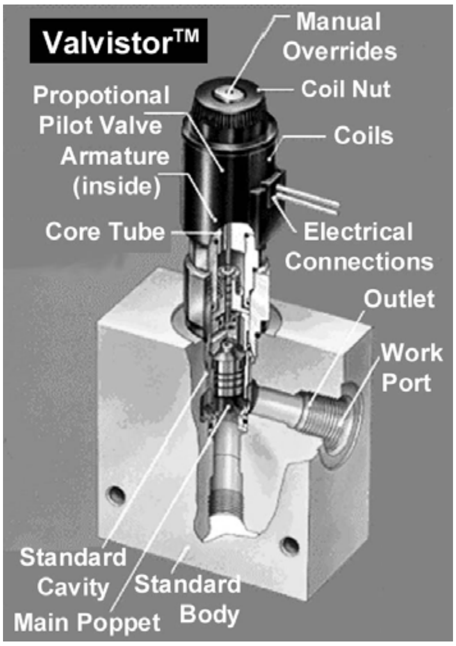


Fig. 1: *A Vickers EPV-16 cartridge type Valvistor*

The valve under investigation is a Vickers Electronic Proportional Valvistor EPV-16 (Vickers, 1994), which is shown in Fig. 1. A schematic diagram is shown in Fig. 2. Using a voltage controlled pulse-width-modulated (PWM) signal, the pilot valve regulates a relatively small flow in the pilot circuit. The pilot flow Q_p generates a pressure difference (P_a–P_p) across the main poppet m_m, and consequently causes it to move. The main poppet position x_m determines the opening of the main orifice and the main flow rate Q_1 from the inlet *a* to the outlet *b*. The movement also changes a variable orifice of the longitudinal slot on the poppet, which acts as a feedback of the main poppet position x_m to affect the pilot pressure P_p. The use of a small pilot flow to control a large main flow is analogous to a transistor. Notably, the pilot flow is routed so as to converge with the main flow that increases the steady state efficiency; this at the same time affects the dynamic performance as to be explained in later sections. The valves used in this study are the flow control components of an Earthmoving Vehicle Powertrain Simulator (EVPS), an electro-hydraulic testbed developed at the University of Illinois at Urbana-Champaign (Prasetiawan, Zhang et al, 1999; Prasetiawan, 2000; Prasetiawan, Zhang et al, 2000).

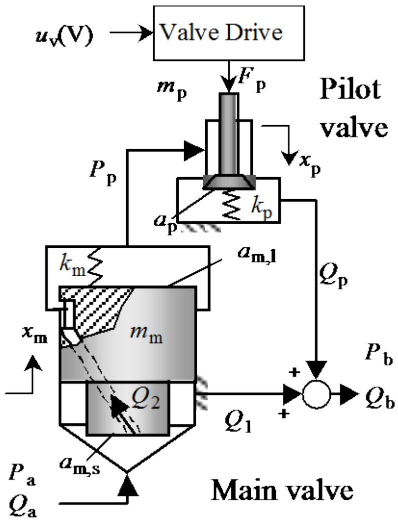
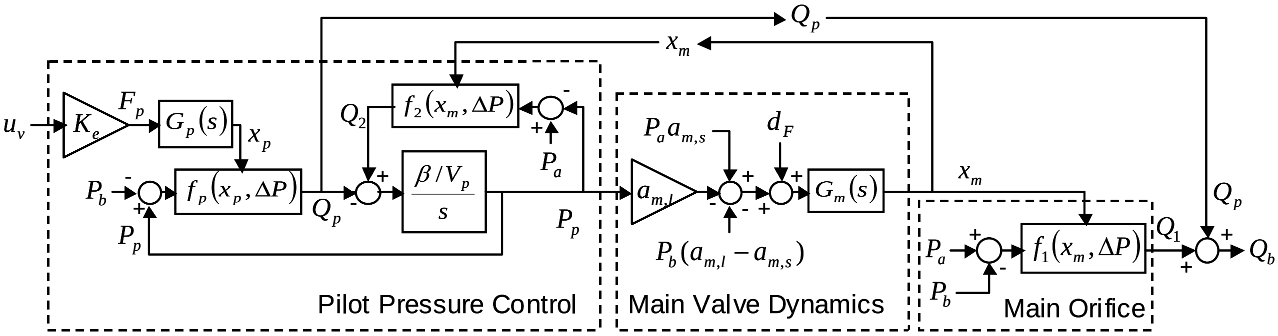


