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Two-Stage Servovalve Development Using a First-Stage Fluidic Amplifier

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This study of first-stage fluidic amplifier usage to date in two-stage servovalves compares first-stage hydraulic amplifiers (torque motor, flapper-nozzle valve) and first-stage fluidic amplifiers. Present trends in fluidic servovalve development are also discussed.

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20. Abstract (Cont'd)

The use of first-stage fluidic amplifiers and second-stage, spring-centered spool valves to date shows that this servovalve configuration is feasible. However, fluidic amplifier gain and null variation with fluid temperature is a problem.

The first-stage fluidic amplifier is comparable to the hydraulic amplifier in terms of input signal power, output power, and frequency response. The hydraulic amplifier has less leakage flow and is less temperature sensitive. However, the fluidic amplifier is more reliable in terms of failure modes which cause actuator hardover and in its ability to operate with higher levels of fluid contamination. A two-stage fluidic servovalve with a first-stage, mechanically actuated fluidic-type amplifier and a second-stage spool valve with mechanical feedback was found to be the best two-stage fluidic servovalve configuration.

Future fluidic systems and subsystems will have to interface with electronic controllers or central processing units. This means that fluidic systems and subsystems must have dual input (fluidic and electronic) servovalves. Microprocessor-level signals may also be used to drive servovalves with first-stage fluidic-type amplifiers--that is, amplifiers with moving parts. Finally, fluidic-type amplifiers are successfully being used in commercially available electrohydraulic servovalves.

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CONTENTS

FIGURES

1. INTRODUCTION

Two-stage fluidic servovalves are needed in fluidic control systems requiring power actuation. Continuing studies have been conducted by the U.S. Army Aviation Research and Technology Laboratories (AVRADCOM), the Harry Diamond Laboratories (HDL), university researchers, and industrial engineers to improve servovalves' dynamic performance, power consumption, reliability, and cost. At present, a fluidic input servovalve consisting of a first-stage fluidic amplifier cascade and a second-stage spool valve offers promise for fluid control systems that require power actuation. This study consists of a review of first-stage fluidic amplifier usage to date and the basic features of first-stage torque motor flapper-nozzle valve hydraulic amplifiers and first-stage hydraulic fluidic amplifiers. It also includes a comparison of the torque motor flapper-nozzle valve and fluidic amplifier in terms of input power level, output power, frequency response, leakage flow, temperature sensitivity, and reliability. Present trends in fluidic input servovalve development are also discussed.

2. FIRST-STAGE FLUIDIC AMPLIFIER USAGE

Torque motor flapper-nozzle hydraulic amplifiers are highly developed and have been used as first-stage amplifiers in two-stage electrohydraulic servovalves for nearly three decades. As early as 1973, fluidic amplifiers were used as first-stage amplifiers in two-stage servovalves. A number of efforts will now be discussed in the development of a two-stage fluidic input servovalve consisting of a first-stage fluidic amplifier and a second-stage spool.

Three efforts to develop a two-stage fluidic input servovalve using a first-stage fluidic amplifier [1-3] were sponsored by AVRADCOM as part of its servoactuator development program. The program objective was to develop reliable, low-cost servoactuators for use in hydrofluidic stability augmentation systems (HYSAS). First-stage fluidic amplifiers were used in the servovalves to increase reliability and reduce the cost of servoactuators. Data were not reported on input power level, output power, frequency response, and leakage flow for these programs using first-stage fluidic amplifiers. However, the results of these development programs will be presented.

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[1]*H. C. Kent and J. R. Sjolund, Hydrofluidic Servoactuator Development, Honeywell, Inc., Minneapolis, MN, United States Army Air Mobility Research and Development Laboratory TR-73-12 (May 1973).*

[2]*J. R. Sjolund, Hydrofluidic Servoactuator Development, Honeywell, Inc., Minneapolis, MN, United States Army Air Mobility Research and Development Laboratory TR-76-25 (September 1976).*

[3]*J. O. Hedeen, Investigation of a Low-Cost Servoactuator for HYSAS, Honeywell, Inc., Minneapolis, MN, United States Army Research and Technology Laboratories TR-78-30 (July 1978).*

The first development program was completed in 1973.[1] A schematic of the servoactuator used is shown in figure 1. The fluidic servovalve portion of the servoactuator, shown within dashed lines, consists of a first-stage fluidic amplifier cascade (three stages), which drives a second-stage spring-centered spool valve. In this two-stage, open loop, flow-control servovalve, actuator position is fed back to the input of the first-stage cascade by a spool valve feedback transducer. The input signal from the control system is amplified by a three-stage, fluidic amplifier cascade before it is applied to the servovalve. A development model was built and tested. The unit was functional; however, it was noisy, ±8 percent of full output, and it had gain variation with fluid temperature change. The second program was completed in 1976.[2] The servoactuator used in this program is shown in figure 2. The fluidic servovalve portion of the servoactuator, shown within the dashed lines, consists of a first-stage fluidic amplifier and a second-stage spring-centered spool valve. This is a two-stage, open loop, flow control servovalve. Actuator position is fed back by a flapper-nozzle feedback transducer. The input signal from the control system is amplified by a single-stage fluidic preamplifier before it is applied to the servovalve. This program was a follow-on to the 1973 program. A development model was built and tested; its response was slightly lower than the first model, and it had gain variation with fluid temperature change.

The third program was completed in 1978.[3] Schematics of the servoactuators used are shown in figures 3 and 4. The fluidic servovalve portion of the servoactuator, shown within the dashed line (fig. 3), consists of a first-stage fluidic amplifier cascade (two stages) and a second-stage spring-centered spool valve. Servovalve load flow is fed back through orifice resistors to the input of the first-stage fluidic amplifier cascade. This is a two-stage, closed-loop, flow-control servovalve. The fluidic servovalve portion of the servoactuator, shown within the dashed lines (fig. 4), consists of a first-stage fluidic amplifier cascade (two stages) and a second-stage spool valve. First-stage fluidic amplifier output flow used to drive the second-stage spool valve is fed back through orifice resistors to the amplifier input. This is a two-stage, closed-loop, pressure control

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[1]*H. C. Kent and J. R. Sjolund, Hydrofluidic Servoactuator Development, Honeywell, Inc., Minneapolis, MN, United States Army Air Mobility Research and Development Laboratory TR-73-12 (May 1973).*

[2]*J. R. Sjolund, Hydrofluidic Servoactuator Development, Honeywell, Inc., Minneapolis, MN, United States Army Air Mobility Research and Development Laboratory TR-76-25 (September 1976).*

[3]*J. O. Hedeen, Investigation of a Low-Cost Servoactuator for HYSAS, Honeywell, Inc., Minneapolis, MN, United States Army Research and Technology Laboratories TR-78-30 (July 1978).*

servovalve. Breadboard models were assembled and tested. Both the flow control and pressure control servovalves were evaluated. These units had lower response and stiffness than the first unit (fig. 1), and they had gain and null change with fluid temperature changes.

From figures 1 to 4 it is seen that each development program involved a spring-centered, second-stage spool, flow-control servovalve. The third program also involved a pressure control servovalve. Each development effort demonstrated the feasibility of using the first-stage fluidic amplifier, second-stage spool servovalve connfiguration; however, fluidic amplifier gain change with temperature was a problem in each effort.

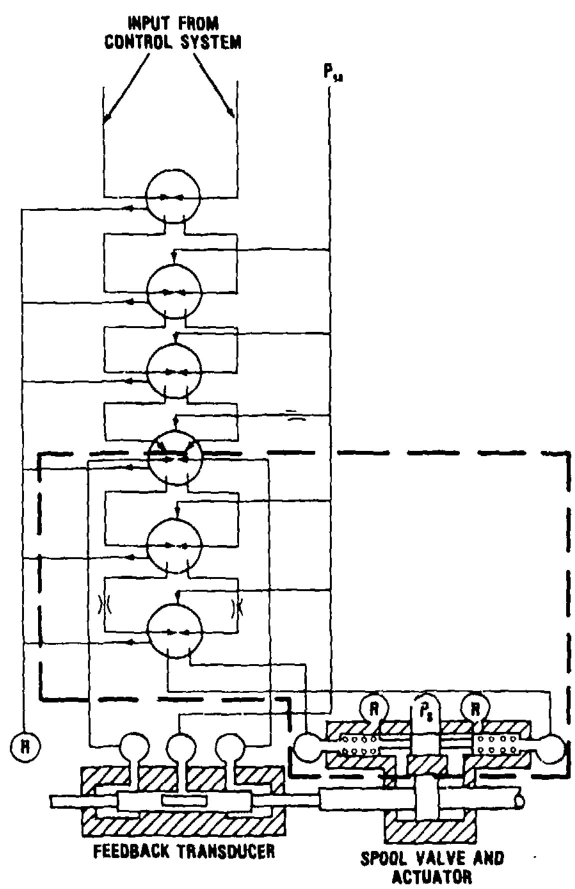


Figure 1. Fluidic servoactuator schematic with two-stage, flow-control servovalve--1973 program. *(from H. C. Kent and J. R. Sjolund, Honeywell, Inc., Minneapolis, MN, United States Army Air Mobility Research and Development Laboratory TR-73-12, May 1973.)*

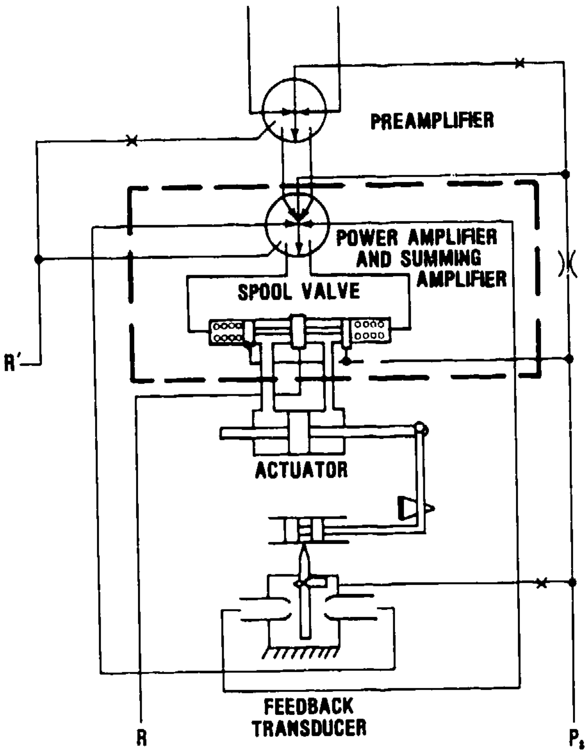


Figure 2. Fluidic servoactuator schematic with two-stage, flow-control servovalve--1976 program. *(From J. R. Sjolund, Honeywell, Inc., Minneapolis, MN, United States Army Air Mobility Research and Development Laboratory TR-76-25, September 1976.)*

