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HYDROFLUIDIC SERVOACTUATOR DEVELOPMENT

Honeywell Inc.
Government and Aeronautical Products Division
Minneapolis, Minn. 55413

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Final Report

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Prepared for

EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
Fort Eustis, Va. 23604

EUSTIS DIRECTORATE POSITION STATEMENT

The purpose of this program was to design and develop a second-generation hydrofluidic servovalve and to demonstrate the performance of a servoactuator using the servovalve. The servovalve uses a hydrofluidic amplifier cascade input state that replaces the bellows-flapper-nozzle of conventional servovalves and a fluidic feedback transducer.

The report has been reviewed by the Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory; it is published for the exchange of information and appropriate application.

Mr. George W. Fosdick of the Systems Support Division served as project engineer for this effort.

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This report covers the design and development of a second-generation hydrofluidic servovalve. The objective of the program was to demonstrate the performance of a servoactuator using a hydrofluidic amplifier cascade input stage that replaces the bellows-driven flapper-nozzle valve of a conventional servovalve, a fluidic feedback transducer, and an actuator. The servovalve was designed to interface with a GFE actuator, which was used in the performance of Contract DAAJ02-73-C-0046, and uses U. S. Army aircraft hydraulic fluid meeting specifications of MIL-H-5606.

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PREFACE

This document is the final report on the development program of a hydrofluidic servoactuator authorized by the Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory (USA AMRDL), Fort Eustis, Virginia, under Contract DAAJ02-75-C-0035. The project engineer on this program was Mr. George Fosdick.

The objective of this program was to design and develop a second-generation hydrofluidic servoactuator, with increased reliability and reduced cost, by using hydrofluidic amplifiers and fluidic feedback to replace the conventional servovalve, flapper-nozzle, first-stage and mechanical feedback. The work effort presented was conducted over a period from 6 May 1975 to 27 March 1976.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Concluded)

LIST OF TABLES

SECTION 1
SERVOACTUATOR DESIGN

This program consisted of the design and development of a hydrofluidic servoactuator using a Government-furnished actuator from Contract DAAJ02-73-C-0046 and the servovalve designed during this contract effort. The design of the servovalve included an amplifier cascade, a flapper-nozzle position feedback transducer, and the servovalve body. Performance of the servoactuator was to meet the requirements of a specification also drawn up during this contract, which defines a servoactuator for the typical SAS application.

DESIGN REQUIREMENTS

The objective of the hydrofluidic servoactuator design is to provide a more reliable, lower-cost unit than the conventional bellows, flapper-nozzle, spool valve, and mechanical feedback configuration. The design technique consists of replacing the bellows-driven flapper-nozzle valve with a hydrofluidic amplifier cascade that drives the spool valve directly. Fluidic summing is provided in the feedback loop through a flapper-nozzle device that senses the actuator position and sends this signal to the summing amplifier. The servovalve from the servoactuator of Contract DAAJ02-73-C-0056 was modified and adapted for use with the actuator.

The servovalve was designed to the performance goals listed below:

Operating Pressure	600 ± 50 psig
Gain	0.15 in./psid ± 3%
Stroke	0.6 ± 0.01 in.

Linearity	± 3% of full stroke
Hysteresis	Included in linearity
Threshold	<0.02 psid
Internal leakage	<1.0 cis
Stall load	80 lb
Saturation Velocity	1.0-2.4 in./sec
Null bias	<0.03 in.
Null shift	
Temperature	<0.03 in. (-20 to +275°F)
Supply pressure	<0.006 in. (± 120 psi)
Return pressure	<0.006 in. (0 to 100 psi)
Quiescent signal pressure	<0.006 in. (± 1 psi)
Frequency Response	15 Hz (amplitude response shall be between +2dB and -3dB at this frequency. Phase lag shall be less than 100 degrees. (Test shall be conducted with fluidic amplifier as a driving source.)

RESPONSE ANALYSIS

The response analysis of this mechanization is based on design parameters for the servoactuator. These parameters are:

Second-stage spool spring rate (K_S) = 660 lb/in.

Second-stage spool diameter = 0.1875 in., A_S - 0.0276 in.^{2}

Second-stage spool gain (G_V) = 107 in.^{3}/sec/in.

Feedback ball stroke at actuator stroke of ±0.3 in. = ±0.0187 in.

Actuator area = 0.15 in.^{2}

Actuator gain (K_A) = $\frac{1}{0.15}$ = 6.67 in./in.^{3}

Nozzle Diameter = 0.0185 in.

Orifice Diameter = 0.0061 in.

Servoactuator design parameters were determined from the given data and the block diagram of Figure 1.

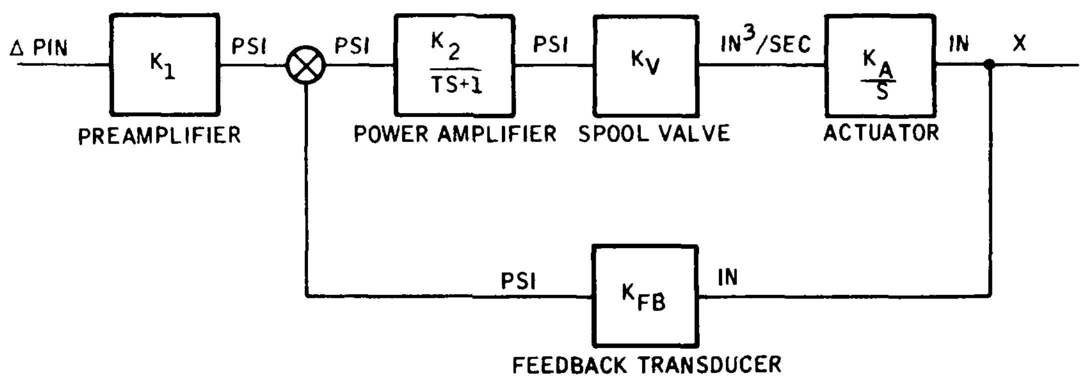

Figure 1. Hydrofluidic Servoactuator Block Diagram.

The constants of the block diagram in Figure 1 are as follows.

K_1, K_2 - Amplifier cascade gain psi/psi

K_V - Spool valve flow-pressure gain in.^{3}/sec/psi

K_A - Actuator gain in./in.^{3}

K_{FB} - Feedback transducer gain psi/in.

The input-output transfer function for this device is

$$\frac{X}{\Delta Pin} = \frac{\frac{K_2K_VK_A}{(TS+1)S}}{1 + \frac{K_2K_VK_AK_{FB}}{(TS+1)S}} = \frac{K_2K_VK_A}{(TS+1)S + K_2K_VK_AK_{FB}} =$$

$$\frac{\frac{K_2K_VK_A}{T}}{S^2 + \frac{S}{T} + \frac{K_2K_vK_AK_{FB}}{T}}$$

The natural frequency squared becomes

$$\omega^2 = \frac{K_2K_VK_AK_{FB}}{T}$$

and the feedback gain is

$$K_{FB} = \frac{\omega^2 T}{K_2K_VK_A} = \frac{(94)^2\ (0.010)}{20\ (0.0045)\ (6.67)} = 147.2 \text{ psi/in.}$$

where

ω = 94 rad/sec (15 Hz)

K_2 = 20

$K_V = \frac{G_VA_S}{K_S} = \frac{107\ (0.0276)}{660} = 0.0045 \text{ in.}^3\text{/sec/psi}$

Position gain (K_{POS}) is

$$K_{POS} = \frac{1}{K_{FB}} = \frac{1}{147.2} = 0.0068 \text{ in./psi}$$

which requires a preamplifier gain (K_1)

$$K_1 = \frac{G_S}{K_{POS}} = \frac{0.15}{0.0068} = 22.06 \text{ psi/psi}$$

where

G_S = 0.15 in./psi

The saturation range requirements of the preamplifier and power amplifier cascades are determined from the maximum actuator velocity.

