Development of a Fluidic, Hydraulic Servovalve

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This study focuses upon the analysis, design and experimental evaluation of a hydraulic servovalve constructed from laminar proportional fluid amplifiers and resistance feedback elements. A servovalve has been developed in which the pressure/flow characteristics can be contoured to match a specific application through appropriate design of the proportional amplifiers and selection of the feedback elements.
20. Abstract (Cont'd)

A set of design equations have been derived to select the appropriate feedback elements. From the design procedures, two valve configurations have been constructed and tested. Experimental evaluations of the two configurations in terms of static pressure and flow gains and dynamic response have agreed closely with analyses...
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1. INTRODUCTION

Hydraulic control systems are widely used in applications where high force levels, fast response and high power to weight ratios are desired. Applications requiring these features include the positioning of aerodynamic control surfaces, precision control of machine tools, marine control equipment and mobile equipment control systems.

The primary power modulation element utilized in high performance hydraulic servosystems are servovalves. These valves are typically constructed as several stages with initial stages consisting of flapper nozzle or jet pipe valves and final stages employing sliding plate or spool valves. A final stage flow control spool valve is illustrated in Figure 1 with the output pressure/flow characteristics typical of a commercial spool valve.\(^1\)

In electrohydraulic servovalves an electrical transducer such as a torque motor or voice coil is coupled with an initial valve stage so that an electrical input can modulate the valve. Valves have been developed also with direct mechanical modulation of the first stage and in the last few years several development efforts have led to valves which accept low level fluidic inputs.\(^2\)

The classes of valves cited above may have many different combinations of input and first stage elements. However in almost all commercial units the final power level modulation stages consist of sliding spool or plate elements. The moving mechanical elements in these valves are associated with a sensitivity to contamination and requirements for tight manufacturing tolerances which result in a significant fraction of valve cost.

The high reliability, insensitivity to extreme environments and low cost associated with no moving part fluidic elements and the potential for weight and size reduction in comparison to conventional valves are attractive features for servovalves. In applications where these attributes are


\(^2\)R. Deadwyler, Two Stage Servo Valve Development Using a First-Stage Fluidic Amplifier, Harry Diamond Laboratories, HDL-TM-80-21 (July 1980).
Figure 1. Closed center spool valve.
important such as in aircraft, marine and ground vehicle control systems and where the quiescent power drain associated with open-center fluid valves can be accommodated, fluidic servovalves have high potential.

This study focuses upon the conceptualization, analysis, design and experimental evaluation of a hydraulic servovalve constructed from laminar, proportional fluidic elements and resistance feedback elements. The basis for the development is the laminar proportional amplifier (LPA). For the laminar proportional element, the static and dynamic characteristics are well documented, procedures are available to aid in the design of multi-stage amplifier systems and standardized manufacturing techniques have been developed. With the use of LPA elements as a basic building block, a servovalve has been developed in which the pressure/flow characteristics can be contoured to match a specific application through the appropriate design of the proportional amplifiers and selection of the feedback elements. In the development a set of design equations is derived to select the appropriate feedback elements. By the design procedures two valve configurations have been constructed and tested. Experimental evaluations of static pressure and flow gains and dynamic response for the two configurations have agreed closely with the analyses.

2. SERVOVALVE CONFIGURATION DEVELOPMENT

2.1 Laminar Proportional Amplifier Gain Block

The basis for the servovalve development is the laminar proportional fluidic amplifier. Several laminate designs have been developed

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including the 3.1.1.8 design shown in Figure 2 which is used in this study. These laminates may be coupled together to form multistage proportional gain blocks. A three stage block has been assembled as described in Section 3. The proportional multistage gain block is represented schematically in Figure 3 with its nondimensional output characteristics illustrated in Figure 4. The output pressure/flow characteristics of the gain block can be represented quite accurately as shown in Figure 4 with the following expression:

\[
\frac{Q_L}{Q_{Ls}} = \tanh\left(\frac{P_{cd}}{V_{cd}}\right) - \frac{P_{od}}{P_{ods}}
\]

(1)

where

- \(Q_L\) = output load flow
- \(P_{od}\) = amplifier output pressure differential
- \(P_{cd}\) = amplifier input pressure differential
- \(Q_{Ls}\) = saturation output load flow
- \(P_{ods}\) = saturation amplifier output pressure differential

and where the saturation control pressure is defined:

\[
P_{cds} = \frac{P_{ods}}{K_p} = \frac{Q_{Ls}}{K_q}
\]

(2)

with the incremental amplifier static pressure gain (\(K_p\)) and flow gain (\(K_q\)) defined as

\[
K_p = \left. \frac{\partial P_{od}}{\partial P_{cd}} \right|_{Q_L = 0}
\]

(3)

\[
K_q = \left. \frac{\partial Q_L}{\partial P_{cd}} \right|_{P_{od} = 0}
\]

(4)

---

Figure 2. Scaled drawing of HDL 3.1.1.8 laminate.
Figure 3. Multistage amplifier configuration.
Figure 4. Multistage amplifier output characteristics.

(a) Tanh function approximation of amplifier output characteristics.

(b) Amplifier pressure gain.

(b) Amplifier flow gain.