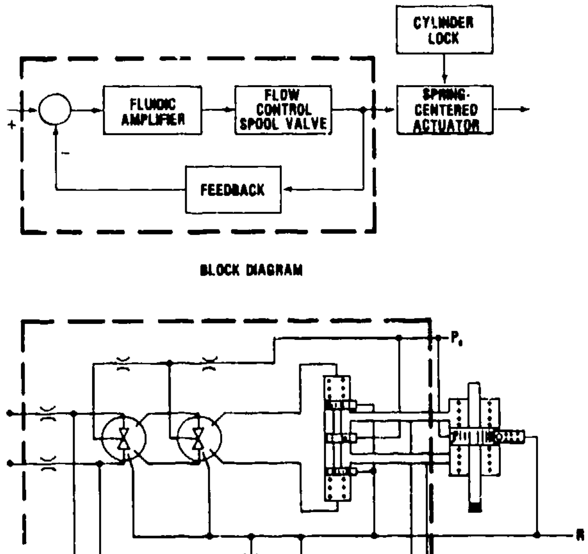


Figure 3. Fluidic servoactuator containing two-stage, flow-control servovalve with pressure feedback. *(Schematic from J. O. Hedeen, Honeywell Inc., St. Louis Park, MN, United States Army Research and Technology Laboratories TR-78-30, July 1978.)*

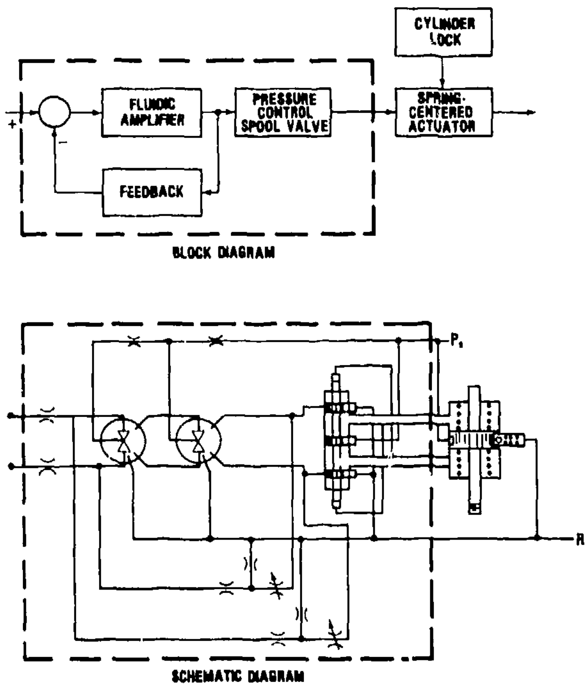


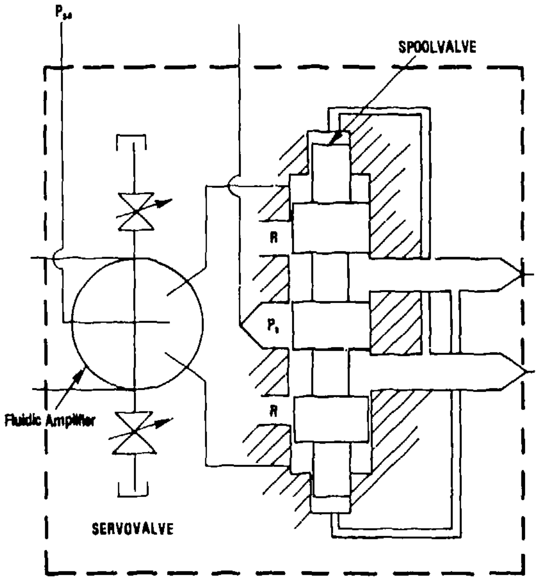
Figure 4. Fluidic servoactuator containing two-stage, pressure control servovalve with pressure feedback. *(Schematic from J. O. Hedeen, Honeywell, Inc., St. Louis Park, MN, United States Army Research and Technology Laboratories TR-78-30, July 1978.)*

HDL sponsored a study of a single-axis hydrofluidic stabilization system, a portion of which involved the development of a two-stage, fluidic input, pressure control servovalve with a first-stage fluidic amplifier and a second-stage spool valve. The servovalve schematic is shown in figure 5 (p. 10). This is an open loop, pressure control servovalve. The unpublished results of the development and testing of the servovalve indicate that this servovalve concept is feasible, although during system tests using maximum loop gain, considerable high-frequency ringing occurred that was associated with the servovalve. Performance was considered somewhat marginal under this condition. It was concluded that fluidic system temperature insensitivity could be obtained over the 40 to 180 F temperature range (~ 4 to ~ 26 C), where no more than four amplifiers are used, by operating the fluidic amplifiers in the turbulent regime. Additional studies of this servovalve configuration are in progress. These studies are also sponsored by AVRADCOM and HDL.

3. FIRST-STAGE TORQUE MOTOR FLAPPER-NOZZLE VALVE

The first-stage amplifier used in two-stage electrohydraulic servovalves is usually of the type shown in figure 6. This type of valve is also called a hydraulic amplifier. A study of its characteristics will provide a useful reference for comparing first-stage fluidic amplifier cascades. The torque motor converts a low-

power-level electrical input signal into a mechanical torque. The torque applied to the flapper arm controls the output flow from the double nozzle or four-way flapper valve. Some of the important characteristics of the hydraulic amplifier are relatively low-level input signal power, high-output power, good frequency response, relatively low leakage flow, relative temperature insensitivity, and reliability. Each of these characteristics will now be considered.



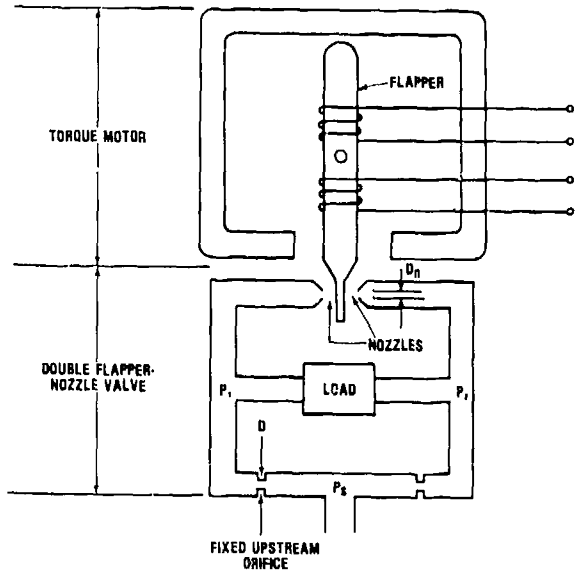


Figure 5. Two-stage fluidic pressure control servovalve with first-stage fluidic amplifier and second-stage spool valve.

Figure 6. Hydraulic amplifier schematic showing torque motor, double flapper-nozzle valve.

3.1 Input Signal Power

A survey of commercially available two-stage servovalves used in aerospace applications shows that input signal power, W_1, is in the 50- to 200-mW range.[4] The torque motor converts the low-level electrical signals into torque on the flapper. Thus, the torque motor essentially acts as an electrical-to-mechanical interface; there is no power gain. The torque motor drives the flapper of the double-nozzle flapper valve.

3.2 Output Power

The flow curve of the flapper-nozzle valve output pressure indicates its available output power. A typical flapper-nozzle output curve for maximum flapper deflection is given in figure 7. This curve was generated with the mathematical approximation shown below. This is an approximation of the actual flapper-nozzle valve output pressure (P_m) flow (Q_m) relationship; the actual relationship is complicated and not useful.[5]

$$P_m = \frac{1}{25}\left[20 - 28\, Q_m^2 - 4\, Q_m \sqrt{5 - Q_m^2}\right] , \tag{1}$$

where

P_m = nondimensional load pressure = p_m/P_s,

P_s = flapper supply pressure, Pa,

p_m = actual load pressure, Pa,

Q_m = nondimensional load flow = $q_m/g_f\sqrt{P_s}$,

g_f = $C_D \frac{\pi}{4} D^2 \sqrt{\frac{2}{\rho}}$, $\sqrt{m^7/kg}$,

C_D = orifice discharge coefficient,

ρ = fluid density, kg/m^3

q_m = actual load flow, m^3/s, and

D = flapper fixed orifice diameter, m.

Nondimensional output power, W, is

$$W = P_m Q_m \quad . \tag{2}$$

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[4] *M. Guillon, Hydraulic Servo Systems Analysis and Design, Plenum Press, New York (1969).*

[5] *J. F. Blackburn et al, Fluid Power Control, MIT Press and John Wiley and Sons (1960).*

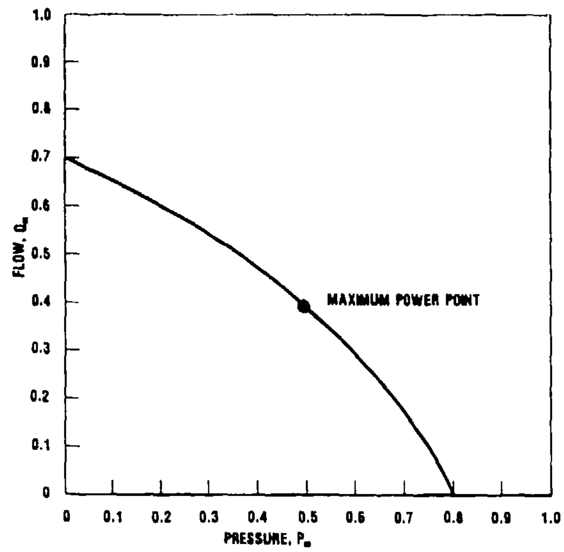

Figure 7. Hydraulic amplifier output characteristic.

Substituting the expression for P_m, equation (1), into equation (2) gives $W = f(Q_m)$. Differentiating this expression with respect to Q_m and setting it equal to zero gives an equation in Q_m which can be solved for maximum output power. Following this procedure gives $Q_m = 0.3956$ and $P_m = 0.4854$ for maximum power output (see fig. 7). The expression for maximum flapper output power, W_0 is

$$W_0 = 0.1920\ g_f\ (P_s)^{3/2} \quad . \tag{3}$$

The maximum power gain, G_F, of the first-stage hydraulic amplifier is

$$G_F = \frac{W_0}{W_I} = \frac{0.1920 g_f (P_s)^{3/2}}{W_I} \quad . \tag{4}$$

For a nominal flapper diameter of 0.25 mm (0.01 in.), discharge coefficient of 0.65, supply pressure 10.35×10^6 Pa (1500 psi) and W_I = 100 mW, the first-stage power gain G_F is approximately 100.

3.3 Frequency Response

The hydraulic amplifier frequency response is determined by the torque motor input circuit and flapper valve dynamics. The amplifier transfer function is

$$T = \frac{K_A}{\left(\tau_m s + 1\right)\left[\left(s/\omega_n\right)^2 + \left(2\zeta/\omega_n\right) s + 1\right]} \quad , \tag{5}$$

where

ω_n = flapper-nozzle natural frequency, 1/s,
ζ = flapper-nozzle damping ratio,
s = Laplace transform variable, 1/s,
τ_m = torque motor time constant, 1/s, and
K_A = hydraulic amplifier gain constant, $m^5/(ns)$.

However, first-stage hydraulic amplifiers and second-stage spool dynamics are negligible in terms of the overall servovalve response. The overall servovalve response is primarily determined by the amount of first-stage output power available to drive the second-stage spool valve. The spool drive flow, Q_p, required to give rated[6] servovalve output flow at frequency, f, is

$$Q_p = A_s X_s \omega \quad , \tag{6}$$

where

ω = $2\pi f$, 1/s,
A_s = spool end area, m^2, and
X_s = maximum spool displacement, m.

Rated flow is essentially maximum servovalve output flow from the second-stage spool valve for a no-load (or zero-load pressure drop) condition. Maximum output flow can be achieved only for maximum spool displacement. Thus, X_s is used in equation (6). If the first stage can supply the required spool drive flow, Q_p, then overall servovalve response will be determined by minimal first- and second-stage dynamics and will be very fast. If first-stage output flow is less than Q_p, servovalve response will be reduced accordingly.