Fig. 2: *Electro-proportional flow valve schematic*



K_e: Solenoid gain
G_p: Pilot valve dynamics
f_p: Pilot orifice
f_2: Slot orifice
G_m: Main valve dynamics
d_F: Disturbance force
f_1: Main orifice

Fig. 3: *Block diagram of a non-linear model of the two-stage flow valve*

The valve can be modelled as two sets of mass-spring-damper systems and a compressible fluid volume in between as shown in Fig. 2. It consists of 5 main states: 2 from the pilot valve, 1 from the compressibility of fluid between two stages, and 2 from the main poppet. The entire system can be divided into three functional subsystems: (1) the pilot pressure control, (2) the main valve dynamics, and (3) the main orifice flow.

The pilot pressure is controlled by the pilot valve dynamics, the pilot orifice flow, the slot orifice flow, and the inter-stage fluid compressibility. When an input voltage u_v is applied, the pilot valve moves in accordance with a force balance equation:

$$m_p \ddot{x}_p + b_p \dot{x}_p + k_p x_p = F_p + (P_p - P_b)\, a_p \tag{1}$$

where

$$F_p = K_e u_v \tag{2}$$

Equation 2 assumes that the solenoid dynamics are fast enough to be neglected. The pressure difference across the orifice and the volumetric change rate $a_p \dot{x}_p$ due to the pilot valve movement both contribute to the pilot flow Q_p:

$$Q_p = K_p x_p \sqrt{P_p - P_b} + a_p \dot{x}_p \approx K_p x_p \sqrt{P_p - P_b} \tag{3}$$

Since the volumetric change is generally insignificant compared to the turbulent flow across the orifice, the term $a_p \dot{x}_p$ can be neglected to simplify the expression of pilot flow Q_p.

Whenever the main valve opens, the channel inside the poppet and the longitudinal slot on the side surface of it also creates a bypass flow Q_2 from the valve inlet up to the pilot stage. The opening of the orifice is variable, which is determined by the main poppet position x_m:

$$Q_2 = K_s x_m \sqrt{P_a - P_p} \tag{4}$$

For the lumped volume between the main valve and the pilot valve, the pressure P_p changes at a rate described by the net flow into the volume:

$$\dot{P}_p = \frac{\beta}{V_p}\left(Q_2 + a_{m,l}\dot{x}_m - Q_p\right) \approx \frac{\beta}{V_p}\left(Q_2 - Q_p\right) \tag{5}$$

Similar to Eq. 3, the volumetric term in Eq. 5 accounts for a change in the pilot circuit volume due to the main poppet valve movement. It can also be ignored since it is reasonably small compared to Q_p and Q_2 .

Secondly, the main valve dynamics determine the main poppet movement x_m under the force of the pilot pressure P_p , of the inlet pressure P_a, and the flow force d_F as a disturbance:

$$\begin{aligned} & m_m \ddot{x}_m + b_m \dot{x}_m + k_m x_m \\ & = a_{m,s} P_a - a_{m,l} P_p + (a_{m,l} - a_{m,s}) P_b + d_F \end{aligned} \tag{6}$$

Finally, the flow through the main orifice is determined by the opening created by x_m:

$$Q_1 = K_m x_m \sqrt{P_a - P_b} \tag{7}$$

The overall system including the above three subsystems is presented as a block diagram in Fig. 3. Although the subsystems are basically connected in series, there are two additional interconnections.

One of the interconnections is the feedback of the main poppet position x_m from the main valve dynamics to the pilot pressure control. For the 2nd order main valve dynamics, the force difference resulting from P_a and P_p is the controlled input. At the same time, the position output x_m is also affected by some disturbance d_F such as the flow force. When the main valve is pushed up by some unexpected disturbance, the pressure drop (P_a–P_p) decreases since the slot opens more, and this counteracts the disturbance. In other words, the slot serves as a negative feedback to decrease the sensitivity of the output position x_m to the disturbance d_F.