The maximum flow into the spool valve (Q_{SM}) can be found by

$$Q_{SM} = \frac{\text{Maximum Actuator Velocity}}{\text{Actuator Gain } (K_A)} = \frac{2 \text{ in./sec}}{6.67 \text{ in./in.}^3} = 0.30 \text{ in.}^3\text{/sec}$$

The power amplifier range (ΔPpo) can be found by

$$\Delta P_{PO} = \frac{Q_{SM}}{K_V} = \frac{0.30}{0.0045} = 66.6 \text{ psid}$$

The preamplifier range for full stroke (ΔP_{PR}) can be found by

$$\Delta P_{PR} = \Delta P_1 K_1 = 2\,(22.1) = 44.2 \text{ psid}$$

The above values for K_1, ΔP_{PO} and ΔP_{PR} were high for amplifier mechanization; therefore, the value of the spool spring rate (K_S) was varied to permit lower values as shown in Table 1. The value of K_S equal to 300 lb/in. was chosen as the system design parameter.

TABLE 1. VALVE DESIGN PARAMETERS

K_S lb/in.	K_V in.^{3}/sec/psi	K_{FB} psi/in.	K_2 psi/psi	K_1 psi/psi	P_{PO} psid	P_{PR} psid
660	.0045	147.2	20	22.1	66.6	44.2
400	.0074	89.5	20	13.4	40.5	26.8
300	.0098	67.6	20	10.1	30.6	20.2
200	.0148	44.7	20	6.7	20.3	13.4

The determination of the damping ratio gives

$$2\delta\omega = \frac{1}{T}$$

$$\delta = \frac{1}{2\omega T} = \frac{1}{2(94)\ (0.010)} = 0.53$$

where ω = 94 rad/sec

T = 0.010 sec (approximate time lag between amplifier and spool valve)

A comparison of the design response goal and the response calculated from the design parameters is shown in the Bode plot of Figure 2.

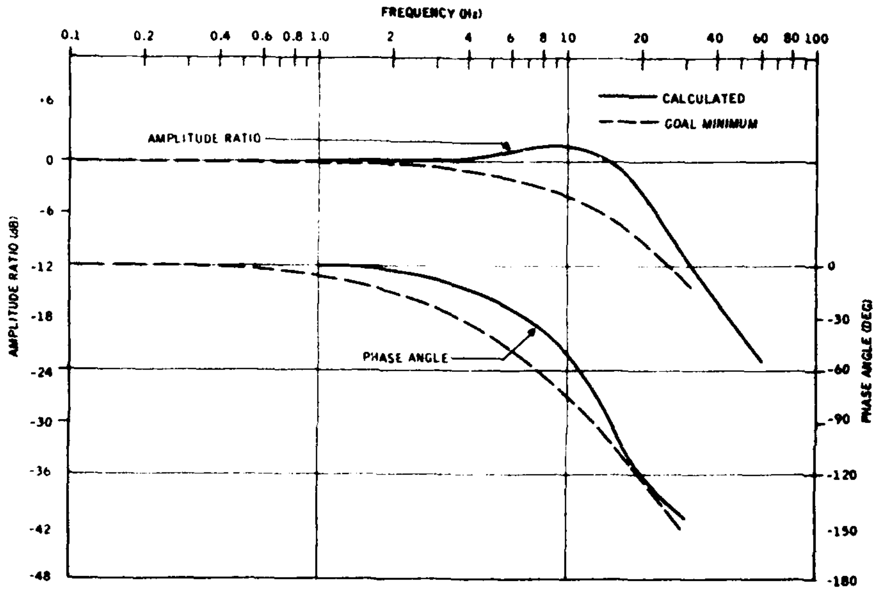

Figure 2. Comparison of the Design Response Goal to the Calculated Goal.

The initial servovalve and actuator system mechanization was determined and is shown in Figure 3. This configuration includes the standard flapper-nozzle concept for a feedback transducer. This mechanization was later changed to include a reverse-flow flapper-nozzle configuration as shown in Figure 4. The flapper-nozzle configuration does not directly affect the servoactuator response. These indirect effects will be discussed in Section 3.

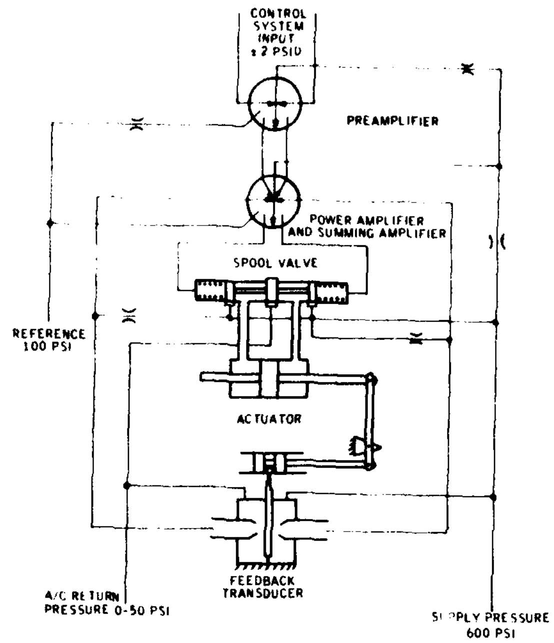

Figure 3. Initial Servoactuator Schematic.

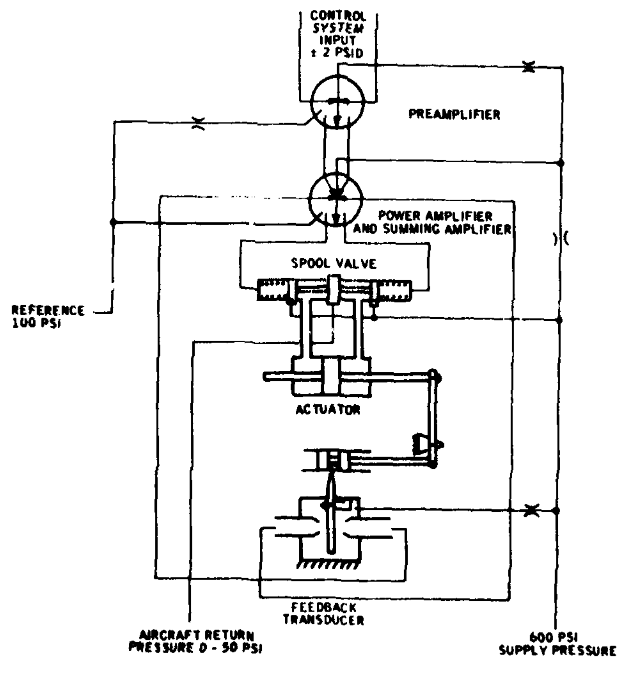

Figure 4. Final Servoactuator Schematic.

SECTION 2
HARDWARE DESIGN AND FABRICATION

GENERAL

Hardware design was directed toward the use of the servoactuator from Contract DAAJ02-73-C-0046 and some components of the servovalve used on Contract DAAJ02-73-C-0056 to eliminate contract expenditures for fabricating standard hardware.

The servovalve noted above was disassembled, and the main body and some miscellaneous parts were used. It was then necessary to build a new flapper and spring and to modify the nozzles and some of the porting. In addition, a fluidic amplifier manifold was built that would directly attach to the valve body and provide for mounting of the preamplifier, the summing amplifier, and the power amplifier.