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[6] *W. J. Thayer, Specification Standards for Electrohydraulic Flow Control Servovalves, Moog Technical Bulletin 117, Moog, Inc., Proner Airport, East Aurora, New York (July 1959, Revised June 1962).*

Maximum first-stage or hydraulic amplifier output flow, Q_F, is[5]

$$Q_F = \frac{\pi}{4} D^2 C_D \sqrt{\frac{P_s}{\rho}} \quad . \tag{7}$$

Equating the maximum output flow from the hydraulic amplifier to the required spool drive flow gives

$$\omega A_s X_s = \frac{\pi}{4} D^2 C_D \sqrt{\frac{P_s}{\rho}} \quad . \tag{8}$$

Solving for ω gives

$$\omega = \left[\frac{C_D \left(\frac{\pi}{4}\right) \sqrt{\frac{P_s}{\rho}}}{A_s X_s} \right] D^2 \quad . \tag{9}$$

Thus, for a given spool end area, A_s, maximum spool displacement, X_s, and supply pressure, P_s, servovalve response, ω, is determined by the diameter, D, of the flapper orifices.

3.4 Leakage Flow

Servovalve null leakage flow, which is nominally less than 10 percent of the rated or no-load flow from the servovalve,[7] will now be considered. Null leakage flow consists of first-stage leakage flow or tare flow and second-stage spool leakage flow. Tare flow, Q_e, is of interest here, and for a double nozzle flapper[8] it is

$$Q_e = 2\, C_D \left(\frac{\pi}{4}\right) D^2 \sqrt{\frac{P_s}{\rho}} = 2\, Q_F \quad . \tag{10}$$

Thus, it can be seen from equations (9) and (10) that high servovalve frequency response is achieved at the expense of leakage flow. Tare flow is twice the maximum output flow from the hydraulic amplifier.

3.5 Temperature Sensitivity

The flapper-nozzle valve is relatively temperature insensitive. Two aspects of the sensitivity of double flapper-nozzle

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[5] *J. F. Blackburn et al, Fluid Power Control, MIT Press and John Wiley and Sons (1960).*

[7] *H. C. Morse, Electrohydraulic Servomechanisms, McGraw-Hill Book Co., New York (1963).*

[8] *H. E. Merritt, Hydraulic Control Systems, John Wiley and Sons (1967).*

valve fluid temperature will be considered--flow gain and null shift. The linearized flow gain of the double flapper-nozzle valve (fig. 6) is[8]

$$\left.\frac{\Delta Q_L}{\Delta X}\right|_{\Delta P_L = 0} = C_{Df} D_N \left(\frac{P_s}{\rho}\right)^{1/2} ,$$

where C_{Df} is the discharge coefficient of the annular curtain area between the nozzle and flapper, and D_n is the nozzle diameter. The discharge coefficient C_{Df} is relatively insensitive to changes in Reynolds number[9] (here due to temperature) with a value of approximately 0.63. Fluid density, ρ, is also essentially constant for changes in fluid temperature; therefore, flapper flow gain is essentially constant and not a function of temperature. The second consideration is flapper valve null shift due to changes in fluid temperature. The null shift of the flapper valve is temperature sensitive. However, most electrohydraulic servovalves use a double nozzle flapper in the first stage because the two nozzles make this configuration relatively immune to changes in its null shift due to resulting changes in supply pressure and/or temperature.[8] Two-stage electrohydraulic servovalves using a first-stage flapper-nozzle valve typically provide normal performance over the 0- to 165-F temperature range, and limited response to input commands at -65 F.

3.6 Reliability

The reliability of hydraulic amplifiers has been continually improved over the years.[10] The first major improvement was the use of a double nozzle-flapper valve rather than the previously used single-nozzle-flapper valves. This change reduced null offsets caused by environment (temperature, pressure, etc.). The second improvement was isolation of the torque motor from the hydraulic oil. Both of these improvements increased amplifier reliability. However, in spite of these improvements, the hydraulic amplifier still has two failure modes that lead to actuator "hardover," which can result in a catastrophic failure. A hardover actuator failure is the uncontrolled driving of the actuator to one or the other extremes at maximum power. Actuator hard-over can be caused by electrical short circuits in the torque motor circuit and by the sudden blockage of one of the orifices (due to large-particle contamination) in the flapper-nozzle valve. In both cases,

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[8] *H. E. Merritt, Hydraulic Control Systems, John Wiley and Sons (1967).*

[9] *A. Lichtarowicz and E. Markland, Calculation of Potential Flow with Separation in a Kight-Angled Elbow with Unequal Branches, J. Fluid Mech., Cambridge University Press, 17, Pt 4 (December 1963).*

[10] *R. H. Maskrey and W. J. Thayer, A Brief History of Electrohydraulic Servomechanisms, American Society of Mechanical Engineering Journal of Dynamic Systems Measurement and Control (June 1978).*

maximum hydraulic amplifier output power is applied to one end of the spool, resulting in actuator hardover at maximum power. In many aircraft applications, servoactuators control aerodynamic surfaces of aircraft. In the event of a hardover failure, the aerodynamic surface is forced to one extreme position and the pilot has little or no control of the surface. In this instance, hardover failure can result in a catastrophe.

4. FIRST-STAGE FLUIDIC AMPLIFIER

Two-stage, fluidic input servovalves are necessary components of fluidic control systems requiring power actuation. These valves have been the subject of continuing study since 1962. The objective has been to develop a valve which would (1) accept low-level, fluidic input signals, (2) have dynamic performance (frequency response) comparable to two-stage electrohydraulic servovalves, (3) be low in cost, and (4) be very reliable. Many fluidic input servovalve configurations have been studied (see appendix A). Most of the first-stage amplifier configurations have involved the use of metallic bellows that act upon a level bar so that pressure signals are converted into torque. Servovalves using a first-stage bellows configuration have been successfully demonstrated.[11] Nevertheless, the use of bellows in the first stage limits the valve's potential because the bellows (1) are costly, (2) are very susceptible to rupture due to overpressure, and (3) limit overall frequency response for high-performance servovalves.[12] Therefore, there has been renewed interest in fluidics for first-stage amplifiers. The fluidic amplifier has no moving parts, low production cost, and high reliability. The following amplifier characteristics will be considered: (1) input signal power level, (2) output power, (3) frequency response, (4) leakage flow, (5) temperature sensitivity, and (6) reliability.

4.1 Input Signal Power

Control systems with large dynamic ranges are of interest here; therefore, laminar proportional amplifiers (LPA's) will be considered. In general, LPA's have extremely low thresholds and can be driven easily by 100-mW (or 0.1 W) input signals.

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[11] *L. J. Banaszak and W. M. Posingies, Hydrofluidic Stability Augmentation System (HYSAS) Operational Suitability Demonstration, United States Army Air Mobility Research and Development Laboratory TR-77-31 (October 1977).*

[12] *D. Lee and D. N. Wormley, Hydraulic Signal-Processing Amplifier Performance in Position Control Systems, Massachusetts Institute of Technology, Cambridge, MA, Harry Diamond Laboratories, HDL-CR-77-191-1 (December 1977).*

4.2 Output Signal Power

Fluidic amplifier output power is indicated by its output pressure flow curve. Figure 8 gives a typical fluidic amplifier output curve where the amplifier nozzle width, b_s, height, h, and supply pressure, P_{sa}, are 5×10^{-4}m, 2.05×10^{4}m, and 110 kPa, respectively. Fluidic amplifier output curves (fig. 8) have an orifice characteristic[13] and can be described by equation (11).

$$Q_n = \sqrt{1 - 2P_n} \quad , \tag{11}$$

where

P_n = nondimensional output or load pressure (p_n/P_{sa}),

P_{sa} = fluidic amplifier supply pressure, Pa,

p_n = actual output pressure, Pa,

Q_n = nondimensional output flow q_n/Q_{sa},

Q_{sa} = fluidic amplifier supply flow, m^3/s, and

q_n = actual output flow, m^3/s.

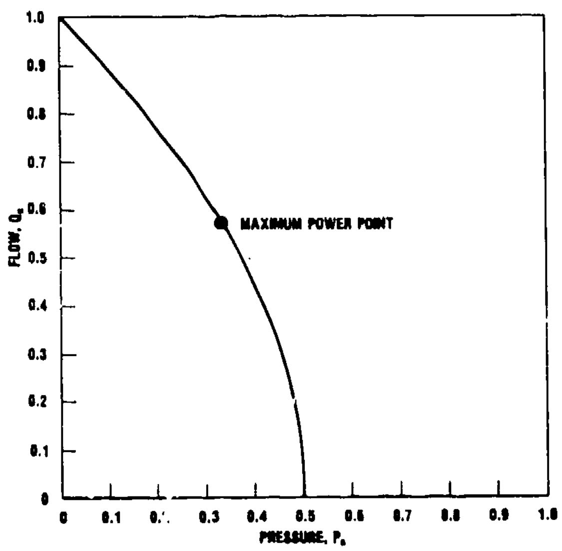

Figure 8. Fluidic amplifier output characteristic.

[13] *J. M. Kirshner, Fluid Amplifiers, McGraw-Hill (1966).*

Nondimensional output power W^1 is given as

$$W^1 = P_n Q_n \quad . \tag{12}$$

Substituting the expression for Q_n, equation (11), into equation (12) gives $W^1 = f(P_n)$. Differentiating $W^1 = f(P_n)$ with respect to P_n, and setting this expression equal to zero gives an equation in P_n for the maximum power condition. Following this procedure gives

$$P_n = \frac{1}{3} \text{ and } Q_n = \sqrt{\frac{1}{3}} \text{ (see fig. 8), or}$$

$$p_n = \frac{1}{3} P_{sa} \text{ and } q_n = \sqrt{\frac{1}{3}}\; Q_{sa} \quad , \tag{13}$$

where

$$Q_{sa} = g_a \sqrt{P_{sa}} \text{ , } m^3/s \quad ,$$

$$g_a = C_{Da} A_{sa} \sqrt{\frac{2}{\rho}} \text{ , } \sqrt{m^7/kg} \quad ,$$

C_{Da} = amplifier supply nozzle discharge coefficient, and

A_{sa} = amplifier supply nozzle area ($b_s h$), m^2.

Therefore, $p_n = \frac{1}{3} P_{sa}$, $q_n = \sqrt{\frac{1}{3}}\; g_a \sqrt{P_{sa}}$, and maximum output power,

$$W_0^1 = 0.1924\; g_a \left(P_{sa}\right)^{3/2} \quad . \tag{14}$$

This expression for maximum amplifier output power is very similar to equation (3) for maximum output power for the flapper-nozzle valve. The maximum power output ratio, $\psi = W_0^1/W_0$, is

$$\psi = 1.002\, \frac{g_a\left(P_{sa}\right)^{3/2}}{g_f\left(P_s\right)^{3/2}} \quad . \tag{15}$$

If the amplifier supply pressure P_{sa} is made equal to the flapper-nozzle valve supply pressure, P_s, and $C_{Da} = C_D$, then $g_a/g_f = (b_s h)/(D^2/4)$, and ψ becomes

$$\psi = 1.002 \frac{(b_s h)}{(\pi D^2/4)} \quad . \tag{16}$$

Thus, the power ratio essentially becomes the ratio of amplifier supply nozzle area to flapper-nozzle orifice area. The nominal flapper orifice diameter has a maximum value of 2.5×10^{-4}m (0.01 in.), and $(b_s h)$ is $(5 \times 10^{-4}\text{m} \cdot 2.05 \times 10^{-4}\text{m})$ for this type of application. For these nominal geometries, the maximum output power ratio is

$$\psi = \frac{W_0^1}{W_0} = (1.002)(2.08) = 2.09 \quad . \tag{17}$$

Thus, the fluidic amplifier operated at the same supply pressure as the flapper-nozzle valve nominally has twice the maximum output power. Maximum fluidic amplifier power gain, G_A, is

$$G_A = \frac{W_O^1}{W_I} = \frac{0.1924 g_a (P_{sa})^{3/2}}{W_I^1} \quad . \tag{18}$$

From equations (4), (17), and (18) G_A has a value

$$G_A = 2.09\ G_F = 200 \quad ,$$

where W_I^1 is 100 mW. However, since the self-staged power gain of fluidic amplifiers is approximately 15, two or more amplifiers in series will be required to achieve a power gain of 200.