The other additional interconnection is the feed-forward of the pilot flow Q_p, which adds to the total flow Q_b:

$$Q_b = Q_1 + Q_p \tag{8}$$

Let

$$\begin{aligned} \begin{bmatrix} x_1 & x_2 & x_3 & x_4 & x_5 \end{bmatrix}^T &= \begin{bmatrix} x_m & \dot{x}_m & P_p & x_p & \dot{x}_p \end{bmatrix}^T \\ y &= Q_b \end{aligned} \tag{9}$$

The overall system model can now be given as:

$$\dot{x} = \begin{bmatrix} x_2 \\ -\frac{k_m}{m_m}x_1 - \frac{b_m}{m_m}x_2 + \frac{a_{m,s}P_a + (a_{m,l} - a_{m,s})P_b - a_{m,l}x_3}{m_m} \\ \frac{\beta}{V_p}\left(K_s x_1 \sqrt{P_a - x_3} - K_p x_4 \sqrt{x_3 - P_b}\right) \\ x_5 \\ \frac{a_p}{m_p}(x_3 - P_b) - \frac{k_p}{m_p}x_4 - \frac{b_p}{m_p}x_5 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ K_e \end{bmatrix} u_v \tag{10}$$

$$y = K_p x_4 \sqrt{x_3 - P_b} + K_m x_1 \sqrt{P_a - P_b} \tag{11}$$

3 Simplified Linear Model

Although it might be necessary to examine the 5-state non-linear model in detailed analysis and designs, a linear simplification is sufficient to represent the dominant dynamics and to reveal the performance limitation.

To reduce the order and simplify the 5-state model in Eq. 10, two additional assumptions are made: (i) the transients of the pilot poppet and the pilot pressure are sufficiently fast compared to the dominant dynamics of the main valve; (ii) the spring in the pilot valve is sufficiently strong such that the pressure disturbance to the poppet position has an ignorable effect. These assumptions can be expressed as:

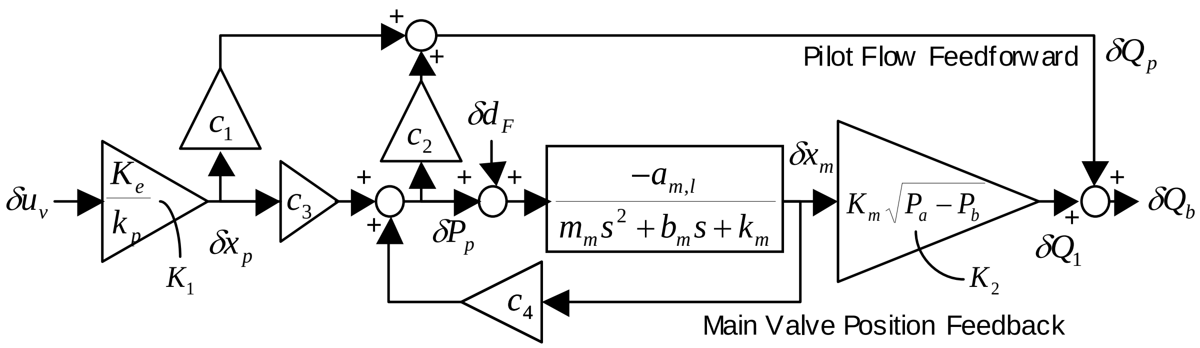


Fig. 4: *Simplified linear model of a two-stage electro-proportional flow valve*

$$\begin{aligned} &\dot{P}_\mathrm{p} = \dot{x}_3 = 0; \\ &\ddot{x}_\mathrm{p} = \dot{x}_5 = 0; \\ &\dot{x}_\mathrm{p} = \dot{x}_4 = 0; \\ &a_\mathrm{p}(x_3 - P_\mathrm{b}) << k_\mathrm{p} x_4 \end{aligned} \qquad (12)$$

It is expressed by the first three equalities in Eq. 12 that the compressibility flow, the inertia force and the viscous force in the pilot stage are negligible. With these assumptions, Eq. 10 is simplified to:

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ 0 \\ 0 \end{bmatrix} = \begin{bmatrix} x_2 \\ -\frac{k_\mathrm{m}}{m_\mathrm{m}} x_1 - \frac{b_\mathrm{m}}{m_\mathrm{m}} x_2 + \frac{a_\mathrm{m,s} P_\mathrm{a} + (a_\mathrm{m,l} - a_\mathrm{m,s}) P_\mathrm{b} - a_\mathrm{m,l} x_3}{m_\mathrm{m}} \\ K_\mathrm{S} x_1 \sqrt{P_\mathrm{a} - x_3} - K_\mathrm{p} x_4 \sqrt{x_3 - P_\mathrm{b}} \\ -k_\mathrm{p} x_4 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ K_\mathrm{e} \end{bmatrix} u_\mathrm{v} \qquad (13)$$