Figure 4. Multistage amplifier output characteristics.
and where the incremental quantities $K_p$ and $K_q$ are related by the incremental amplifier flow/pressure characteristic:

$$K_qP = \frac{\partial Q}{\partial P_{od}} P_{cd} = \text{constant} = \frac{K_q}{K_p}$$

$$\text{(5)}$$

2.2 Servovalve Configuration

The basic gain block output pressure/flow characteristics described by equation (1), and shown in Figure 4 are relatively linear with respect to the output pressure/flow characteristic for a specified value of input pressure differential. The gain block output characteristics have a high sensitivity of output flow to output pressure. To achieve the typical spool valve flow control characteristics shown in Figure 1, modification of the gain block characteristics is required so that the sensitivity of output flow to output pressure is significantly reduced. The desired modification can be achieved by using feedback principles. The conceptual development of a fluidic servovalve is illustrated in Figure 5 where the influence of positive pressure feedback and negative flow feedback is illustrated.

In Figure 5a, the basic proportional amplifier is augmented with input resistors $R_1$ and pressure feedback resistors $R_{fp}$. As a differential input signal $(P_{id})$ is applied to the configuration and pressure develops across the load $(P_{od})$, this pressure is fed back through $R_{fp}$ to increase the differential pressure $(P_{cd})$ at the amplifier control ports and increase the load pressure. This positive pressure feedback alters the slope of the output pressure/flow characteristic and modifies the intercept of the characteristics on the output pressure axis as shown in the figure.

In Figure 5b, the proportional amplifier is augmented with a flow sensing resistor $R_Q$, flow feedback resistors $R_{fq}$ and input resistors $R_1$. As an input pressure differential $(P_{id})$ is applied and flow develops
(a) Proportional amplifier with positive pressure feedback

(b) Proportional amplifier with negative flow feedback

\[ P_L = \text{Pressure drop across load} \]
\[ R_{fp} = \text{Pressure feedback resistance} \]
\[ Q_L = \text{Output load flow} \]
\[ R_{fq} = \text{Flow feedback resistance} \]
\[ P_{id} = P_{il} - P_{ir} = \text{Servovalve input pressure differential} \]
\[ R_1 = \text{Servovalve input resistance} \]
\[ R_Q = \text{Flow sensing resistor} \]

Figure 5. Fluidic servovalve development.
through the load, a pressure drop occurs across the flow sensing resistor. This pressure drop is fed back through resistors $R_{fp}$ to decrease the pressure differential ($P_{cd}$) at the amplifier control port and reduce the flow to the load. This negative flow feedback reduces the sensitivity of the output flow to output pressure and generates a flow control type of characteristic. When both pressure and flow feedback features are combined the valve configuration of Figure 6 is obtained. With the appropriate selection of feedback elements the characteristics of flow or pressure control servovalves may be obtained. The basic configuration depicted in Figure 6 is analyzed in the following sections.

2.3 Servovalve Static Characteristics

The static characteristics of the servovalve configuration may be derived by combining the description of the gain block given in equation (1) with the resistance element characteristics using flow and pressure summation circuit analysis laws. In the derivation, it is assumed that feedback and input resistance elements are linear.

By applying flow summation at the right and left amplifier control ports in Figure 6, the following equation may be derived:

$$\frac{P_{il} - P_{ir}}{R_i} + \frac{P_{or} - P_{oL}}{R_{fp}} + \frac{P_{oL'} - P_{or}}{R_{fp}} = \frac{P_{cd} - P_{cr}}{\sum R} \tag{6}$$

where each pressure is identified in Figure 6 and where $\sum R'$ is the parallel combination of $R_i$, $R_a$, $R_{fq}$ and $R_{fp}$ with

- $R_i =$ servovalve input resistance
- $R_a =$ amplifier control port input resistance
- $R_{fq} =$ flow feedback resistance
- $R_{fp} =$ pressure feedback resistance

By noting the relationships

$$P_{id} = P_{il} - P_{ir} \tag{7}$$
$$P_{oL'} - P_{oL} = P_L \tag{8}$$
$$P_{or} - P_{oL'} = R_{Q L} \tag{9}$$

14
Figure 6. Fluidic servovalve schematic.
where
\[ R_Q = \text{flow sensing resistor} \]
\[ Q_L = \text{output load flow} \]

and defining the maximum output load flow \( Q_{Lm} \) and maximum pressure drop \( P_{Lm} \) across the load as

\[ Q_{Lm} = \frac{K_q}{1 + K_{qp} R_a} P_{cds} \]  

(10)

\[ P_{Lm} = K_p P_{cds} \]  

(11)

Equations (1) and (6) may be combined to yield

\[ \bar{Q}_L + \bar{P}_L = \tanh[\alpha \bar{P}_{id} + \beta \bar{Q}_L + \gamma \bar{P}_L] \]  

(12)

where

\[ \bar{P}_{id} = P_{id}/P_{idm} \]
\[ \bar{Q}_L = Q_L/Q_{Lm} \]
\[ \bar{P}_L = P_L/P_{Lm} \]

with \( P_{idm} \) = maximum servovalve input pressure differential

Equation (12) is the general servovalve relationship between the output pressure/flow characteristic and the input control pressure. The nonlinear, nondimensional equation is characterized by three parameters:

\[ \alpha = \frac{R'_P}{R_1} \frac{P_{idm}}{P_{cds}} = \frac{R'_P}{R_1} \frac{K_q P_{idm}}{(1 + K_{qp} R_a) Q_{Lm}} \]  