4.3 Frequency Response

Frequency response of a typical LPA using hydraulic oil is shown in figure 9.[12] Again, the overall servovalve frequency response is basically determined by the amount of first-stage output power

[12] *D. Lee and D. N. Wormley, Hydraulic Signal-Processing Amplifier Performance in Position Control Systems, Massachusetts Institute of Technology, Cambridge, MA, Harry Diamond Laboratories, HDL-CR-77-191-1 (December 1977).*

available to drive the second-stage spool valve. The required spool drive flow for servovalve response, ω, is given as $\omega A_s X_s$. The amplifier output characteristic, the lower curve in figure 10, gives the output flow and pressure that it can provide. If the point describing the required spool drive flow, Q_p, and pressure, P_p, were plotted on the output characteristic of a first-stage amplifier cascade and it fell to the right of the curve, then the amplifier could not drive the spool to obtain maximum servovalve response at frequency, ω. In this instance, another amplifier could be added in parallel to the last stage of the amplifier cascade. The amplifier cascade output characteristic would now be described by the upper curve in figure 10. Since point (Q_p, P_p) is to the left of the upper curve, the amplifier cascade can drive the spool to obtain maximum servovalve response at frequency, ω (that is, servovalve response limited or determined solely by first-stage amplifier and second-stage spool dynamics).

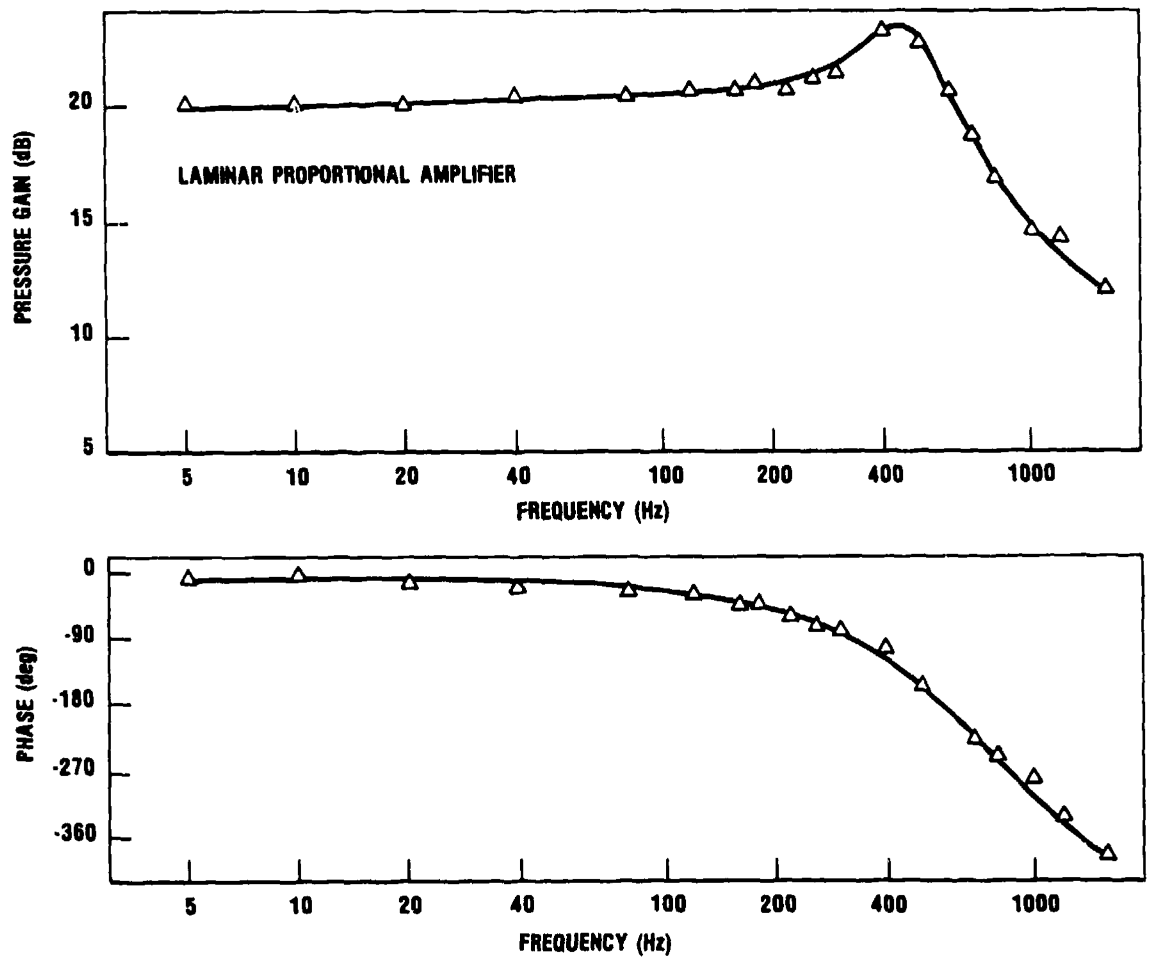

Figure 9. Single-stage fluidic amplifier block load, pressure-gain frequency response characteristics. *(Data from D. Lee and D. N. Wormley, Massachusetts Institute of Technology, Harry Diamond Laboratories, HDL-CR-77-191-1, December 1977.)*

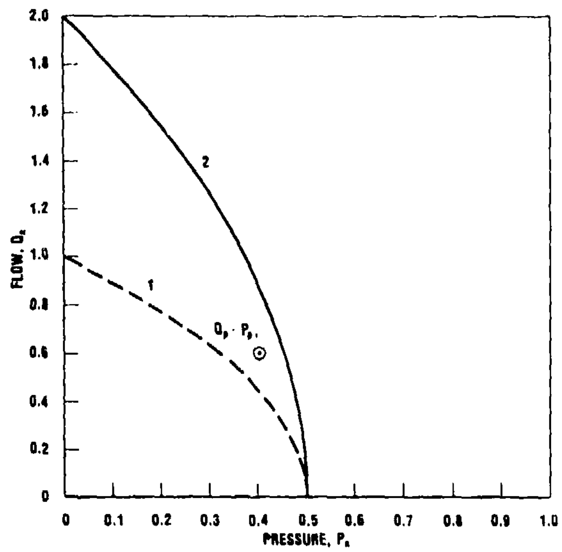

Figure 10. Fluidic amplifier output characteristics for single amplifier (1) and two amplifiers in parallel (2).

The fluidic amplifier and flapper-nozzle output curves are plotted in figure 11. The fluidic amplifier output pressure flow curve (eq (20),

$$q_n/Q_{sa} = \sqrt{1 - 2\left(P_n/P_{sa}\right)} \quad , \tag{20}$$

has been written in terms of the flapper pressure-flow characteristics to give

$$q_n/2.08\ g_f\sqrt{P_s} = \sqrt{1 - 2\left(P_n/P_s\right)} \quad , \tag{21}$$

where $P_{sa} = P_s$ and $g_a = 2.08\ g_f$. The fluidic amplifier has higher maximum output power than the flapper-nozzle valve, as previously noted; however, if the required spool drive pressure flow point is plotted on figure 11 and it falls to the right of the intersection of the curves, then the flapper will be able to supply more output power than the fluidic amplifier to drive the spool. If it falls to the left of the intersection, then the fluidic amplifier can supply more output power to drive the spool.

The magnitude of the required spool drive pressure and flow on the first-stage output characteristic is important in terms of the type

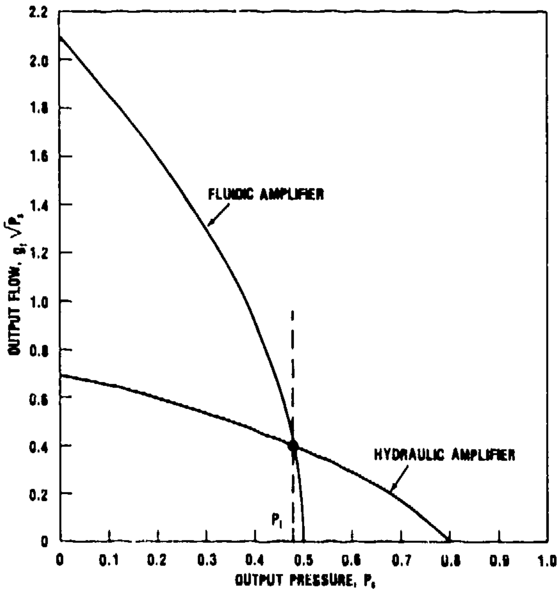

Figure 11. Hydraulic and fluidic amplifier output characteristics.

of spool position feedback that is used. Second-stage spool position feedback may be achieved in three basic ways.

1. By placement of stiff springs at the spool ends that act to center the spool against the pressure differential caused by the first stage. This type of servovalve configuration can be seen in figures 1, 2, and 5. In each case the first stage is a fluidic amplifier, which drives a spring-centered, second-stage spool valve. The servovalves shown in figures 1, 2, and 5 are two-stage, open loop servovalves.

2. By direct position feedback, as shown in the two-stage, electrohydraulic servovalve of figure 12. The first stage is a hydraulic amplifier which has direct spool position feedback, and the second stage is a spool valve. This is a two-stage, closed loop servovalve.