To linearize this simplified model, let

$$x_\mathrm{i} = x_{\mathrm{i,o}} + \delta x_\mathrm{i}, \quad u_\mathrm{v} = u_\mathrm{v,o} + \delta u_\mathrm{v}, \quad y = y_\mathrm{o} + \delta y \qquad (14)$$

A linearized model is obtained about the equilibrium as:

$$\begin{bmatrix} \delta \dot{x}_1 \\ \delta \dot{x}_2 \\ 0 \\ 0 \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\frac{k_\mathrm{m}}{m_\mathrm{m}} & -\frac{b_\mathrm{m}}{m_\mathrm{m}} & -\frac{a_\mathrm{m,l}}{m_\mathrm{m}} & 0 \\ c_4 & 0 & -1 & c_3 \\ 0 & 0 & 0 & -1 \end{bmatrix} \begin{bmatrix} \delta x_1 \\ \delta x_2 \\ \delta x_3 \\ \delta x_4 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ \frac{K_\mathrm{e}}{k_\mathrm{p}} \end{bmatrix} \delta u_\mathrm{v} \qquad (15)$$

$$\delta y = K_\mathrm{m} \sqrt{P_\mathrm{a} - P_\mathrm{b}}\, \delta x_1 + c_2 \delta x_3 + c_1 \delta x_4 = \delta Q_1 + \delta Q_\mathrm{p} \qquad (16)$$

The coefficients c_1 and c_2 for the pilot flow are:

$$\begin{aligned} c_1 &= K_\mathrm{p} \sqrt{x_{3,\mathrm{o}} - P_\mathrm{b}} = K_\mathrm{p} \sqrt{P_{\mathrm{p,o}} - P_\mathrm{b}} \\ c_2 &= \frac{K_\mathrm{p} x_{4,\mathrm{o}}}{2\sqrt{x_{3,\mathrm{o}} - P_\mathrm{b}}} = \frac{K_\mathrm{p} K_\mathrm{e} u_{\mathrm{v,o}}}{2 k_\mathrm{p} \sqrt{P_{\mathrm{p,o}} - P_\mathrm{b}}}, \end{aligned} \qquad (17)$$

where $P_{\mathrm{p,o}}$ is the equilibrium value of P_p. The coefficients c_3 and c_4 for the pilot pressure are:

$$\begin{aligned} c_3 &= -\frac{2K_\mathrm{p}^2 (x_{3,\mathrm{o}} - P_\mathrm{b}) x_{4,\mathrm{o}}}{K_\mathrm{p}^2 x_{4,\mathrm{o}}^2 + K_\mathrm{S} x_{1,\mathrm{o}}^2} = -\frac{(P_\mathrm{a} - P_{\mathrm{p,o}})(P_{\mathrm{p,o}} - P_\mathrm{b})}{x_{\mathrm{p,o}}(P_\mathrm{a} - P_\mathrm{b})} \\ c_4 &= \frac{2K_\mathrm{p}^2 (P_\mathrm{a} - x_{3,\mathrm{o}}) x_{1,\mathrm{o}}}{K_\mathrm{p}^2 x_{4,\mathrm{o}}^2 + K_\mathrm{S} x_{1,\mathrm{o}}^2} = \frac{(P_\mathrm{a} - P_{\mathrm{p,o}})(P_{\mathrm{p,o}} - P_\mathrm{b})}{x_{\mathrm{m,o}}(P_\mathrm{a} - P_\mathrm{b})} \end{aligned} \qquad (18)$$

The block diagram of the simplified linear model is presented in Fig. 4.

Define $K_1 = K_\mathrm{e}/k_\mathrm{p}$, $K_2 = K_\mathrm{m}\sqrt{P_\mathrm{a} - P_\mathrm{b}}$, the valve transfer function $G_\mathrm{v}(s)$ with two zeros is derived as:

$$\begin{aligned} G_\mathrm{v}(s) &= \frac{\delta Q_\mathrm{b}(s)}{\delta u_\mathrm{v}(s)} \\ &= K_1 \frac{(c_1 + c_2 c_3)(m_\mathrm{m} s^2 + b_\mathrm{m} s + k_\mathrm{m}) + a_\mathrm{m,l}(-K_2 c_3 + c_1 c_4)}{m_\mathrm{m} s^2 + b_\mathrm{m} s + k_\mathrm{m} + a_\mathrm{m,l} c_4} \end{aligned} \qquad (19)$$

In summary, the 5-state valve model is simplified as a second-order system by ignoring the fast dynamics of both the pilot stage and the fluid compressibility between stages. The dynamics primarily come from the main poppet valve. The internal position feedback decreases the disturbance sensitivity and compensates for flow forces. The pilot flow that is used to control the main poppet valve is combined with the main flow for steady state efficiency, which introduces two zeros due to the feed forward path from δQ_p to δQ_b.