(13)

\[ \beta = \frac{R'_K R_a Q}{1 + K_{qp} R_a} \left[ \frac{1}{R_{fp}} - \frac{1}{R_{fq}} \right] \]  

(14)

\[ \gamma = \frac{K R'_P}{R_{fp}} \]  

(15)
The coefficient $\alpha$ is related to the input pressure gain while $\beta$ and $\gamma$ are related to the flow and pressure feedback coefficients. The variable $\bar{P}_{id}$ varies from 0 to 1 and parameterizes the $\bar{P}_L/\bar{Q}_L$ output characteristic in terms of nondimensional input pressure.

The valve blocked load pressure is derived from equation (12) by setting $\bar{Q}_L = 0$ and may be written as

\[
\frac{1}{2} \ln \left[ \frac{1 + \bar{P}_L}{1 - \bar{P}_L} \right] - \gamma \bar{P}_L = \alpha \bar{P}_{id}
\]  

(16)

while the valve no load flow may be derived from equation (12) by setting $\bar{P}_L = 0$ to obtain

\[
\frac{1}{2} \ln \left[ \frac{1 + \bar{Q}_L}{1 - \bar{Q}_L} \right] - \beta \bar{Q}_L = \alpha \bar{P}_{id}
\]  

(17)

The blocked load pressure equation (16) and no load flow equation (17) are plotted as a function of input pressure in Figures 7 and 8. The blocked load pressure increases as the input pressure increases and as the parameter $\gamma$ is increased corresponding to a reduction in the pressure feedback resistor $R_{fp}$. The no load output flow increases with increasing input pressure and increasing values of $\beta$. The value of $\beta$ may be positive ($R_{fp} < R_{fq}$) or negative ($R_{fq} < R_{fp}$) depending upon the values of the pressure and flow feedback resistors.

Values of positive $\beta$ increase the flow while values of negative $\beta$ decrease the flow for a given level of control pressure.

The complete valve characteristics computed from equation (12) are shown in Figure 9 for selected values of $\alpha$, $\beta$ and $\gamma$. These curves show that small values of $\alpha$ and $\beta$ generate characteristics that are similar to those of a flow control valve. As $\alpha$ is reduced the flow curves become a more linear function of input pressure. As $\alpha$ is increased the valve characteristics tend to approach those of a pressure control valve. From plots such as those generated from equation (12) a valve may be designed by selecting appropriate values of $\alpha$, $\beta$ and $\gamma$ to have characteristics similar to those of either a conventional pressure or flow
Figure 7. Servovalve blocked load pressure.
Figure 8. Servovalve no load flow.
Figure 9. Servovalve characteristics for values of $\alpha$, $\beta$ and $\gamma$
Figure 9. Servo valve characteristics for values of $\alpha$, $\beta$ and $\gamma$. (con'd)
control valve or to have a blend of pressure/flow control characteristics. Thus, the valve may be specifically designed for matching a given load.

2.4 Component Selection

The three quantities $\alpha$, $\beta$ and $\gamma$ completely characterize the nondimensional output pressure/flow characteristics of the general valve configuration. The values of the valve feedback resistances can be selected to achieve the desired values of $\alpha$, $\beta$ and $\gamma$. While $\alpha$, $\beta$ and $\gamma$ can be selected independently, the resistances are interrelated and are also dependent upon amplifier parameters.

If it is assumed that a basic multistage gain block has been constructed to provide the maximum required levels of absolute pressure and flow, then the amplifier values of $K_p$, $K_q$, $P_{cds}$, $Q_{ls}$, $P_{ls}$ and $R_a$ are known. When these values are coupled with a set of desired values of $\alpha$, $\beta$ and $\gamma$ selected to yield the overall valve characteristics, the following method may be used to select dimensionless values of feedback resistances.

The value of $\alpha$ may be written as

$$\alpha = \frac{R'}{R_1} \frac{K_p P_{idm}}{P_{Lm}} \approx \frac{R_a}{R_a + R_1} \frac{K_p P_{idm}}{P_{Lm}} \quad (18)$$

where it is noted that because $R_a$ and $R_1$ are usually small compared with other resistances $R'/R_1$ is approximated by $R_a/(R_a + R_1)$. Thus from $\alpha$ and amplifier parameters, $R_1$ and the approximation to $R'$ may be determined when a maximum control input pressure differential $P_{idm}$ is selected. Once $R'$ is calculated then $R_{fp}$ may be determined directly from the requirement to match $\gamma$ in equation (11). The resistance $R_{fq}$ is determined from the requirement to match $\beta$ in equation (10) after the flow sensing resistor $R_Q$ has been determined.

The flow sensing resistor $R_Q$ is selected to provide a low resistance so that the flow is not restricted through the load and provide a sufficiently large resistance so that a reasonable value of $R_{fq}$ is obtained for the given value of $\beta$. If $R_{fq}$ is too small then the amplifier output is loaded down and large feedback flows occur.
Based upon the detailed development of valve design procedures, a good guideline for $R_Q$ is given by

$$\frac{0.2}{K_{QP}} < R_Q < \frac{0.4}{K_{QP}}$$ (19)

Once $R_Q$ is selected all other design resistance values may be directly computed.