3. By use of a mechanical spring to convert spool position to a force signal which is fed back to the torque motor. This type of servovalve configuration can be seen in the two-stage, electrohydraulic servovalve, figure 13. The first stage is a hydraulic amplifier, which drives a second-stage spool valve. This is a two-stage, closed loop servovalve.[8]

[8] *H. E. Merritt, Hydraulic Control Systems, John Wiley and Sons (1967).*

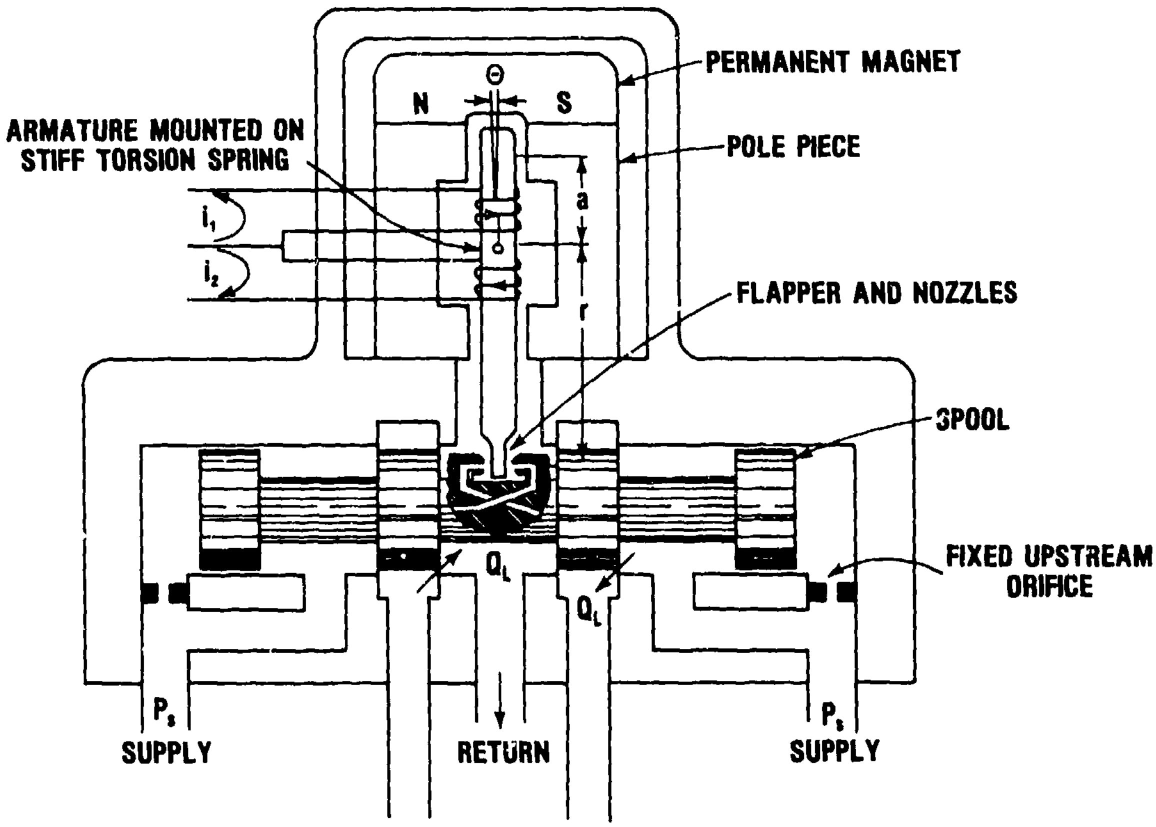

Figure 12. Two-stage, flow-control electrohydraulic servovalve with direct feedback. *(Schematic from H. E. Merritt, John Wiley and Sons, 1967.)*

The use of stiff springs at the spool ends provides an open loop type of servovalve. Centering springs with very high stiffness (large spring constants) are used so that the force required to move the springs is much greater than the flow forces on the spool. In this way the first-stage amplifier can drive the second-stage spool in a linear, open loop manner.[14] However, the use of stiff centering springs necessitates a very high first-stage output pressure. Therefore, if the required spool drive pressure for a given application were greater than P_l (fig. 11), then the flapper-nozzle first stage could provide greater output power and thus higher overall servovalve frequency response than

[14]*R. Deadwyler and F. M. Manion, Design Considerations for Improved Fluidic Input Servovalve Performance, Harry Diamond Laboratories HDL-TM-79-4 (April 1979).*

the first-stage fluidic amplifier. If a first-stage fluidic amplifier were used in this instance, a number of parallel amplifiers would be required. The penalty here is additional leakage flow. An alternative would be to use less stiff centering springs; the penalty here is reduced linearity.

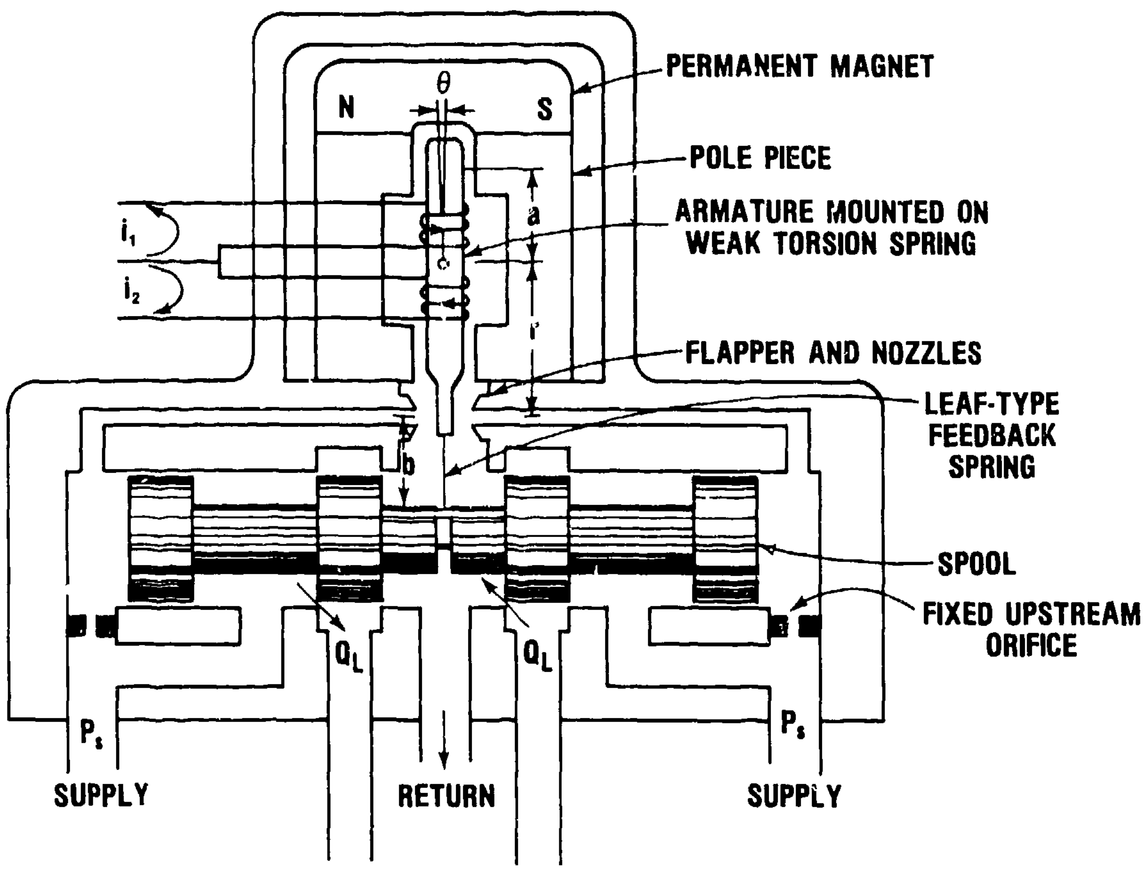

Figure 13. Two-stage, flow-control electrohydraulic servovalve with mechanical feedback. *(Schematic from H. B. Merritt, John Wiley and Sons, 1967.)*

Direct position and mechanical spring position feedback are similar in that both provide a closed loop, position control servovalve. A two-stage closed loop servovalve is desirable because it allows for the best overall servovalve control. Two of the fluidic servovalves described in section 2 are two-stage, closed loop servovalves (see fig. 3 and 4), which have fluidic resistor feedback circuits. The difficulty with this feedback technique is feedback circuit temperature sensitivity. Spring position feedback has had the

greatest impact on servovalve development--and by far the widest use;[4,15,16] therefore, only spring position feedback will be discussed. Spring position feedback (fig. 13) has been fully described.[8] Its advantage is that much larger spool driving pressures (or forces) are available to drive the spool than are required. There is essentially no pressure drop across the spool, so that spool position is controlled by spool drive flow.[16] Using spring feedback, the required spool drive pressure flow point will generally be to the left of the intersection of the fluidic amplifier and hydraulic amplifier curves (fig. 11), so that a fluidic amplifier first stage could provide greater output power (higher overall servovalve frequency response) than a first-stage hydraulic amplifier. Needless to say, mechanical spring position feedback cannot be used with fluidic amplifiers as such, because fluidic amplifiers with no moving parts cannot accept mechanical inputs. However, in order to take advantage of the potential offered by first-stage fluidic amplifiers and second-stage spool valves with mechanical feedback, fluidic-type amplifiers with moving parts have been developed (fig. 14, 15, and 16). The amplifier shown in figure 14 is called an electrofluidic transducer or pin amplifier.[17] The amplifier mechanism is the mechanical deflection of the supply jet by the pin so that a differential pressure is developed at the receiver ports.

The amplifier shown in figure 15 is called a fluidic deflector-jet amplifier.[18] The amplifier mechanism is the mechanical deflection of the supply jet by the deflector such that a differential pressure is developed at the receiver ports. The amplifier shown in figure 16 is

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[4] *M. Guillon, Hydraulic Servo Systems Analysis and Design, Plenum Press, New York (1969).*

[8] *H. E. Merritt, Hydraulic Control Systems, John Wiley and Sons (1967).*

[15] *D. C. Clark, Electronic Controls for Fluid Power, Hydraulics and Pneumatics (June 1978).*

[16] *W. J. Thayer, Transfer Functions for Moog Servovalves, Moog Technical Bulletin 103, Moog, Inc., Proner Airport, East Aurora, New York (December 1958, Revised January 1965).*

[17] *W. T. Harvey and J. W. Merritt, Electrofluidic Transducer (Pin Amplifier), U.S. Patent No. 3,638,671 (February 1972).*

[18] *Moog Catalog, No. 261, Moog Inc., Proner Airport, East Aurora, New York (n.d.).*

called an electro-fluidic transducer or amplifier.[19] The amplifier mechanism is the deflection of the supply nozzle such that the jet emanating from the nozzle develops a differential pressure at the receiver ports. This amplifier is very similar to the jet pipe valve.[5] Each of these amplifiers is electrically actuated by a torque motor and has been used as the first stage in two-stage electrohydraulic servovalves.[18,19] Two of the servovalves are shown in figures 17 and 18. The two-stage servovalve shown in figure 17 has a first-stage torque motor with a fluidic deflector-jet amplifier and a second-stage spool valve with mechanical feedback. This is a two-stage, closed loop servovalve. The two-stage servovalve shown in figure 18 has a first-stage torque motor and electro-fluidic amplifier and a second-stage spool valve with electrical feedback. This is a two-stage, closed loop servovalve. The use of fluidic-type amplifiers in these electrohydraulic servovalves attests to the important role fluidic elements play in the present commercial servovalve market. These fluidic applications also point to their potential for increased usage in future commercial applications where reliability will be a vital aspect of system design.

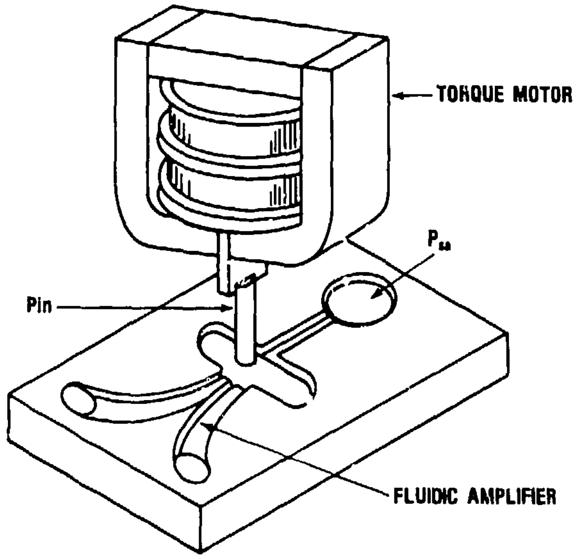

Figure 14. AiResearch Electrofluidic transducer or amplifier.