4 Model Validation

To validate the model, experiments were performed to obtain frequency and time domain responses. The frequency response of the system is obtained by applying a sinusoidal input to the valve around some nominal input value at a frequency range of 0.1-60 Hz. This is done while the valve is part of the electro-hydraulic system in Fig. 5. The flow source into the valve is from a variable-displacement electro-hydraulic pump. The flow out of the valve is used to drive a fixed-displacement gear motor (4.216 cm^3/rad), which is loaded by a closed-coupled gear pump under a controlled pressure. Therefore, the “Valvistor” controls the speed of the gear motor. The motor speed is picked up by a tachometer and the actual valve flow Q_b is inferred by multiplying the motor’s displacement per revolution. In this scenario, we have neglected leakage across the motor. The frequency domain dynamics of the hydraulic hose and gear motor-pump couple were previously isolated and found to be a second-order system (Prasetiawan, 2000) by using a fast variable displacement pump to generate the sinusoidal flow input. Considering this 2nd order dynamics, the flow valve dynamics can be separated out of the overall frequency response from the valve input to the motor speed output. In time domain validations, the motor speed response to a square wave valve input was compared with the overall simulation including the valve model and the motor model.

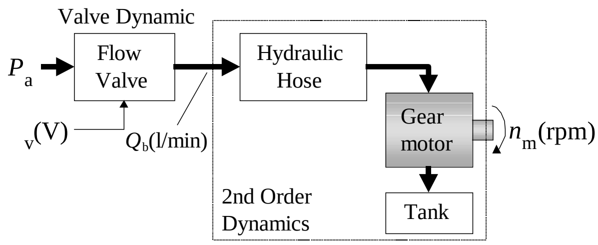


Fig. 5: *Flow valve identification test setup*

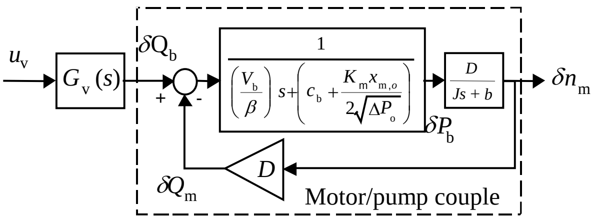


Fig. 6: *Overall system block diagram*

In the model derivation, the inlet and outlet pressure changes are not considered. Readings from the pressure sensors available on the EVPS indicated that during the frequency domain experiments the downstream pressure P_b changes with flow variations. As a result, the dynamics of the downstream pressure should be included in the model analysis to improve accuracy, however constant pressure assumptions serve our purpose for illustrating the existence of system zeros. The linearized version of the valve's total output flow is:

$$\delta Q_b = K_m \sqrt{\Delta P_o}\,\delta x_m - \frac{K_m x_{m,o}}{2\sqrt{\Delta P_o}}\delta P_b + \delta Q_p \, , \qquad (20)$$

where $\Delta P_o = P_a - P_{b,o}$. The flow decreasing effect of δP_b can be regarded as a leakage equivalence in the hose/motor system, which is illustrated in Fig. 6. At the same time, although δP_b also affects the main poppet position, it does so only on a small annulus area (approximately 25% of $a_{m,l}$). Compared to the inlet pressure and the pilot pressure acting on larger areas, the force from δP_b can be neglected since it is small enough and it does not change the qualitative structure of the valve model.