2.5 Small Perturbation Characteristics

The static characteristics of the servovalve for incremental deviations from an operating point may be derived by linearizing the nonlinear static characteristic. The result of linearizing equation (12) is

$$\bar{Q}_L = \frac{1}{(1-\beta)} [\alpha \Delta P_{id} + (\gamma - 1) \Delta P_L]$$ (20)

where $\Delta(\ )$ indicates an incremental deviation.

The servovalve flow gain $G_q$ and pressure gain $G_p$ derived from equation (20) are

$$G_p = \left. \frac{\partial P_L}{\partial P_{id}} \right|_{Q_L=0} = \frac{1}{R_1} \frac{K_p R'}{1 - \frac{K_p}{R_r} - \frac{1}{R_p}}$$ (21)

$$G_q = \left. \frac{\partial Q_L}{\partial P_{id}} \right|_{P_L=0} = \frac{1}{R_1} \frac{1}{1 + K_p R' \left( \frac{1}{R_q} - \frac{1}{R_r} \right) + \frac{1}{K_p}}$$ (22)

---

The valve blocked load pressure gain and no load flow gain are expressed in equations (21) and (22) directly in terms of feedback element and amplifier parameters. The pressure gain can be maximized by setting $R_{fp} = K_p R'$ which is equivalent to setting $\gamma = 1$. If this condition is met the pressure gain approaches infinity and the output curves approach horizontal lines in Figure 9 indicating a load insensitivity. The blocked load pressure gain is sensitivity to $R_{fp}$. For large $R_{fp}$ the positive pressure feedback is small and $G_p/K_p$ approaches $R_a/(R_a + R_f)$ and for small values of $R_{fp}$, $G_p/K_p$ increases markedly as $R_{fp}$ is reduced.

The flow gain given in equation (22) decreases as $R_{fp}$ is reduced and increases as $P_{fp}$ is reduced.

Both the valve pressure gain $G_p$ and flow gain $G_q$ are functions of the amplifier pressure gain $K_p$ and flow gain $K_q$. Because of the servo-valve positive pressure and negative flow feedback, the servo-valve pressure gain increases faster than the amplifier pressure gain while the servo-valve flow gain increases less rapidly than the amplifier flow gain.

3. PROTOTYPE SERVOVALVE CONSTRUCTION AND EVALUATION

3.1 Servovalve Construction

The general relationships derived in the previous section have been used to design two prototype servovalve configurations. Both prototype servovalves have been constructed by using the three stage gain amplifier summarized in Table 1 and laminar flow resistors constructed from small cross section rectangular passages. The servovalve's general configuration is illustrated in Figure 10. The basic three-stage amplifier is shown with the stages stacked back to back, separated by manifold plates. The pressure and flow feedbacks as well as the flow sensing resistors are connected directly to the output ports of the final amplifier stage through a manifold. The low pressure sides of the feedback resistors are connected to the control ports of the amplifier first stage through flexible plastic hoses. Vent ports labeled $V_1$, $V_2$ and $V_3$ are piped to the reservoir. A flapper nozzle valve driven by a torque motor.
# TABLE 1: FLUIDIC AMPLIFIER CONFIGURATION

## A. LAMINATE DESCRIPTION

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design HDL</td>
<td>HDL 3.1.1.8</td>
</tr>
<tr>
<td>Laminate Height $h_s$</td>
<td>$0.1 \text{ mm} \ [0.004 \text{ in.}]$</td>
</tr>
<tr>
<td>Supply Nozzle Width $b_s$</td>
<td>$0.5 \text{ mm} \ [0.020 \text{ in.}]$</td>
</tr>
</tbody>
</table>

## B. THREE-STAGE AMPLIFIER

<table>
<thead>
<tr>
<th>Stage</th>
<th>Section aspect ratio, $(\sigma = h_s/b_s)$</th>
<th>Number of sections per amplifier</th>
<th>Supply pressure, $P_s$ (kPa) [psi]</th>
<th>Estimated supply flow $Q_s$ (m$^3$/s) [cis]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.6</td>
<td>2</td>
<td>620 [90]</td>
<td>$0.70 \times 10^{-5} \ [0.43]$</td>
</tr>
<tr>
<td>2</td>
<td>0.6</td>
<td>3</td>
<td>2068 [300]</td>
<td>$1.9 \times 10^{-5} \ [1.18]$</td>
</tr>
<tr>
<td>3</td>
<td>0.4</td>
<td>4</td>
<td>6895 [1000]</td>
<td>$3.1 \times 10^{-5} \ [1.91]$</td>
</tr>
</tbody>
</table>

## C. AMPLIFIER INCREMENTAL PARAMETERS

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$p_{ods}$</td>
<td>$3654 \text{ kPa} \ [530 \text{ psid}]$</td>
</tr>
<tr>
<td>$Q_{Ls}$</td>
<td>$1.80 \times 10^{-5} \text{ m}^3 \ [1.1 \text{ cis}]$</td>
</tr>
<tr>
<td>$P_{cds}$</td>
<td>$9.96 \text{ kPa} \ [1.45 \text{ psid}]$</td>
</tr>
<tr>
<td>$K_p$</td>
<td>367</td>
</tr>
<tr>
<td>$K_q$</td>
<td>$1.805 \times 10^{-6} \text{ m}^3/\text{s/kPa} \ [0.76 \text{ cis/psid}]$</td>
</tr>
<tr>
<td>$K_{qp}$</td>
<td>$4.99 \times 10^{-9} \text{ m}^3/\text{s/kPa} \ [0.0021 \text{ cis/psid}]$</td>
</tr>
</tbody>
</table>
Figure 10  Schematic of servovalve design.
is used as an input signal generator to the servovalve. It is connected to
the servovalve through the input resistor $R_i$.