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[5] *J. F. Blackburn et al, Fluid Power Control, MIT Press and John Wiley and Sons (1960).*

[18] *Moog Catalog, No. 261, Moog Inc., Proner Airport, East Aurora, New York (n.d.).*

[19] *Robert Bosch Corporation, Electro-hydraulic Servovalve Series 0814-SMV2 and 0814-SMV3, 2800 South 25th Ave., Broadview, ILL (n.d.).*

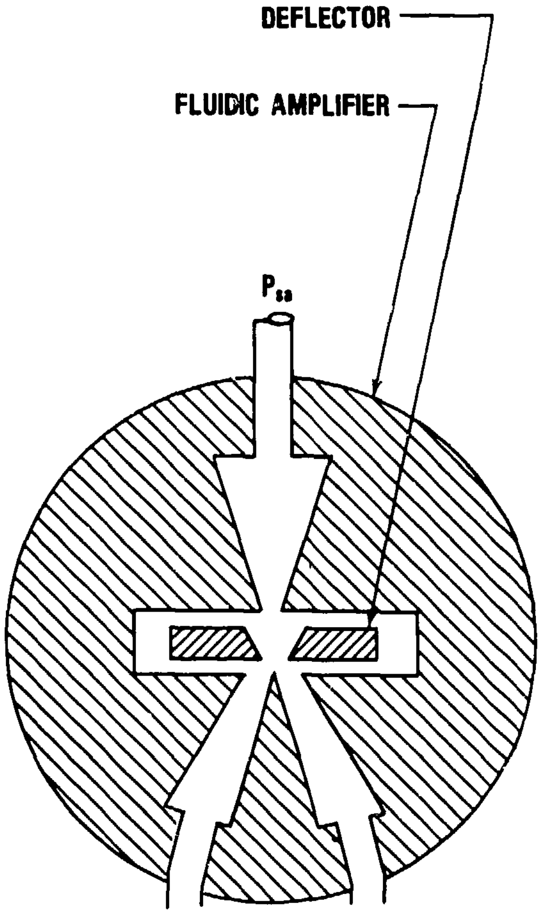
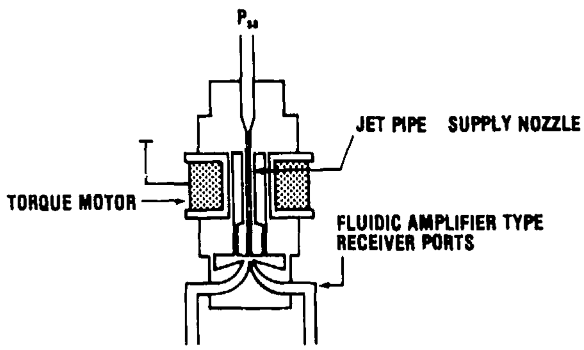


Figure 15. Moog deflector jet fluidic amplifier.

Figure 16. Bosch electrofluidic converter or amplifier (jet pipe).



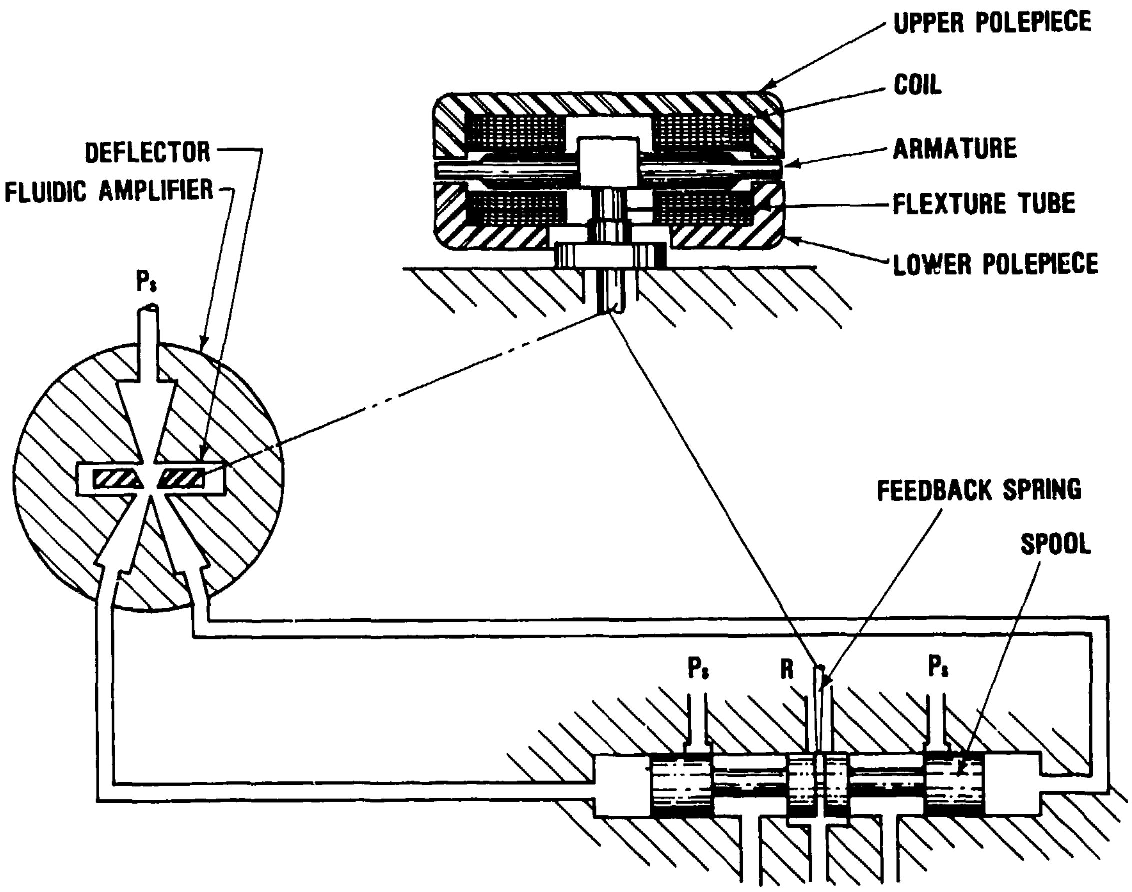


Figure 17. Two-stage, flow control electrohydraulic servovalve with mechanical feedback.

The two-stage servovalves shown in figures 17 and 18 have the best fluidic servovalve configuration. They have closed loop feedback control, which ensures the best overall servovalve control, the minimum number of fluidic amplifiers in the first stage (less parallel staging required) and a feedback technique that is relatively temperature insensitive, as indicated by servovalve specificaitons.[18,19] A two-stage fluidic-input servovalve can be developed that employs this configuration. It requires the development of an efficient pure fluid means of moving the pin, the deflector, or the nozzle in the moving-part fluidic-type amplifiers described in figures 14, 15, and 16.

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[18] *Moog Catalog, No. 261, Moog Inc., Proner Airport, East Aurora, New York (n.d.)*

[19] *Robert Bosch Corporation, Electro-hydraulic Servovalve Series 0814-SMV2 and 0814-SMV3, 2800 South 25th Ave., Broadview, ILL (n.d.).*

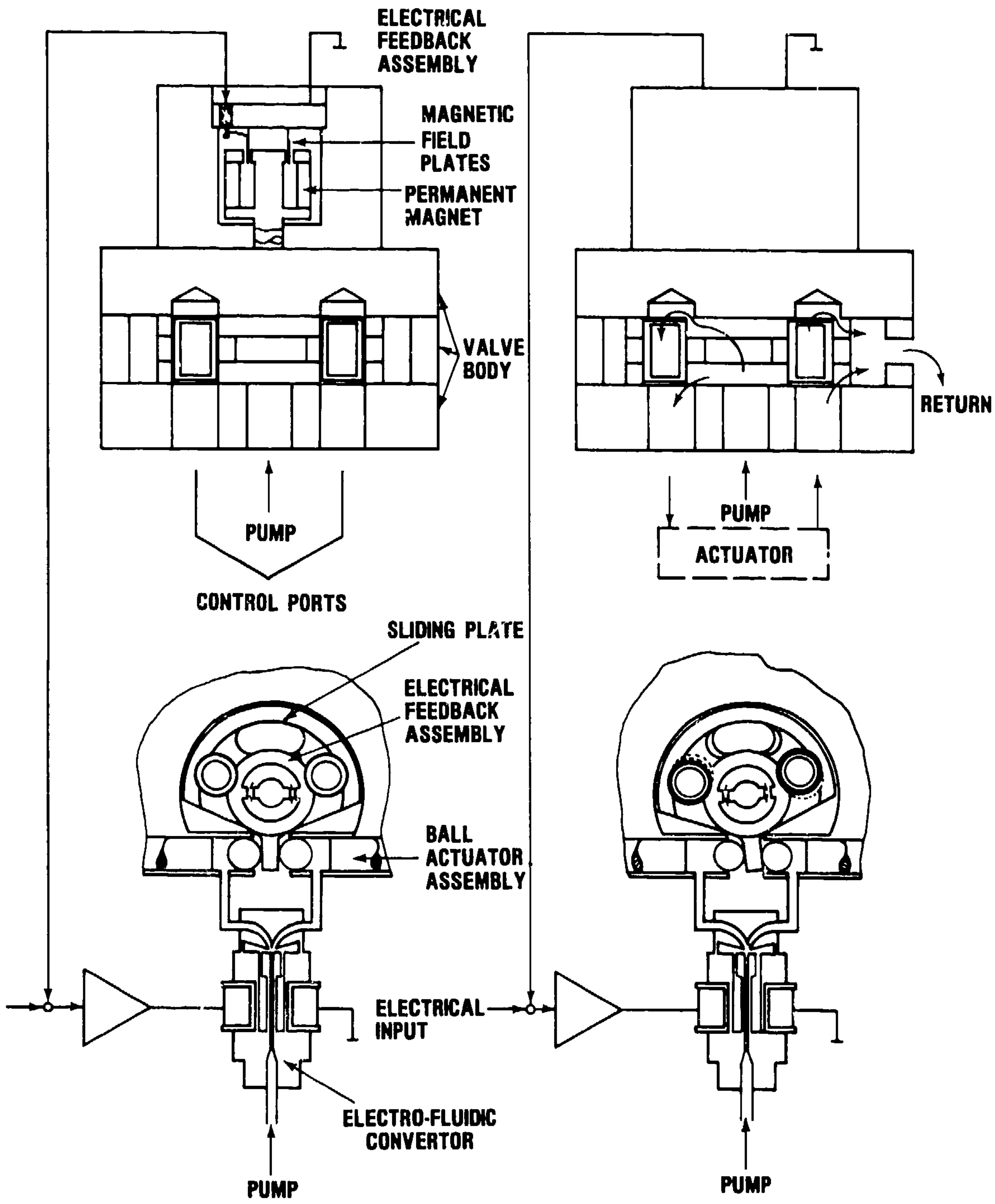

Figure 18. Two-stage, flow-control electrohydraulic servovalve with electrical feedback.