COMPONENT DESIGN

Amplifiers

The amplifiers used on the servovalve were high-pressure amplifiers developed under a Honeywell internal development program. The preamplifier and power amplifier were used unchanged, and the summing amplifier was modified slightly for a more suitable performance in this application. These amplifiers as mounted on the servovalve manifold are shown in Figure 5.

Fluidic Circuit Manifold

The fluidic circuit manifold is also shown in Figure 5. This manifold was designed to mount on the spool valve body, and porting was designed to receive 600 psi operating pressure and the flapper-nozzle signal from the valve body. In addition, porting was provided to interconnect the amplifiers and to return the signal and amplifier dump flow to the valve body. Mounting provisions for three high-pressure amplifiers – a preamplifier, a summing amplifier and a power amplifier – were included.

Spool Valve Body Modification

The spool valve and body used was built by Hydraulic Research and Manufacturing. The spool valve was not changed; however, the body was modified to accommodate porting of the power amplifier output to the ends of the spool valve and to permit porting of the flapper-nozzle signal to the summing amplifier. In addition, the spool valve springs were replaced with springs of a lower rate, and the size of the flapper-nozzle bridge orifices was changed. Figure 6 shows the valve body connected to the fluidic manifold. The final flapper-nozzle spring can be seen in the center of this picture.

Feedback Transducer Design

The flapper-nozzle feedback transducer was designed to provide the desired feedback gain. It was to have a pressure output range (at the summing amplifier input) of ±46 psid for an input of ±0.0187 inch displacement. The most desirable concept was the reverse-flow flapper-nozzle; however, it was first thought to be impossible to design this concept into the existing hardware restraints. As a result, the first flapper-nozzle design included the standard resistance bridge concept.

Figure 5. Servovalve Manifold and Amplifier.

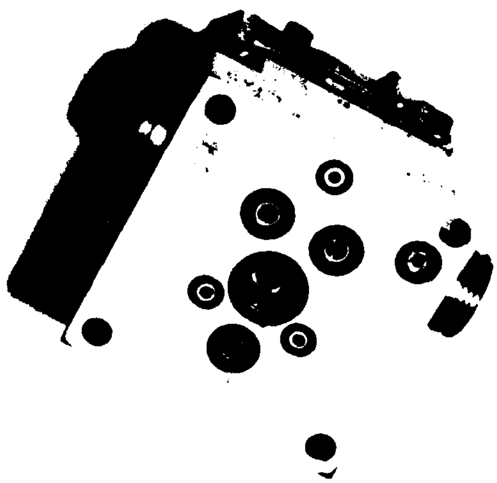

Figure 6. Valve Body and Flex Pivot.

A schematic of this concept is shown in Figure 7. Mechanization of this concept was convenient, as the valve body provided the resistance bridge indicated by the orifices D_O on the diagram. A sketch of the mechanization of this concept is shown in Figure 8.

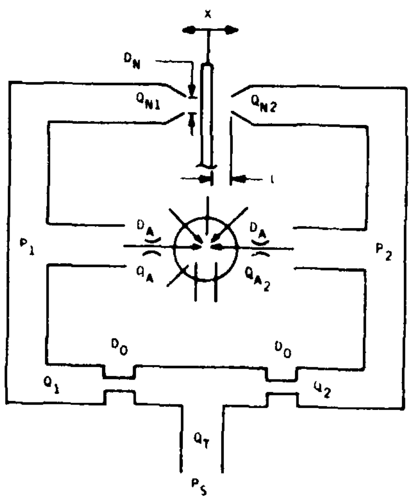

Figure 7. Standard Flapper-Nozzle Schematic.

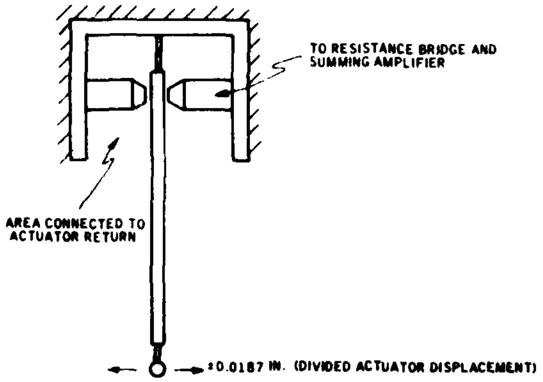

Figure 8. Standard Flapper-Nozzle Mechanization.

In designing the standard flow flapper-nozzle configuration, the following relationships were derived (refer to the schematic, Figure 7).

$$Q_1 = Q_A + Q_N \tag{1}$$

$$Q_N = \frac{\pi D_O^{\ 2}}{4} \quad C_D \sqrt{P_S - P_1} \tag{2}$$

$$Q_A = \frac{\pi D_A^{\ 2}}{4} \quad C_D \sqrt{P_1} \tag{3}$$

$$Q_N = \pi D_N (L + X) C_D \sqrt{P_1} \tag{4}$$

Substituting into equation 1 gives

$$\frac{\pi D_O^{\ 2}}{4} \quad C_D \sqrt{P_S - P_1} = \frac{\pi D_A^{\ 2}}{4} \quad C_D \sqrt{P_1} + \pi D_N (L + X) \ C_D \sqrt{P_1}$$

assuming the flow coefficient (C_D) is the same for all restrictions gives

$$\frac{\pi D_O^{\ 2}}{4} \sqrt{P_S - P_1} = \sqrt{P_1} \left[\frac{D_A^{\ 2}}{4} + D_N (L + X)\right]$$

$$\frac{P_S - P_1}{P_1} = \left[\frac{D_A^{\ 2} + 4 D_N (L + X)}{D_O^{\ 2}}\right]^2$$

$$\frac{P_S}{P_1} = \left[\frac{D_A^{\ 2} + 4D_N (L + X)}{D_O^{\ 2}}\right]^2 + 1$$

$$P_1 = \frac{P_S}{\left[\frac{D_A^{\ 2} + 4D_N (L + X)}{D_O^{\ 2}}\right]^2 + 1} \tag{5}$$

$$P_2 = \frac{P_S}{\left[\underline{D_A}^{2} \frac{+ 4D_N (L - X)}{D_O^{\ 2}}\right]^2 + 1} \tag{6}$$

To provide flow control and relatively good linearity, the value of L is chosen at $\frac{D_N}{8}$, which results in

$$P_1 = \frac{P_S}{\left[\frac{2D_A^2 + D_N^2 + 8D_N X}{2D_O^2}\right]^2 + 1} \tag{7}$$

$$P_2 = \frac{P_S}{\left[\frac{2D_A^2 + D_N^2 - 8\, D_N X}{2D_O^2}\right]^2 + 1} \tag{8}$$

Using Equations 7 and 8, neutral pressure (P_N) for X = O and maximum pressure differential (ΔP_M) were calculated for various values of D_N and D_O. Figures 9 and 10 are plots of these calculations from which values of D_N and D_O can be chosen. Figure 11 is a plot showing values of P_1 and P_2 and the resulting ΔP for a chosen set of values. Using Equations 4 and 5, the total neutral leakage (Q_T) is 0. 75 in.^{3}/sec, where $Q_T = 2Q_N + 2Q_A$.

Later, because of problems such as nonlinearity of its signal and stability problems resulting from lack of isolation from the actuator return, this design was abandoned.

The reverse-flow flapper-nozzle concept was then mechanized for better linearity and complete isolation from the actuator return. A schematic of this concept is given in Figure 12, and a sketch of this mechanization is shown in Figure 13.