The experimental frequency response data is taken at the operating condition with the flow valve input being 3.5 ± 0.3 V out of a (0, 10 V) range. At this operating condition, the valve is capable of modulating the motor speed effectively, however the range of linearity is limited from 3.0 to 4.0 V as shown in the steadystate valve mapping found in (Prasetiawan, Zhang et al, 2001). The obtained frequency response from valve input to the motor speed output is fit to a fourth-order system with two zeros. The fit is given as:

$$\frac{\delta n_m}{\delta u_v} = 3.609 \cdot 10^5 \frac{(s+1.25)(s+11.5)}{(s+28+61j)(s+28-61j)(s+1)(s+30)} \left[\frac{\text{rad/s}}{\text{V}}\right] \qquad (21)$$

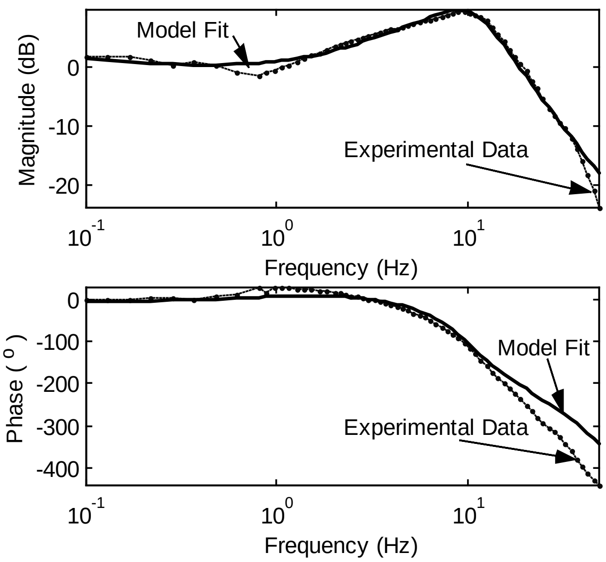


Fig. 7: *Comparison of experimental data and model fit of $n_m(s)/u_v(s)$ transfer function*

The quality of this model fit in the frequency domain can be seen in Fig. 7. Since the second-order dynamics of the hydraulic hoses and the coupled motor-pump have been previously identified to have two complex conjugate poles in Eq. 21, the valve transfer function can be readily determined to be:

$$\frac{\delta Q_v(s)}{\delta u_v(s)} = K_v \frac{(s+1.25)(s+11.5)}{(s+1)(s+30)} \left[\frac{\text{l/min}}{\text{V}}\right] \qquad (22)$$

This transfer function qualitatively verifies the model structure with two zeros in Eq. 19, where K_v is a steady state gain relating the input voltage to the output flowrate.

Similarly, time response data of the flow valve is obtained at $u_{v,o}$ = 3.5 V. A step input with amplitude of 0.25 V is applied to the system. Comparison of the model and the experimental response is in Fig. 8.