4.4 Leakage Flow

Tare flow for a first-stage fluidic amplifier cascade, Q_e, is the sum of the supply flows for each amplifier in the cascade. It is given as

$$Q'_e = \sum_{i=1}^{j} b_1 g_1 \sqrt{P_1} + b_2 g_2 \sqrt{P_2} + \ldots + b_j g_j \sqrt{P_j} \quad , \tag{22}$$

where P_i = amplifier supply pressure for the ith stage, Pa

$$g_i = C_{Di} A_i \sqrt{\frac{2}{\rho}},\ m^7/kg \quad .$$

C_{Di} = supply nozzle discharge coefficient of the ith stage,
A_i = supply nozzle area of the ith stage, m^2, and
b_i = number of parallel sections in the ith stage.

In general, fluid amplifier leakage flow is greater than a comparable flapper-nozzle valve, even if only one amplifier were used, because its power nozzle area is normally greater than the flapper orifice area. Thus,

$$Q'_e = 2.08\ g_f \sqrt{P_s} > Q_e = \sqrt{2}\, g_f \quad P_s \quad , \tag{23}$$

where

$C_{Da} = C_D$. The minimum first-stage fluidic amplifier leakage flow is typically 65 percent of the rated or no-load servovalve output flow.

4.5 Temperature Sensitivity

Fluidic amplifiers have gain and null temperature sensitivity as noted in section 2. The gain sensitivity of a first-stage fluidic amplifier cascade can be reduced by using the minimum number of amplifiers in the cascade. In addition, a number of temperature-compensation techniques[20,21] have been developed to reduce amplifier

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[20]G. Mon and H. Robinson, *Temperature Compensation of Laminar Proportional Amplifiers Using A Linear Resistor Bypass*, Harry Diamond Laboratories, HDL-TM-78-16 (September 1978).

[21]W. M. Posingies, *Advanced Fluidic Temperature Studies*, Honeywell, Inc., St. Louis Park, MN, United States Army Research and Technology Laboratories TR-78-33 (October 1978).

gain and null temperature sensitivity. However, most of these techniques reduce amplifier temperature sensitivity at the expense of higher leakage flow. First-stage fluidic amplifiers operating in the turbulent regime and moving-part fluidic amplifiers appear to be relatively temperature insensitive.

4.6 Reliability

Fluidic systems have a proven record of reliability as a result of operational suitability demonstrations[11] and commercial applications.[22] The key element in fluidic systems is the fluid amplifier. Its supply nozzle area is relatively large (compared to the flapper-nozzle orifice areas); therefore, it can operate with relatively high levels of fluid contamination. Moreover, if the supply nozzle should suddenly become blocked due to large-particle contamination, then the amplifier will fail neutrally; that is, neither amplifier output port will be driven to saturation. The control nozzle area may also be blocked due to large-particle contamination. A sudden blockage of one of the control nozzles will cause one of the output ports to be driven to saturation. The sudden blockage of an output port will lead to the same problem. However, the control nozzle areas and output port areas are usually larger than the supply nozzle area. Therefore, the probability of failures due to control nozzle and output port blockage is low. These facts increase the reliability of fluidic systems and reduce system filtration requirements.

Finally, the use of a first-stage fluidic amplifier will minimize the probability of actuator hardover, as described in section 3.6, because (1) the torque motor (with its potential failure mode due to electrical short circuits) has been eliminated and (2) the fluidic amplifier will generally fail neutrally.

5. COMPARISON BETWEEN FIRST-STAGE HYDRAULIC AMPLIFIERS AND FLUIDIC AMPLIFIERS

The first-stage hydraulic amplifier and the first-stage fluidic amplifier will be compared in terms of input signal power, output power, frequency response, leakage flow, and temperature sensitivity. A summary statement concerning the usefulness of first-stage fluidic amplifiers is also given.

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[11] *L. J. Banaszak and W. M. Posingies, Hydrofluidc Stability Augmentation System (HYSAS) Operational Suitability Demonstration, United Army Air Mobility Research and Development Laboratory TR-77-31 (October 1977).*

[22] *W. T. Fleming and H. R. Gamble, Reliability Data for Fluidic Systems, AiResearch Manufacturing Co., Phoenix, AZ, Harry Diamond Laboratories HDL-CR-78-092-1 (December 1976).*

5.1 Input Signal Power

Both the first-stage hydraulic amplifier and the fluidic amplifier can operate with input signal levels in the 100-mW range.

5.2 Output Power

The output power available from the flapper-nozzle valve and from the fluidic amplifier are comparable; one can be greater than the other over a portion of the ouput range or over the complete range, depending on the relative geometries and supply pressures. Figure 19 contains output characteristics for nominal geometries and various fluidic amplifier supply pressures.

5.3 Frequency Response

The dynamics of both the hydraulic amplifier and the fluidic amplifier are small compared to the overall dynamics of two-stage servovalves. These first-stage elements determine overall servovalve response primarily by the amount of output power they provide to drive or control the second-stage spool. The output power of both these elements is comparable; however, the hydraulic amplifier has higher pressure recovery or available output pressure. This is an important difference if the second stage is a spring-centered spool valve (and it must be if the first stage is a fluidic amplifier with no moving parts), because this type of second stage requires high spool driving pressures. Since the hydraulic amplifier has high output pressure, it has more output power to drive the second-stage spring-centered spool valve over its maximum range to get rated servovalve output flow. Comparable response can be obtained with a first-stage fluidic amplifier by using additional amplifiers in parallel.

5.4 Leakage Flow

First-stage fluidic amplifiers will have higher leakage flow than first-stage hydraulic amplifiers because (1) the fluidic amplifier has a supply nozzle area that is generally larger than the hydraulic amplifier orifice area and (2) more than one fluidic amplifier stage will generally be required. First-stage fluidic amplifier leakage flow is typically a minimum of 5 to 6 times greater than a comparable first-stage hydraulic amplifier.

5.5 Temperature Sensitivity

First-stage hydraulic amplifiers are less temperature sensitive than first-stage fluidic amplifiers. A fluidic amplifier cascade operated in the laminar regime must be temperature compensated to prevent gain and null changes with fluid temperature variations.

However, recent studies indicate that a first-stage fluidic amplifier cascade of no more than four amplifiers operated in the turbulent regime may have temperature sensitivity comparable to the hydraulic amplifier and not require temperature compensation. Moving-part fluidic amplifiers also have temperature sensitivity comparable to hydraulic amplifiers.

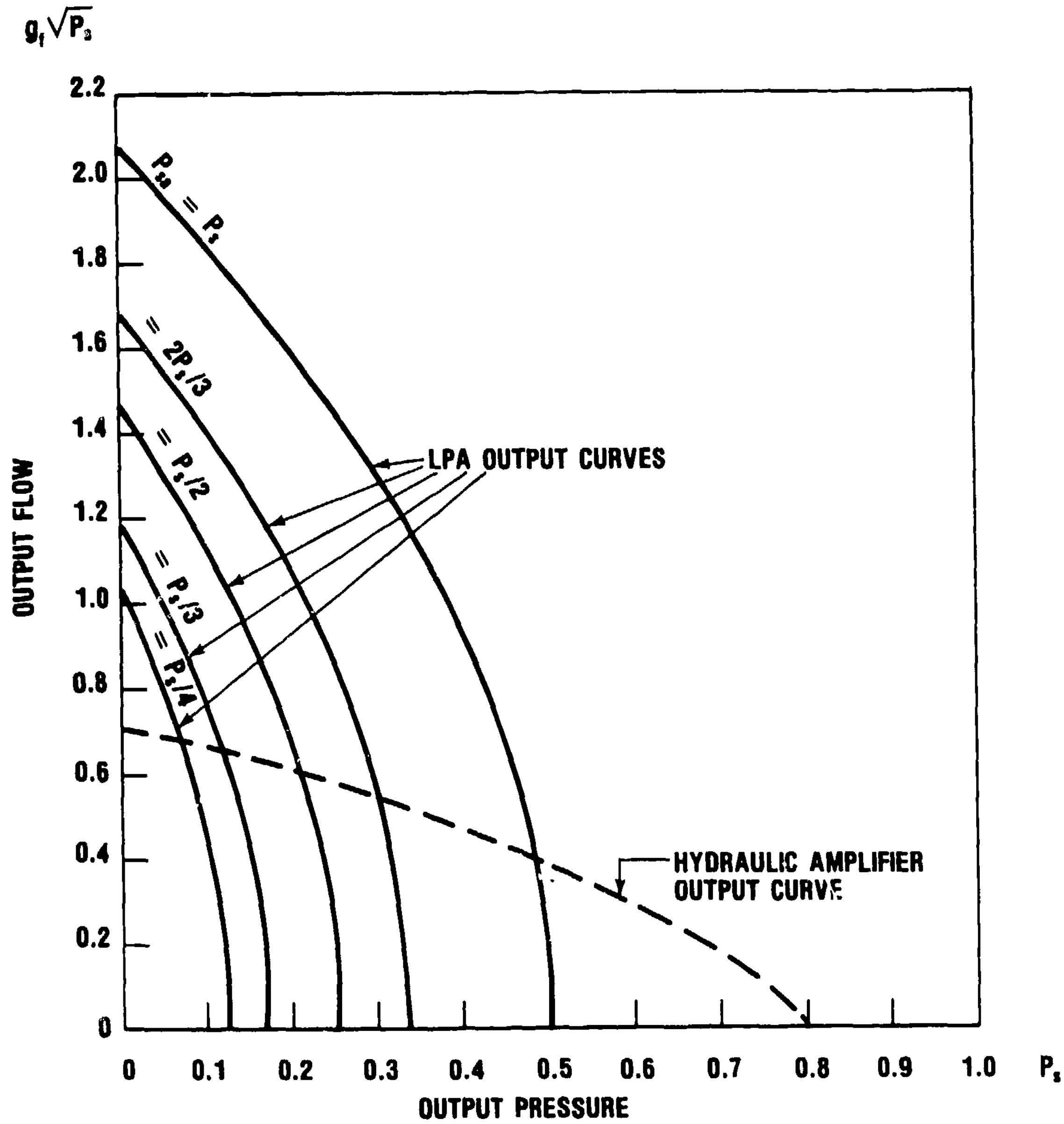

Figure 19. Output characteristics of hydraulic amplifier with supply pressure P_S and laminar proportional amplifiers with various supply pressures P_{Sa}.

5.6 Reliability

First-stage fluidic amplifiers are more reliable than first-stage hydraulic amplifiers in terms of minimizing the probability of actuator hardover failures. Actuator hardover failure is minimized with the first-stage fluidic amplifier because (1) the electrical torque motor (which can develop short circuits and cause actuator hardover) has been eliminated, (2) the minimum clearance areas of the fluidic amplifier are generally larger than those of the hydraulic amplifier, and (3) should a sudden blockage occur in the fluidic amplifier, it will probably fail neutrally.

5.7 Summary

The first-stage fluidic amplifier is comparable to the hydraulic amplifier in terms of input power level, output power, and frequency response. The hydraulic amplifier has less leakage flow and is less temperature sensitive. However, the fluidic amplifier is more reliable in terms of failure modes which can cause actuator hardover and in its ability to operate with higher levels of fluid contamination. In many applications, the increased reliability offered by fluidic amplifiers more than offsets the disadvantage of additional leakage flow. The problem of fluidic amplifier gain and null temperature sensitivity may be overcome by (1) temperature compensation of fluidic amplifiers operated in the laminar regime, (2) operating the fluidic amplifiers in the turbulent regime, and (3) the use of moving-part fluidic amplifiers whose gain and null characteristics are relatively temperature insensitive. The use of moving-part fluidic amplifiers makes it possible to use mechanical feedback from the second-stage spool. This in turn reduces the number of fluidic amplifiers needed in the first-stage cascade and thus reduces first-stage temperature sensitivity and leakage flow.