The pressure versus flapper displacement for the reverse-flow flapper-nozzle mechanization of Figure 12 was determined in a manner

$$P_N = \frac{P_S}{\left[\dfrac{2D_A^2 + D_N^4 - 8D_N x}{2D_O^2}\right]^2 + 1}$$

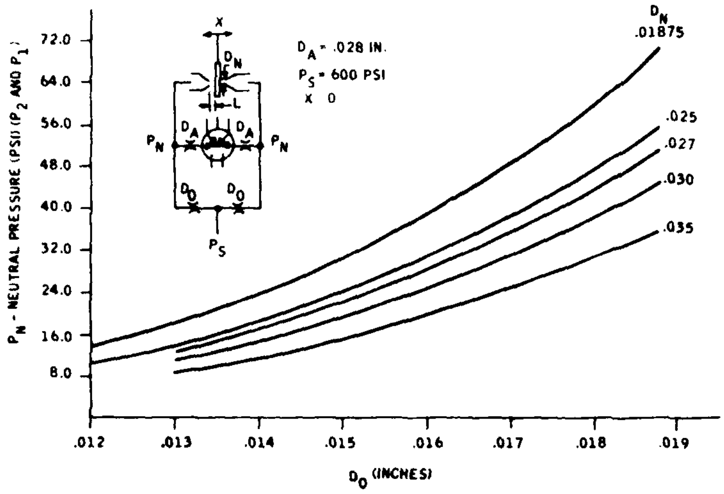

Figure 9. Neutral Pressure Versus D_O Normal Flapper Configuration.

$$\Delta P_M = \sqrt{(P_1 - P_N)^2}$$

$$P_1 = \frac{P_S}{\left[\dfrac{2D_A^2 + D_N^2 - 8D_N x}{2D_O^2}\right]^2 + 1}$$

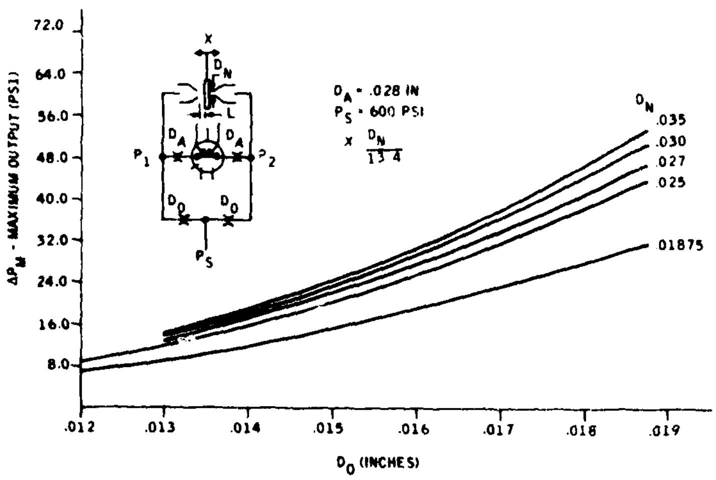

Figure 10. ΔP_M Versus D_O Normal Flapper Configuration.

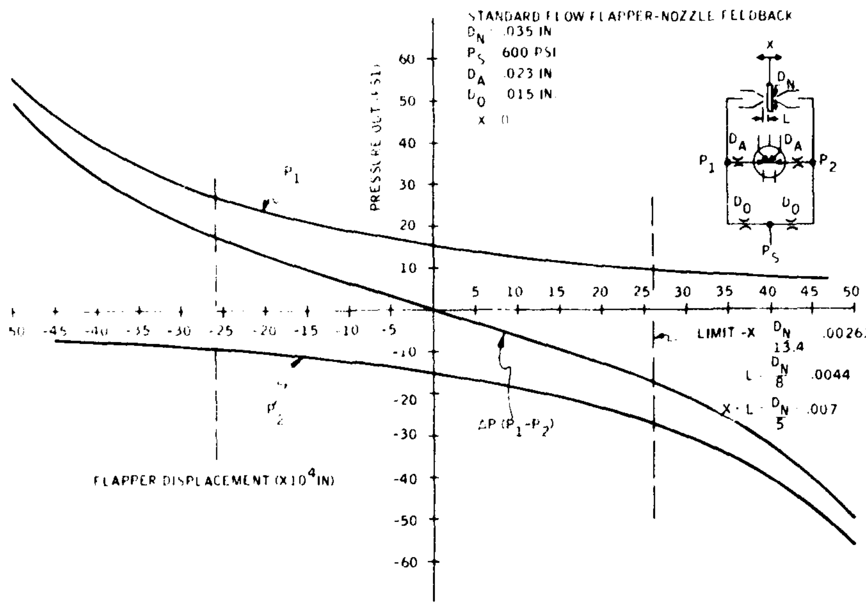

Figure 11. Standard Flow Flapper-Nozzle Feedback.

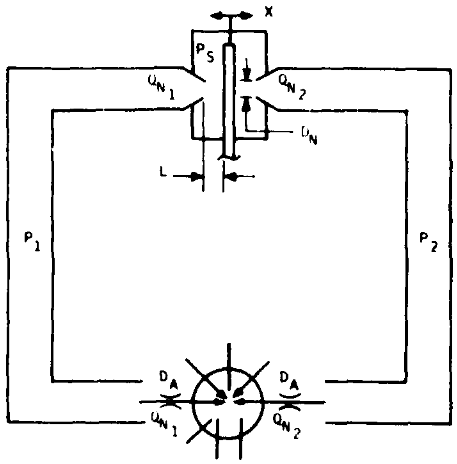

Figure 12. Reverse-Flow Flapper-Nozzle Schematic.

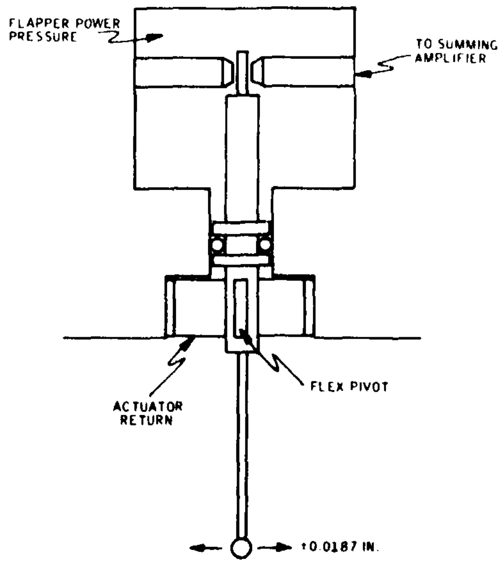

Figure 13. Reverse-Flow Flapper-Nozzle Mechanization.

similar to that used with the standard flapper, except in this case

$Q_N = Q_A$

and the resulting equations become

$$P_1 = \frac{P_S}{\left[\frac{2D_A^2}{D_N^2 + 8D_N X}\right]^2 + 1} \tag{9}$$

$$P_2 = \frac{P_S}{\left[\frac{2D_A^2}{D_N^2 - 8D_N X}\right]^2 + 1} \tag{10}$$

Figure 14 is a plot of Equations 9 and 10 showing P_N and ΔP_M for various values of D_N from which design values were chosen. A plot of P_1 and P_2 and ΔP versus flapper displacement is shown in Figure 15. The total neutral flow is 0.43 in.^{3}/sec using Equation 3 and the relationship $Q_T = 2Q_A$.

The reverse-flow flapper-nozzle design was the final configuration used, and its mechanical design for deflection and stresses was derived. Figure 16 depicts the final design.