The general servovalve configuration is a breadboard configuration
that permits measurement of pressures at each stage in the amplifier and
feedback network. It has been constructed by using HDL 3.1.1.8 laminates.
However, the overall configuration has not utilized the standard packaging
techniques which allow an integrated package to be constructed since it
was desired to make measurements in the servovalve at many intermediate
locations.

In all tests performed on the servovalves, Univis J-43 oil main-
tained at an operating temperature of 27°C was used. Under this condition
the fluid density is $8.69 \times 10^2$ N-s$^2$/m$^4$ and the viscosity is $1.88 \times 10^{-2}$
N-s/m$^2$.

Two servovalves have been constructed and tested. The two
valves are similar except for the values of feedback resistors. Servovalve
1 was constructed with $\alpha = 1.2$, $\beta = 0$ and $\gamma = 1$ while servovalve 2 was
constructed with $\alpha = 1.2$, $\beta = -0.2$ and $\gamma = 1$.

3.2 Resistor Characteristics

Laminar flow resistors made from small cross section area
rectangular ducts are used in the servovalve. The resistors were made
by cutting a slot in brass shim stock and sandwiching the shim stock be-
tween two manifold plates. This type of resistor is simple to construct
and is self-bleeding, i.e., air bubbles are forced out of the passageway
with the oil flow. An example resistor is shown in Figure 11. The shim
resistor includes serveral holes for mounting, alignment and porting.
The thickness of the shim, the width of the milled slot and the length
of the slot determine the resistance as well as the viscosity of the
fluid.

The dimensions and resistance values of the resistors used
in the servovalves are summarized in Table 2. The measured pressure-flow
characteristics of the feedback and input resistors are shown in Figure
12. They are linear over the range of the test.
Figure 11. Pressure feedback resistor design.
### TABLE 2: RESISTOR DIMENSIONS AND VALUES

<table>
<thead>
<tr>
<th>Resistor</th>
<th>Width [mm]</th>
<th>Height [mm]</th>
<th>Length [mm]</th>
<th>Measured resistance N-s/m²</th>
<th>Estimated inertance N-s²/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>R₁</td>
<td>0.533</td>
<td>0.813</td>
<td>12.7</td>
<td>$2.48 \times 10^{10}$</td>
<td>$2.53 \times 10^{7}$</td>
</tr>
<tr>
<td>R₁</td>
<td>0.304</td>
<td>0.813</td>
<td>8.89</td>
<td>$5.30 \times 10^{10}$</td>
<td>$3.07 \times 10^{7}$</td>
</tr>
<tr>
<td>R₂</td>
<td>3.175</td>
<td>0.406</td>
<td>27.94</td>
<td>$3.83 \times 10^{10}$</td>
<td>$1.85 \times 10^{7}$</td>
</tr>
<tr>
<td>R₂</td>
<td>6.35</td>
<td>0.203</td>
<td>27.94</td>
<td>$9.30 \times 10^{10}$</td>
<td>$1.85 \times 10^{7}$</td>
</tr>
<tr>
<td>Rᶠᵖ</td>
<td>0.254</td>
<td>0.203</td>
<td>57.15</td>
<td>$7.22 \times 10^{12}$</td>
<td>$9.56 \times 10^{8}$</td>
</tr>
<tr>
<td>Rᶠᵖ</td>
<td>0.203</td>
<td>0.203</td>
<td>57.15</td>
<td>$1.20 \times 10^{13}$</td>
<td>$1.19 \times 10^{9}$</td>
</tr>
<tr>
<td>Rᶠᵖ</td>
<td>0.254</td>
<td>0.203</td>
<td>57.15</td>
<td>$7.22 \times 10^{12}$</td>
<td>$9.56 \times 10^{8}$</td>
</tr>
</tbody>
</table>
Figure 12. Resistor characteristics.
3.3 Multistage Amplifier Characteristics

A summary of the multistage amplifier parameters is given in Table 1. The three stage amplifier output pressure/flow characteristics are presented in Figure 4, where they have been compared with the analytical expression of equation (12). As noted in Section 2, the analytical expression agrees well with the data. As shown by the data the multistage amplifier has good saturation characteristics and an input pressure of about 10 kPa produces a saturation output pressure of 3650 kPa.

3.4 Servovalve Static Characteristics

Two servovalve configurations have been constructed and tested. The two valves both consist of the multistage gain amplifier described in the previous section and the laminar flow resistors. The parameters for the two servovalve configurations are summarized in Table 3.