Two-stage servovalves using first-stage fluidic amplifiers can operate in high shock and vibration environments and with high levels of fluid contamination. These advantages more than justify continued development of first-stage fluidic amplifiers with and without moving parts.

6. TRENDS IN FLUIDIC SERVOVALVE DEVELOPMENT

As stated in the previous sections, development of two-stage fluidic input servovalves has been aimed at improved servovalve response, leakage flow, reliability, and cost. However, the development of microprocessors in recent years has introduced additional considerations to valve development. First, the availability of low-cost, reliable microprocessors means that most control systems will have some type of

electronic controller. Fluidic systems and subsystems will therefore have to interface with electronic controllers or central processing units. The electronic signal could be converted to a hydraulic signal and summed at the summing junction as another input to the fluidic system. However, since microprocessor signal processing is faster and more accurate, the signal should be introduced at the servovalve--possibly with override capability. This means that fluidic systems and subsystems must have dual input (fluidic, electronic) servovalves.[23] The U.S. Army Research and Technology Laboratories (AVRADCOM) has let a contract to develop this type of servovalve.

A second consideration is the use of microprocessors with fluid power systems. The most tempting use of microprocessors in electrohydraulic systems is to replace the analog or digital summing point and to condition the error information for driving the hydraulic controller.[24] Microprocessor-level signals could also be used to drive the low-input-power-level electrohydraulic servovalves with fluidic-type first-stage elements.[19] Moreover, microprocessor-level signals could be converted to hydraulic signals and used to drive fluidic-input, two-stage servovalves.

Finally, fluidic-type amplifiers that have moving parts are successfully being used in commercially available electrohydraulic servovalves as noted in section 4.3. These servovalves have the advantage of high reliability, particularly in high vibration and shock environments, and can operate with low-level input signals.

7. CONCLUSIONS

A review of the developments of two-stage fluidic input servovalves, using a first-stage fluidic amplifier and a second-stage spool valve, indicates that this servovalve configuration is feasible. However, fluidic amplifier gain and null variation with fluid temperature change is a problem.

The first-stage fluidic amplifier is comparable to the hydraulic amplifier in terms of input signal power levels, output power, and frequency response. The hydraulic amplifier has less leakage flow and is less temperature sensitive. However, the fluidic amplifier is more

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[19] *Robert Bosch Corporation, Electro-hydraulic Servovalve Series 0814-SMV2 and 0814-SMV3, 2800 South 25th Ave., Broadview, ILL.*

[23] *L. P. Biafore and B. Holland, Fluidics--Feasibility Study Electro/Hydraulic/Fluidic Direct Drive Servo Valve, Rockwell International Corp., Columbus, OH, NADC TR-78033-60 (March 1979).*

[24] *R. H. Maskrey, Possible Uses of Microprocessors for Fluid Power, Hydraulics and Pneumatics (June 1979).*

reliable in terms of failure modes which cause actuator hardover and its ability to operate with larger sizes of particulate contamination. In many applications the increased reliability offered by fluidic amplifiers more than offsets the additional leakage flow. The problem of gain and null temperature sensitivity may be overcome by (1) temperature compensation of fluidic amplifiers operated in the laminar regime, (2) operating the fluidic amplifiers in the turbulent regime, and (3) the use of moving-part fluidic amplifiers with gain and null characteristics that are relatively temperature insensitive.

A two-stage fluidic input servovalve with a first-stage, mechanically actuated fluidic-type amplifier and a second-stage spool valve with mechanical feedback is the *best* fluidic input servovalve configuration. The use of a mechanically actuated fluidic amplifier makes it possible to use mechanical feedback from the second-stage spool valve. (The servovalve thus becomes a closed loop, spool position, control system.) This reduces the number of fluidic amplifiers needed in the first-stage amplifier cascade and thus reduces the first-stage leakage flow and temperature sensitivity. The development of this servovalve configuration awaits the development of an efficient fluid means of moving the mechanical actuator.

The development of microprocessors in recent years has introduced additional considerations to valve development. First, the availability of low-cost, reliable microprocessors means that most control systems will have some type of electronic controller. Fluidic systems and subsystems will therefore have to interface with electronic controllers or central processing units. This means that fluidic systems and subsystems must have dual input (fluidic and electronic) servovalves. A second consideration is the use of very low signal threshold fluidic input servovalves with microprocessors. Microprocessor-level signals could be used to drive existing electrohydraulic servovalves or could be converted to hydraulic fluidic signals and used to drive fluidic input servovalves.

Finally, fluidic-type amplifiers (that is, amplifiers with moving parts) are successfully being used in commercially available electro-hydraulic servovalves. These servovalves have the advantage of high reliability, particularly in high shock and vibration environments, and can operate with relatively high levels of fluid contamination.

ACKNOWLEDGEMENT

The author acknowledges the assistance of Dr. George Mon and Mr. John Goto in the formulation and solution, respectively, of the hydraulic amplifier load-flow equation. The author also thanks Dr. Tadeusz Drzewiecki for his thorough review and suggestions concerning the material presented in this paper.

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APPENDIX A--TWO-STAGE FLUIDIC SERVOVALVE RESEARCH AND DEVELOPMENT

A literature search was initiated in connection with this study. The search revealed research and development studies of the following two-stage fluidic servovalve configurations.

APPENDIX A.--TWO-STAGE FLUIDIC INPUT SERVOVALVE RESEARCH AND DEVELOPMENT

Reference No.	Servovalve configuration		Sponsor[a]	Date
	First stage	Second stage		
1	Bellows, flapper-nozzle valve	Spool valve	NADC (Honeywell, Moog)	1962
2	Venjet amplifiers, Vortex amplifiers	Vortex amplifiers	NASA (Bendix)	1965
3	Venjet amplifiers, Vortex amplifiers	Vortex amplifiers	NASA (Bendix)	1968
4	Bellows, flapper-nozzle valve	Vortex valves	AMRDL (GE)	1969
5	Bellows, flapper-nozzle valve, amplifier cascade	Spool valve	AFFDL (GE)	1970
6	Bellows, flapper-nozzle valve	Vortex valves	AMRDL (GE)	1970
7	Bellows, flapper-nozzle valve	Spool valve	AMRDL (Honeywell)	1971
	Bellows, flapper-nozzle valve	Vortex valves	AMRDL (Honeywell)	1971
8	Fluidic amplifier cascade	Spool valve	AMRDL (Honeywell)	1973
9	Bellows, flapper-nozzle valve	Spool valve	NADC (Honeywell Moog)	1974
10	Bellows, flapper-nozzle valve	Spool valve	AFFDL (Honeywell)	1975
11	Bellows, flapper-nozzle valve	Poppet valves	AMRDL (Honeywell)	1975
12	Fluidic amplifier cascade	Spool valve	AMRDL (Honeywell)	1976
13	Fluidic amplifier bellows, flapper-nozzle valve	Spool valve	HDL (MIT)	1976

APPENDIX A.--TWO-STAGE FLUIDIC INPUT SERVOVALVE RESEARCH AND DEVELOPMENT (Cont'd)

Reference No.	Servovalve configuration		Sponsor[a]	Date
	First stage	Second stage		
14	Bellows, flapper-nozzle valve	Spool valve	AMRDL (Honeywell)	1977
15	Fluidic amplifier; bellows, flapper-nozzle valve	Spool valve	HDL (MIT)	1977
16	Fluidic amplifier cascade	Spool valve	AMRDL (Honeywell)	1978
17	Fluidic amplifier; bellows, flapper-nozzle valve	Spool valve	HDL	1979
18	Fluidic amplifier cascade	Spool valve	HDL (MIT)	In progress

[a]*AFFDL: United States Air Force Flight Dynamics Laboratory.*
AMRDL: United States Army Air Mobility Research and Development Laboratory.
NADC: United States Naval Air Development Center.
NASA: National Aeronautics and Space Administration.
HDL(MIT): Harry Diamond Laboratories (Massachusetts Institute of Technology).

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NOMENCLATURE

A_i fluidic amplifier supply nozzle area for the ith stage, m^2

A_s spool end area, m^2

A_{sa} fluidic amplifier supply nozzle area ($b_s \cdot h$), m^2

b_i number of parallel sections in the ith stage of a fluidic amplifier cascade

b_s fluidic amplifier supply nozzle width, m

C_D hydraulic amplifier (flapper) fixed orifice discharge coefficient

C_{Da} fluidic amplifier supply nozzle discharge coefficient

C_{Df} hydraulic amplifier, flapper-nozzle curtain area discharge coefficient

C_{Di} fluidic amplifier supply nozzle discharge coefficient for the ith stage

D flapper fixed orifice diameter, m

D_N flapper-nozzle orifice diameter, m

f frequency, 1/s (Hz)

g_a $C_{Da}A_{sa}\sqrt{2/\rho}$, fluidic amplifier supply flow coefficient

G_A maximum fluidic amplifier power gain, watts/watts

g_f $C_D\ (\pi/4)D^2\ \ 2/\rho$, flapper-nozzle orifice flow coefficient, $\sqrt{m^7/kg}$

G_F maximum hydraulic amplifier power gain, watts/watts

g_i $C_{Di}A_i\sqrt{2/\rho}$, fluidic amplifier supply flow coefficient for the ith stage, $\sqrt{m^7/kg}$

h fluidic amplifier supply nozzle height, m

i summation index, number of the individual stage of a fluidic amplifier cascade

j total number of stages of the fluidic amplifier cascade

K_A hydraulic amplifier gain constant, $m^5/N{\cdot}s$

P_m nondimensional hydraulic amplifier output pressure, p_m/P_s

P_n nondimensional fluidic amplifier output pressure, p_n/P_{sa}

P_s hydraulic amplifier (flapper) supply pressure, Pa

P_{sa} fluidic amplifier supply pressure, Pa

p_m actual hydraulic amplifier output pressure, Pa

p_n actual fluidic amplifier output pressure, Pa

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NOMENCLATURE (Cont'd)

Q_e	hydraulic amplifier leakage or tare flow, m^3/s
$Q^1{}_e$	fluidic amplifier leakage or tare flow, m^3/s
Q_F	maximum hydraulic amplifier output flow, m^3/s
Q_i	fluidic amplifier supply flow for the ith stage, m^3/s
Q_m	nondimensional hydraulic amplifier output flow
Q_n	nondimensional fluidic amplifier output flow, q_n/Q_{sa}
Q_p	spool valve drive flow, m^3/s
Q_{sa}	fluidic amplifier supply flow, m^3/s
q_m	actual hydraulic amplifier output flow, m^3/s
q_n	actual fluidic amplifier output flow, m^3/s
s	Laplace Transform variable, 1/s (Hz)
T	hydraulic amplifier transfer function, $m^5/N \cdot s$
W	nondimensional hydraulic amplifier output power, P_mQ_m
W^1	nondimensional fluidic amplifier output power, P_nQ_n
W_O	maximum hydraulic amplifier output power, watts
W_{IO}	maximum fluidic amplifier output power, watts
W_I	input signal power to the hydraulic amplifier, watts
W^{I}_{I}	input signal power to the fluidic amplifier, watts
x_s	maximum spool displacement, m
ξ	flapper-nozzle valve damping ratio
ρ	fluid density, kg/m^3
τ_m	torque motor time constant, s
ψ	H^1_0/H_0
ω_n	$2\pi f$, 1/s (Hz)
ω	flapper-nozzle valve natural frequency

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