Using standard formulas and estimating the flex pivot from previous designs and using Berylium Copper with maximum shear = 90,000 psi, the safe shear is

$$P_R = \frac{2b^2 h\ S_V}{9} = \frac{2(.035)^2(.12)\ 45,000}{9} = 1.47 \text{ in. lb}$$

and the safe torque is

1.47 in. lb X 2 = 2.94 in. lb

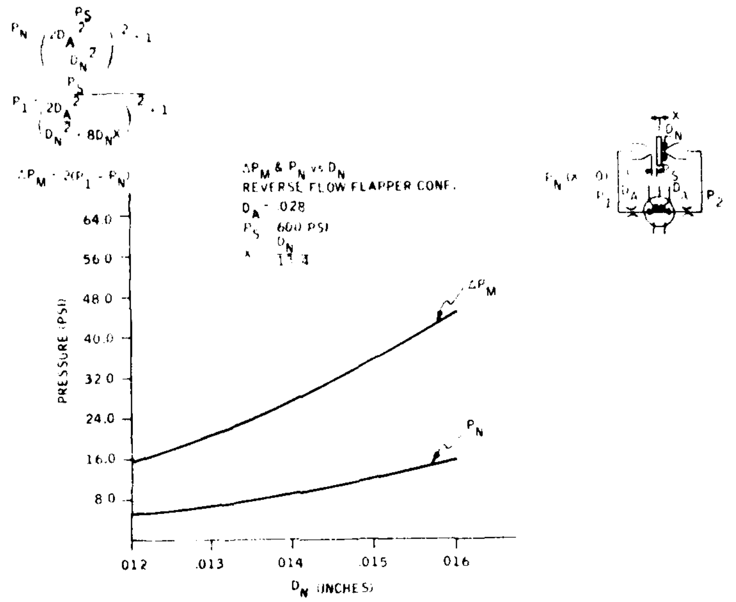

Figure 14. ΔP_M and P_N Versus D_N Reverse-Flow Flapper Configuration.

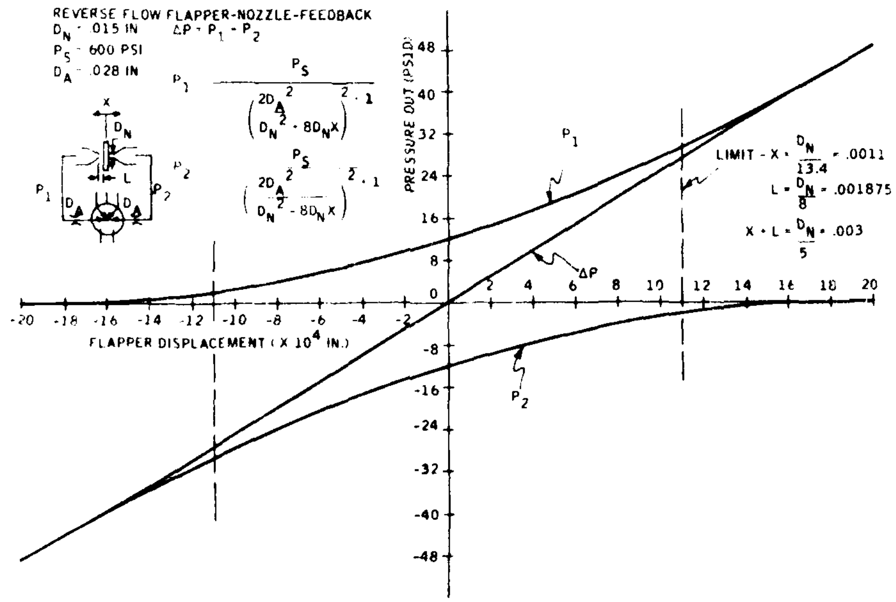

Figure 15. Reverse-Flow Flapper-Nozzle Feedback.

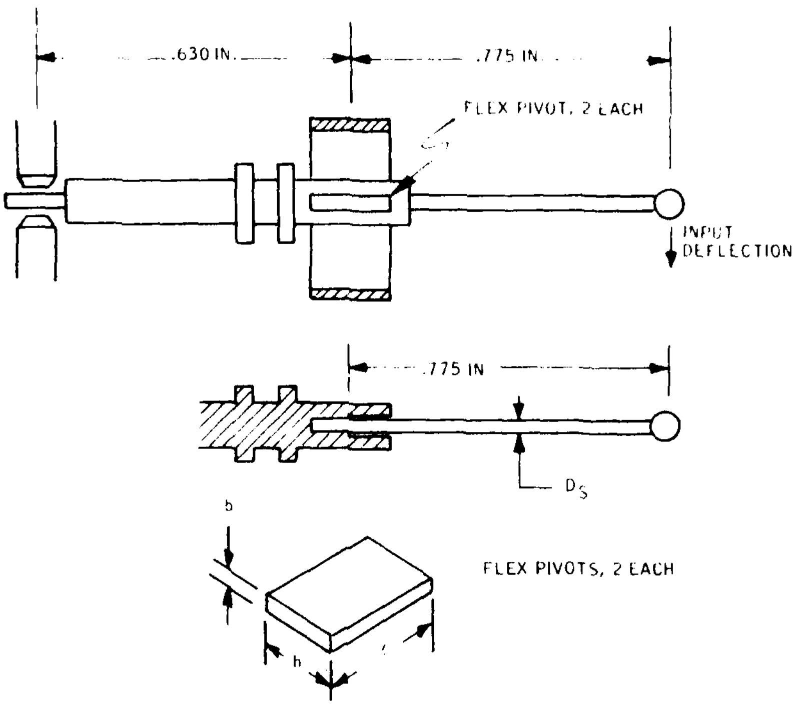

Figure 16. Final Flex Pivot Design.

The deflection at the nozzles was chosen to be 0.002 in., and thus the rotation of the flex pivot must be

$$\text{Arc tan } \frac{.002}{.630} \text{ or } .1833 \text{ degree}$$

Then using another standard formula

$$M = \frac{\phi\,(.28)\,b^3\,h\,G}{\ell\,57.3} = \frac{(.1833)\,(.28)\,(.035)^3\,(.12)\,(7.8 \times 10^6)}{(.115)\,(57.3)}$$

$$M = 0.31257 \text{ in.-lb at maximum deflection}$$

Thus the shear stress in torsion is safe by a factor of 9+.

Finally, the feedback spring wire was designed using standard formulas

$$F = \frac{MY^2}{3EI} = 0.01875 = \frac{.31257\,(.775)^2}{3(30 \times 10^6)\,(.05\, D_S^{\,4})}$$

Therefore

$$D_S^{\,4} = \frac{.31257(.775)^2}{3(30 \times 10^6)(.01875)(.05)} = 2.225 \times 10^{-6}$$

$$D_S = 0.03862 \text{ in. diameter}$$

The stress in the spring wire can then [illegible] calculated through

$$S_S = \frac{MC}{I} = (.31257)\,\frac{1}{.1D_S^{\,3}} = 54264 \text{ psi}$$

With these stresses, a hardened 302 stainless steel was used, and the 1/16-inch-diameter feedback ball was silver soldered to the wire.

SECTION 3
DEVELOPMENT TESTING

COMPONENT DEVELOPMENT

Amplifier Circuit

The amplifiers used on this contract were developed on a Honeywell internal development program. The circuit consists of three amplifiers: a preamplifier, a summing amplifier, and a power amplifier. The preamplifier and power amplifier are the same design, but operate at different pressure levels. A gain curve of each amplifier running under typical operating conditions is plotted in Figure 17. The preamplifier has orifices in series with the control ports to establish the proper gain and null of the servoactuator. The summing amplifier gain curves below show the output versus the input from the preamplifier and the output versus the position of the actuator ram. These amplifiers performed well, having much lower noise than any previous amplifiers used at such high Reynold numbers. However, amplifier noise improvement is desired.