The static characteristics of the two configurations have been measured in all four quadrants of operation using the experimental techniques described by Lee.\(^7\) The experimental data are displayed in Figures 13 and 14 and are compared with the analytically predicted characteristics. The experimental data for both servovalve configurations agree closely with the theoretical data. Some asymmetry is illustrated in the experimental characteristics which is due to asymmetry in the multistage amplifier characteristics. It is expected that these asymmetries can be eliminated when the multistage amplifier is fabricated with more uniform manufacturing methods rather than built up in a breadboard configuration.

The differences between the two valve configurations due to the values of the flow coefficient \(\beta\) are reflected in both the experimental and theoretical data. Configuration 1 (\(\beta = 0\)) has load pressure/flow characteristics which have a significant slope and represent a blend of pressure/flow control characteristics. Configuration 2 (\(\beta = -0.21\)) has load pressure/flow characteristics that are relatively flat indicating a low sensitivity to load

---

### TABLE 3: SERVOVALVE CONFIGURATIONS

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Parameter Value</th>
<th>Parameter Value</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Configuration 1</td>
<td>Configuration 2</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>1.24</td>
<td>1.24</td>
</tr>
<tr>
<td>$\beta$</td>
<td>0</td>
<td>-0.21</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>0.99</td>
<td>1.0</td>
</tr>
<tr>
<td>$R_i$</td>
<td>$2.48 \times 10^{10}$ N-s/m$^5$</td>
<td>$5.3 \times 10^{10}$ N-s/m$^3$</td>
</tr>
<tr>
<td>$R_Q$</td>
<td>$3.83 \times 10^{10}$ N-s/m$^5$</td>
<td>$9.3 \times 10^{10}$ N-s/m$^3$</td>
</tr>
<tr>
<td>$R_{fp}$</td>
<td>$7.22 \times 10^{12}$ N-s/m$^5$</td>
<td>$1.2 \times 10^{13}$ N-s/m$^3$</td>
</tr>
<tr>
<td>$R_{fq}$</td>
<td>$7.22 \times 10^{12}$ N-s/m$^5$</td>
<td>$7.22 \times 10^{12}$ N-s/m$^3$</td>
</tr>
<tr>
<td>$R_a$</td>
<td>$8.55 \times 10^{10}$ N-s/m$^5$</td>
<td>$8.55 \times 10^{10}$ N-s/m$^3$</td>
</tr>
<tr>
<td>$P_{Lm}$</td>
<td>3654 kPa</td>
<td>3654 kPa</td>
</tr>
<tr>
<td>$Q_{Lm}$</td>
<td>$1.51 \times 10^{-5}$ m$^3$/s</td>
<td>$1.22 \times 10^{-5}$ m$^3$/s</td>
</tr>
</tbody>
</table>
Figure 13. Configuration 1 servo valve output characteristics.
Figure 14. Configuration 2 servovalve output characteristics.
pressure. The output characteristic of configuration 1 is similar to that of the spool valve shown in Figure 1 while the output characteristics of configuration 2 are less sensitive to load pressure than those of a spool valve. The data show that as \( \beta \) is decreased the curves approach an ideal flow control valve in which the output flow is insensitive to load pressure.

3.5 Servovalve Efficiency and Quiescent Flow

Measures of servovalve effectiveness include the fraction of total valve power delivered to a load and the quiescent flow drain. The normalized efficiency, \( \eta \), of the fluidic servovalve can be defined as the power delivered to the load divided by the maximum load pressure/load flow product:

\[
\eta = \frac{P_L Q_L}{P_{Lm} Q_{Lm}}
\]  

(23)

This measure of efficiency is plotted in Figure 15 for comparison with the efficiency of a spool valve defined as

\[
\eta = \frac{P_L}{P_s} \sqrt{1 - \frac{P_L}{P_s}}
\]  

(24)

and double flapper nozzle valve with equal upstream and downstream orifices where the efficiency is defined

\[
\eta = \frac{P_L Q_L}{P_{Lm} Q_{Lm}}
\]  

(25)

The figure indicates that all three valves reach peak efficiencies with \( 0.51 < P_L/P_{Lm} < 0.7 \) with the spool valve maximum of 0.38, the flapper nozzle valve of maximum 0.33 and the fluidic valve maximum of 0.27. This comparison shows that the fluidic valve power ratio is 70\% of the spool valve and 82\% of the double flapper nozzle valve.

In addition to the power ratio, the quiescent flow through the valve is of interest. The spool valve is a closed center valve and ideally the spool has 0\% leakage while in practice spool leakage rates are often 5 to 10\% of the spool maximum flow. The double flapper nozzle valve and the
<table>
<thead>
<tr>
<th>Valve</th>
<th>Efficiency, $\eta$</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spool</td>
<td>$\frac{P_L Q_L}{P_s Q_L m}$</td>
<td>Closed center</td>
</tr>
<tr>
<td>Flapper nozzle</td>
<td>$\frac{P_L Q_L}{P_L m Q_L m}$</td>
<td>Equal null upstream and downstream orifice areas</td>
</tr>
<tr>
<td>Fluidic</td>
<td>$\frac{P_L Q_L}{P_L m Q_L m}$</td>
<td>$\alpha = 1.2$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\beta = -0.2$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\gamma = 1.0$</td>
</tr>
</tbody>
</table>

Figure 15. Servovalve efficiencies.
fluidic valve are both open center valves. The flapper nozzle valve has a maximum load flow to quiescent flow ratio of 1 while the fluidic servovalve has a ratio of 0.31.