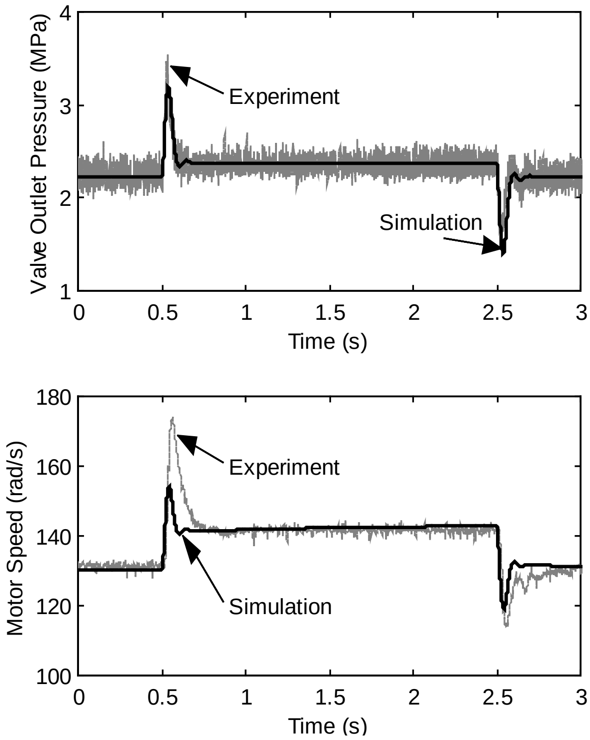


Fig. 8: *Time domain experimental validation*

5 Discussion

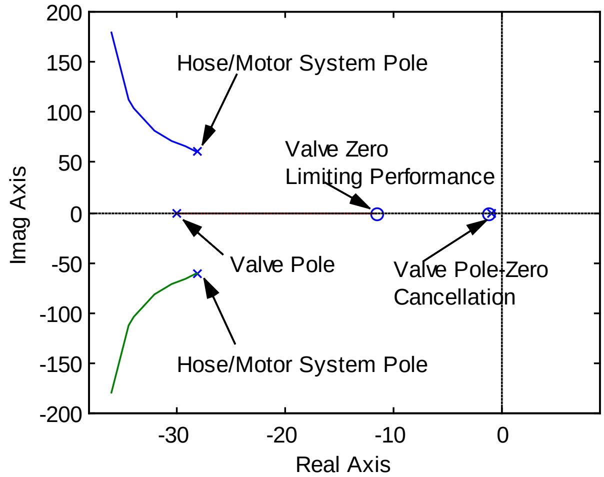


Fig. 9: *Root locus of a Valvistor-controlled system*

As mentioned previously, the "Valvistor" design provides an effective and efficient way to perform electro-hydraulic flow control. Unfortunately from a control designer's viewpoint, the mechanical structure introduces two zeros in the open loop transfer function. Open loop zeros at locations relatively close to the origin limit the closed loop bandwidth (Wilkie, Johnson et al, 2001). For the given valve, the low frequency zero at 1.25 rad/s essentially cancels the low frequency pole at 1 rad/s. This cancellation is desirable otherwise the valve bandwidth would be even more limited to around 1 rad/s (0.16 Hz). However, the design introduces another zero that is slower than the high frequency pole associated with the valve. This means that a feedback control system using the valve as an actuator should be designed with care since this zero limits its closedloop bandwidth. The point can be clearly explained using a root locus of a valve-controlled motor system (Eq. 21 and Fig. 9) under simple proportional gain feedback control. For the condition of Fig. 9, the bandwidth of the closed loop system is always limited to be less than 30 rad/s (4.77 Hz) regardless of the feedback gain magnitude.

In case one wished to remove the effect of the zeros on the closed loop system, the pilot flow could be sent directly to the tank to improve the dynamic performance at a cost of steady state efficiency. Alternately, the valve parameters could be altered to move the undesirable system zeros further into the LHP (left hand plane) to minimize its effect on the closed loop bandwidth, which is equivalent to decreasing (c_1+c_2c_3) or increasing (-K_2c_3+c_1c_4) in Eq. 19. For example, it can be achieved with a smaller c_1 by decreasing the pilot flow gain, a larger K_2 by increasing the main valve size, or a larger c_4 by increasing the slot width. Naturally, these design changes could have to be done within the constraints of the overall valve design.

6 Conclusions

In conclusion, this paper presents a model of a popular type of two-stage electronic flow valve and demonstrates important control-oriented performance limitations inherent in the design. A relatively detailed 5-state model can be created based on first principles. Through model reduction, involving justifiable assumptions, the final simplification involves two zeros for the linearized system. This model is then verified experimentally. The location of the system zeros in the complex plane is a direct result of the physical valve parameters, particularly the amount of pilot circuit flow that gets bypassed to the main flow. The knowledge of this valve zero serves as a guideline for control design when this type of valve is used in a closed loop electrohydraulic system.

Nomenclature

$a_{m,l}$	cross-sectional area of the main poppet valve (large side)
$a_{m,s}$	cross-sectional area of the main poppet valve (small side)
a_p	cross-sectional area of the pilot valve
b	damping coefficient of the gear motor-pump couple
b_m	damping coefficient of the main poppet valve
b_p	damping coefficient of the pilot valve
c_b	downstream hose leakage coefficient
D	displacement of the gear motor and pump
F_p	input force
J	moment of inertial of the gear motor-pump couple

K_e electronic gain
K_m flow gain of the main poppet valve
k_m spring constant of the main poppet valve
K_p flow gain of the pilot valve
k_p spring constant of the pilot valve
K_s longitudinal slot flow gain
K_v constant flow gain of the pilot valve
m_m mass of the main poppet valve
m_p mass of the pilot valve
n_m motor speed
P_a pressure at the flow valve inlet
P_b pressure at the flow valve outlet
P_p pressure in the pilot circuit
Q_a flow rate at the flow valve inlet
Q_b flow rate at the flow valve outlet
Q_p flow rate of the pilot circuit
Q_1 flow rate of the main poppet valve
Q_2 flow rate in the main poppet slot
Q_3 flow rate of the pilot valve
s Laplace operator
u_v Valve input
V_b volume of the downstream hose
V_p volume of the pilot circuit line
x_m position of the main poppet valve
$x_{m,o}$ equilibrium position of the main poppet valve
x_p position of the pilot valve
β fluid bulk modulus
ΔP_o equilibrium pressure drop

Acknowledgements

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