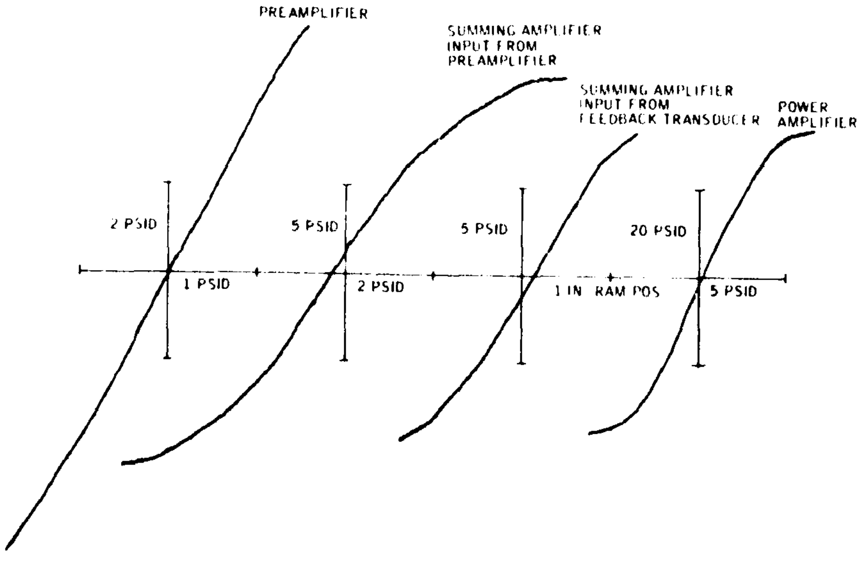

Figure 17. Amplifier Gain Curves.

The performance of these amplifiers over a wide range of temperatures was very good. The gain curves for various temperatures for the preamplifier is shown in Figure 18. The lack of linearity was caused by a variable load resulting from the servovalve feedback. Figure 19 illustrates the same preamplifier running with the second servovalve feedback configuration. The amplifier's performance in this case is much better.

Feedback Transducer

The two feedback transducers that were tested varied greatly in performance, particularly from temperature effects.

The first configuration used was the standard-flow flapper-nozzle as illustrated in Figures 7 and 8. This configuration demonstrated a great sensitivity to null shift and linearity problems, and in this particular application it was not possible to isolate this configuration from the actuator. The curves of Figure 20 illustrate the type of signal that is achieved and the problems mentioned. The isolation problem could be avoided on a bench test setup by running only the series actuator and not the boost actuator. With the boost actuator turned on, the actuator return pressure pulses caused the feedback transducer output to change and set up a condition of total instability.

The reverse-flow flapper-nozzle concept was later mechanized, and much better performance was achieved. Null stability and actuator interaction were no longer a problem, and greatly improved linearity was observed. The curves of Figure 21 illustrate the performance of this device.

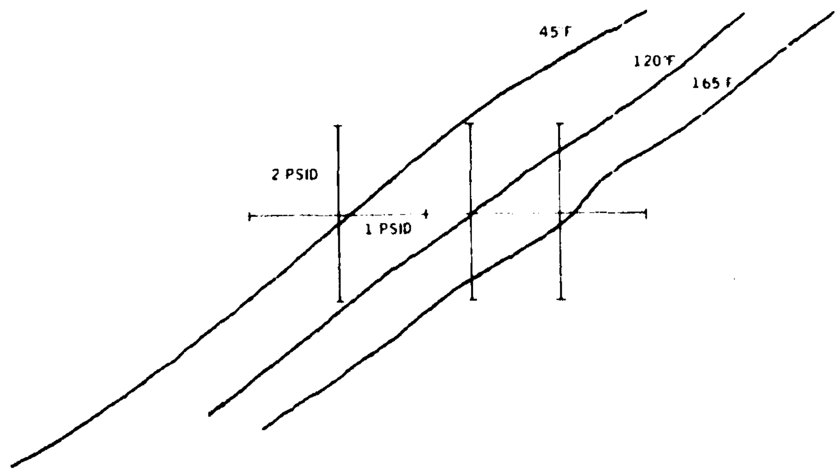

Figure 18. Preamplifier Gain Curves at Various Temperatures on the Initial Servovalve Configuration.

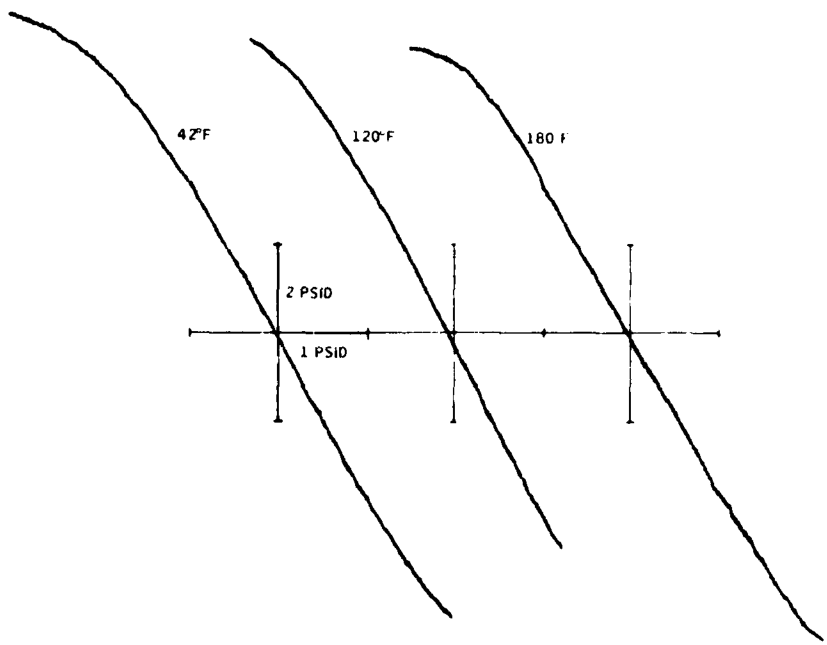

Figure 19. Preamplifier Gain Curves at Various Temperatures on the Final Servovalve Configuration.

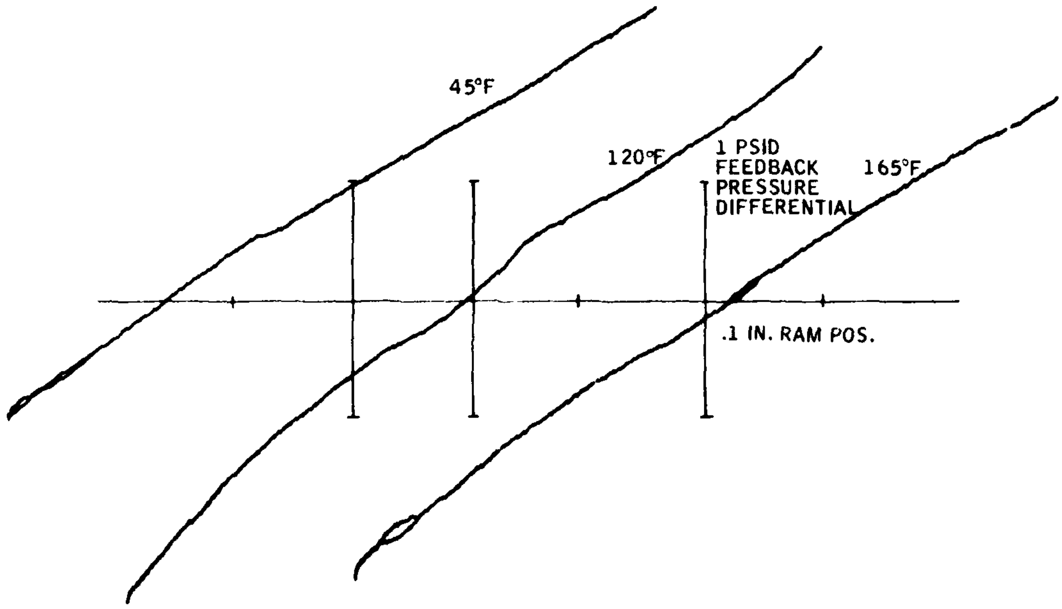

Figure 20. Feedback Output Versus RAM Position (Standard-Flow Flapper-Nozzle).