In the fluid servovalve, the maximum flow delivered to the load is $1.8 \times 10^{-5} \ m^3/s$ while the quiescent flow in the three stages of the amplifier is $5.76 \times 10^{-5} \ m^3/s$. It is anticipated that through modification of the multistage amplifier design the ratio of load to quiescent flow may be increased significantly.

4. SERVOVALVE DYNAMIC PERFORMANCE

4.1 Dynamic Model

A dynamic model for the servovalve valid for small perturbations from an operating point may be synthesized from models for the multistage amplifier and the circuit resistors.

The circuit input and feedback resistors are constructed from laminar flow passages which have a measurable inerterance as well as resistance. Dynamically each resistor is represented as an impedance:

$$Z_j(s) = R_j + L_j s$$  \hspace{1cm} (26)

where

- $Z_j$ = impedance of element $j$
- $R_j$ = resistance of element $j$
- $L_j$ = $j$th fluid inerterance
- $s$ = Laplace operator

The value of each $R_j$ is taken from the experimental measurements cited in Section 3 while the inerterance is estimated from the relationship

$$L_j = \rho \frac{L_j}{A_j}$$  \hspace{1cm} (27)

where

- $\rho$ = fluid density
- $L_j$ = $j$th passage length
- $A_j$ = $j$th passage area
The values of inertance for each resistor are listed in Table 2.

The multistage amplifier for incremental deviations may be represented as a dynamic pressure and flow gain:

\[
\frac{\Delta P_{cd}(s)}{\Delta P_{cd}(s)} = H_p(s) \tag{28}
\]

\[
\frac{\Delta Q_L(s)}{\Delta P_{cd}(s)} = H_q(s) \tag{29}
\]

where

\[H_p(s) = \text{amplifier dynamic pressure gain transfer function}\]
\[H_q(s) = \text{amplifier dynamic flow gain transfer function}\]

Studies by Lee\(^7\) have shown that the two amplifier transfer functions may be represented as

\[
H_p = K_p \frac{1 - (T_p/2)s + (T_p^2/8)s^2}{1 + (T_p/2)s + (T_p^2/8)s^2} \tag{30}
\]

\[
H_q = K_q \frac{1 - (T_q/2)s + (T_q^2/8)s^2}{1 + (T_q/2)s + (T_q^2/8)s^2} \tag{31}
\]

where

\[T_p = \text{amplifier pressure gain delay time}\]
\[T_q = \text{amplifier flow gain delay time}\]

The amplifier pressure and flow gain transfer functions given in

equations (30) and (31) are compared in Figures 16 and 17 with experimentally measured amplifier characteristics for \( T_p = 1.1 \times 10^{-3} \) s and \( T_q = 1.1 \times 10^{-3} \) s. There is close agreement between the experimental data and the analytical expressions.

The complete servovalve dynamic transfer function for small deviations may be derived replacing the static gains and resistances in equations (21) and (22) with the dynamic equivalents to obtain

\[
G_p(s) = \frac{1}{Z_1(s)} \frac{H_p(s)Z'(s)}{1 - \frac{1}{Z_{fp}(s)} H_p(s)Z'(s)}
\]

(32)

\[
G_q(s) = \frac{1}{Z_1(s)} \frac{H_q(s)Z'(s)}{1 + H_q(s)Z'(s)Z_q(s) \left[ \frac{1}{Z_{fq}(s)} - \frac{1}{Z_{fp}(s)} + \frac{1}{H_p(s)Z'(s)} \right]}
\]

(33)

These two expressions provide an analytical representation of the servovalve dynamic pressure and flow gain responses.

4.2 Experimental Dynamic Performance

The dynamic frequency response characteristics of the servovalve have been measured by using the techniques described by Lee.\(^7\) The experimentally measured pressure and flow gains as a function of frequency are plotted in Figures 16 and 17 for valve configuration 1. The data show that the pressure gain reaches 90 deg phase shift at 7 Hz and the flow gain reaches 90 deg phase shift at approximately 60 Hz.

The analytical expressions given in equations (32) and (33) also are plotted. In the analysis the impedances are represented by the values of resistance and inductance cited in Table 2. However, for cases in which

---

Steady state gain: 86.7 dB

Equation (30)

Experimental
○ - Amplifier
□ - Valve

Analytical

Figure 16. Amplifier and servovalve blocked load frequency response.
Figure 17. Amplifier and servo valve no load frequency response.
\( \frac{L_j}{R_j} < 1.6 \times 10^{-4} \) s, the inertance \( L_j \) was neglected since it represents less than 10% of the impedance at frequencies below 100 Hz. Also the impedance for the resistor \( R_a \) was modified to include the capacitance in the flexible plastic hoses. Thus, the impedance \( Z_a \) was represented with inertance neglected and hose capacitance included as

\[
Z_a = \frac{R_a}{R_a C_1 s + 1}
\]  

(34)

where

\[
C = 171 \text{ m}^2/\text{N}
\]  
as determined by Lee.\(^7\)

The dynamic model including the effects of hose capacitance agrees closely with the experimental data.