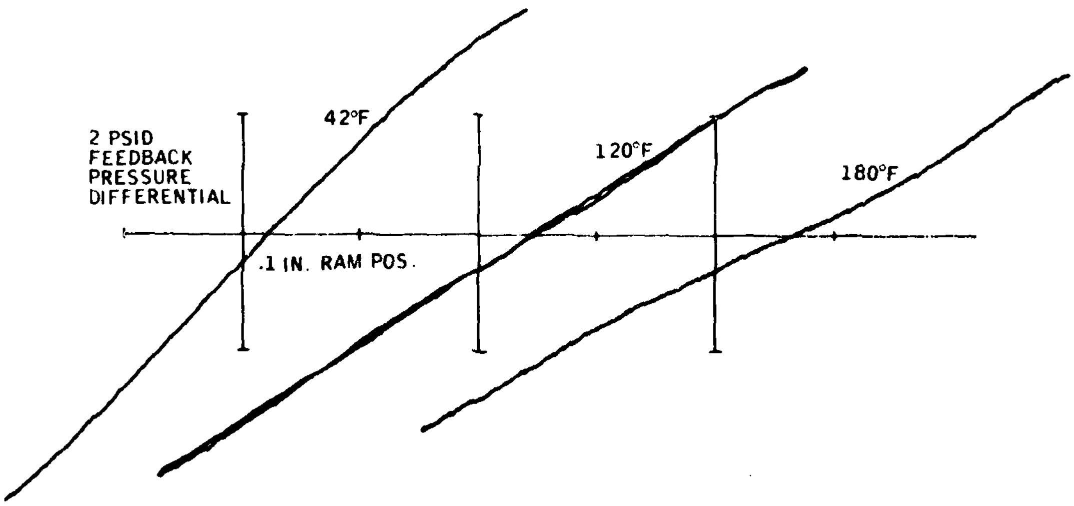

Figure 21. Feedback Output Versus RAM Position (Reverse-Flow Flapper-Nozzle).

Temperature effects on gain is a problem that still has not been solved for this device. The gain change of the two concepts as a function of temperature is reversed, but both have too much change. This problem can be solved by either design modification or temperature compensation; however, contract time and funds did not permit further study of this problem.

SERVOACTUATOR DEVELOPMENT

Initial Configuration Tests

The performance of the initial configuration is illustrated by the gain curves of Figure 22 and by the Bode plots of Figure 23. It can be observed that the gain and the response both vary as a function of temperature. The major cause of this variation is the change in gain of the feedback transducer, which was discussed earlier.

An additional problem with this configuration was the great difficulty in calibration. The servovalve was not readily adjustable, but even more important was the requirement of numerous orifices. All of these orifices not only establish the gain and null of the servovalve, but also affect the fluid pressure levels. Fluid pressure levels must be maintained within certain specific limits in fluidic circuits, and the complex resistance network in this configuration made this difficult.

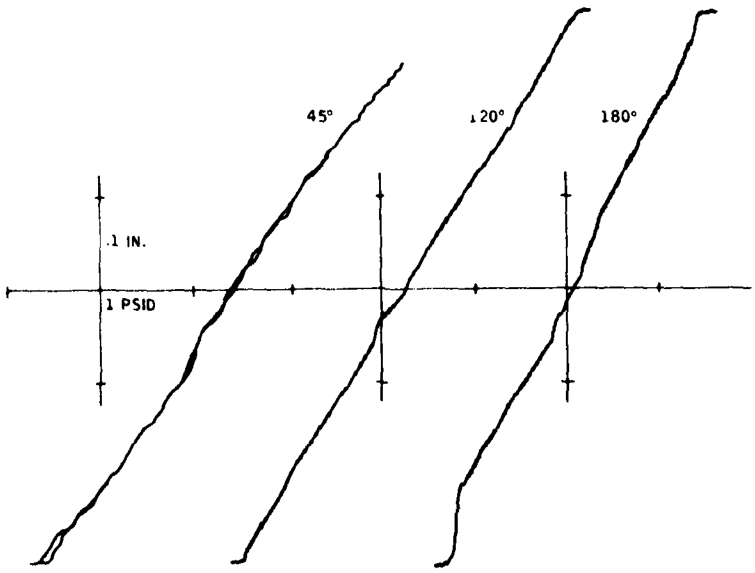

Figure 22. Servoactuator Gain Curves (Initial Configuration).

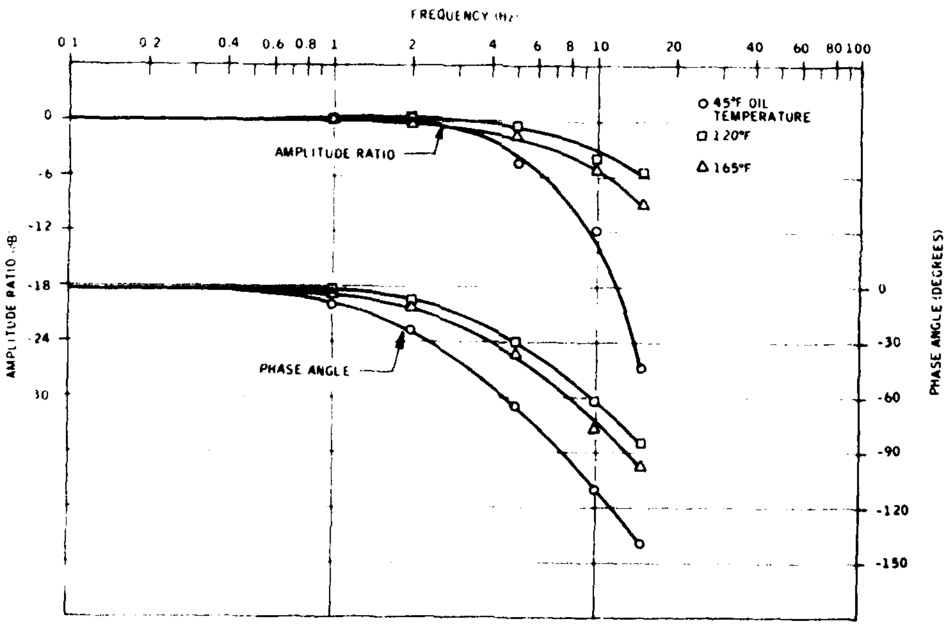

Figure 23. Servoactuator Frequency Response (Initial Configuration).

Final Configuration Tests

The final configuration of the servoactuator used the reverse-flow flapper-nozzle for a feedback transducer. In addition, the servovalve as modified had adjustable gap nozzles and a supply orifice, both of which were easily removable. For these reasons, calibration of the final configuration servovalve was much more readily accomplished. It was also found that desirable levels and gains in the amplifier circuit were more readily established with this simple feedback system.