In the valve tested, the use of the flexible feedback hoses contribute to the valve dynamic response. If the hose capacitance is eliminated, the analytical model indicates that the flow gain frequency response can be extended beyond 100 Hz before 90 deg phase shift occurs while the pressure gain frequency response is extended beyond 20 Hz at 90 deg phase shift.

It is anticipated that construction of the servovalve in an integrated package would extend the flow gain frequency response into the 100 Hz range before 90 deg phase shift is reached.

In commercial electrohydraulic servovalves of the same flow capability as the fluidic servovalve, the flow gain frequency response has been measured which indicates the 90 deg phase shift is reached at about 100 Hz.\(^6\)


SUMMARY AND CONCLUSIONS

A pure fluid servovalve configuration has been developed which has output pressure/flow characteristics that can be contoured to be similar to flow or pressure control servovalves or a blend of these characteristics. General static design relationships have been derived for configurations employing multistage fluid amplifiers and linear resistance elements. The relationships are useful in determining the feedback element parameters required to achieve a specified output characteristic.

Two prototype valve configurations were constructed by using standard laminar proportional amplifiers. The data obtained with these valves verifies the general static and dynamic analyses. It has demonstrated the ability to contour static characteristics through changes in feedback elements and indicated a dynamic flow response potential which is comparable to commercial electrohydraulic valves of the same flow capability. The valve configurations tested were constructed in breadboard fashion and thus both improvements in dynamic response and reduction in overall size is possible when integrated circuit designs are employed.

The study has identified several areas for further development. To achieve a minimum size and weight valve with standard components, servovalves should be constructed with HDL standard integrated laminates, spacers and resistors in an integrated package. A preliminary layout study has shown that construction of a valve similar to configuration I with the standard elements is possible in a package approximately 3 x 3 x 5 cm which weighs 0.65 kg. Thus, fluid servovalves have potential for reduced size and weight in comparison to typical spool valves of the same capability.

In the current study, the servovalves were evaluated with hydraulic oil maintained at ambient temperature. Further effort is planned to evaluate valve performance as a function of temperature.
One of the attributes of the pure fluidic valve is the relatively high quiescent flow requirement associated with open center valves. Further optimization of the multistage gain amplifier is merited to reduce the ratio of the quiescent to maximum load flow.
NOMENCLATURE

$A_j$ jth passage area

$b_s$ supply nozzle width (figure 2)

$C$ fluid capacitance

$G_p$ servovalve pressure gain

$G_q$ servovalve flow gain

$H_p$ amplifier dynamic pressure gain

$H_q$ amplifier dynamic flow gain

$K_p$ incremental amplifier static pressure gain

$K_q$ incremental amplifier static flow gain

$K_{qp}$ incremental amplifier output admittance (equation 5)

$l_j$ jth passage length

$L_1$ load (figure 5)

$L_j$ jth fluid inertance

$P_{cd}$ amplifier input pressure differential

$P_{cds}$ amplifier saturation input differential

$P_{cl}$ amplifier left input pressure (figure 3)

$P_{cr}$ amplifier right input pressure (figure 3)

$P_{id}$ servovalve input pressure differential

$P_{idm}$ maximum servovalve input pressure differential

$P_{il}$ servovalve left input pressure (figure 5)

$P_{ir}$ servovalve right input pressure (figure 5)

$P_L$ pressure drop across load (equation 8)

$P_{LM}$ maximum pressure drop across load (equation 11)

$P_{od}$ amplifier output pressure differential

$P_{ods}$ saturation amplifier output pressure differential

$P_{ol}$ pressure defined in figures 3 and 6

$P_{ol}'$ pressure defined in figure 6

$P_{or}$ pressure defined in figures 3 and 6

$P_s$ supply pressure

$P_l$ pressure defined in figure 1
P₂
Qₐ
QLₜₖₜ
QLₘₖₜ
Qₘₖₜ
Rₐ
Rᶠₚ
Rᶠₗₚ
Rₗ
Rᶠ
R'ₜ
S
Tₚ
Tₗ
V
x
Zₐ
Zᶠₚ
Zᶠₗₚ
Zₗ
Zₛ
Zᶠ
Z'ₜ
α
β
γ
η
ρ

pressure defined in figure 1
output load flow
saturation output load flow
maximum output load flow (equation 10)
supply flow
amplifier control port input resistance
pressure feedback resistance
flow feedback resistance
servo valve input resistance
flow sensing resistor
parallel combination of Rₗ, Rₐ, Rᶠₗₚ, Rᶠₚ
Laplace Operator
amplifier pressure gain delay time
amplifier flow gain delay time
vent (figure 10)
valve spool displacement (figure la)
impedance defined in equation 34
pressure feedback impedance
flow feedback impedance
servo valve input impedance
impedance of element j
flow sensing impedance
parallel combination of Zₗ, Zₐ, Zᶠₚ, Zᶠₗₚ
nondimensional valve coefficient (equation 13)
nondimensional valve coefficient (equation 14)
nondimensional valve coefficient (equation 15)
efficiency
fluid density