This configuration still has a temperature problem in that the servoactuator gain increases with temperature, and the response decreases. Again, this is due primarily to changes in gain in the feedback transducer, as mentioned in the component section of the report.

Figure 24 illustrates the gain of the servoactuator as well as null stability and noise as a function of temperature, and Figure 25 illustrates the amplitude and phase relationships of the servoactuator at different temperatures. It can be observed that the response of the servoactuator is much faster at the lower temperatures. Again, this is due to increased feedback gains at the low temperatures.

The response of both configurations across the temperature range is less than the established goals; however, to date, it is definitely faster than the purchased fluidic SAS servoactuators. Better servoactuator response could have been obtained if space had been provided earlier for a second power amplifier. In a production version, the second power amplifier could be designed into the servovalve and the response goals could be met.

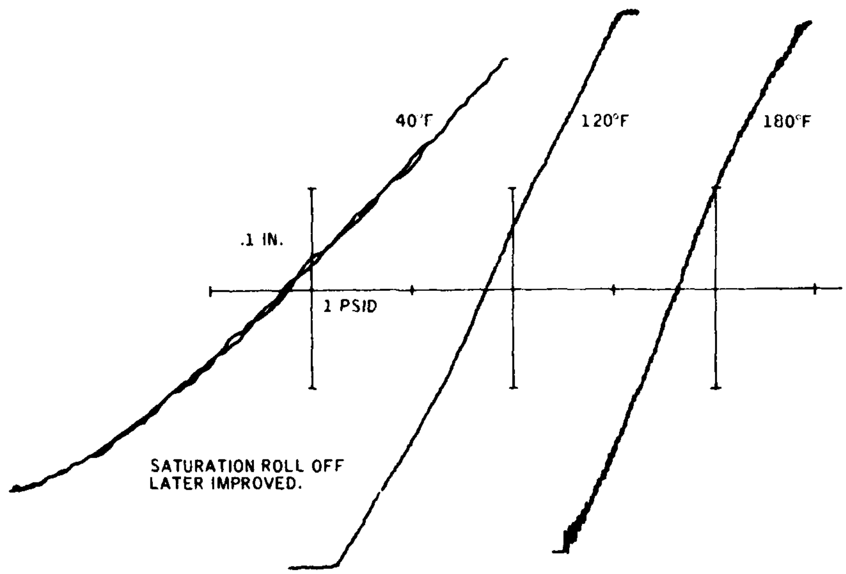

Figure 24. Servoactuator Gain Curves (Final Configuration).

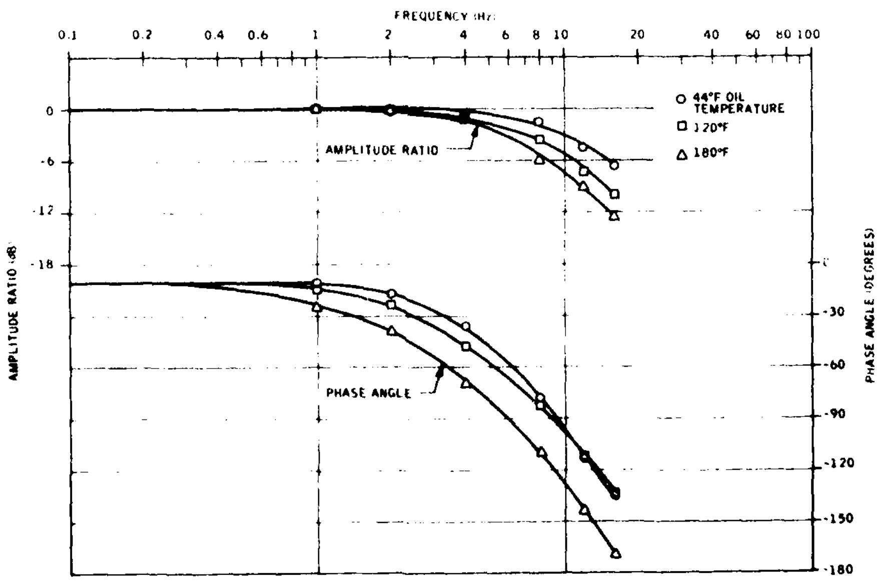

Figure 25. Servoactuator Frequency Response (Final Configuration).

SECTION 4

PERFORMANCE AND ENVIRONMENTAL TESTS PER TEST PLAN

PERFORMANCE TESTS

Servovalve

The test plan called for numerous tests on the servovalve. Some were run on the servovalve alone, some on the servovalve and actuator. This test plan is presented in Appendix A.

Table 2 gives performance test data such as null shift, threshold, and gain, and compares performance against design goals. Linearity and response tests were not run on the servovalve separately. In both cases instrumentation for accurate determination of performance was not available; therefore, performance on linearity and response was determined on the servoactuator. The basic performance test was run at ambient room temperature with 100°F oil temperature.

Servoactuator

Table 2 also includes performance data on the servoactuator and compares it to the goals. Again, the basic performance test was run at ambient room temperature and 100°F oil temperature. Curves of the input-output gain and of the dynamic performance are illustrated in Figures 26 and 27, respectively.

TABLE 2. RESULTS OF SERVOVALVE PERFORMANCE TESTS

Spec. No.	Test	Goal	Actual
3.2.1.2	Leakage	None (static) 1.0 cis (neutral)	None (static) 1.3 cis (neutral)
3.2.1.3.1	Flow gain	6.0 cis/psid applied to the preamplifier input	5.9 cis/psid
3.2.1.3.2	Pressure gain	To be determined	106,000 psi/psi
3.2.1.3.3	Null	< ±0.2	0.04 psi
3.2.1.3.4	Linearity	±0.12 psid input	See actuator linearity
3.2.1.3.5	Threshold	<0.02 psid	<0.02 psid
3.2.1.3.6	Neutral cylinder port pressure	50 ± 15%	48.3%
3.2.1.4	Dynamic response	See servoactuator response	

Servoactuator Performance Spec. No.	Test	Goal	Temp, Air/Oil Ambient/100°F	0/100°F	160/160°F	After Vibration
3.2.1.5.1	Gain	$\frac{\pm 0.3 \pm .015 \text{ in}}{\pm 2 \text{ psid}}$	0.165	0.140	0.265	0.165
3.2.1.5.2	Linearity and hysteresis	±0.015 band	±0.013	±0.010	±0.023	±0.013
3.2.1.5.3	Threshold	<0.02 psid input	<0.02	<0.02	<0.02	<0.02
3.2.1.5.4	Stall load	> 80 lb	> 95 lb	-	-	-
3.2.1.5.5	Saturation velocity	>1.0 - 2.4 in./sec	2.99	2.35	2.1	2.6
3.2.1.5.6	Null bias	±0.03 in.	0	0	0	0
3.2.1.5.7	Null shift	T < 0.03	0	0.005	0.05	0
		P_S < 0.006	0	-	-	±0.002
		P_R < 0.006	0	-	-	±0.002
		P_1 < 0.006	±0.004	-	-	±0.001
3.2.1.5.8	Dynamic performance	See Figure 27				
3.2.1.5.8.1	Phase/amplitude					
3.2.1.5.8.2	Null hunting	<0.018 in.	< ±0.005 ±0.009*	< ±0.005	±0.007	< ±0.005
3.2.2.2	Proof	No leaks		-	-	-
3.2.1.6.1	Temperature	See table above				
3.2.1.6.2 (3.2.1.5.6)	Vibration	Null check	0	-	-	No shift

*One extra test run at -20°F.

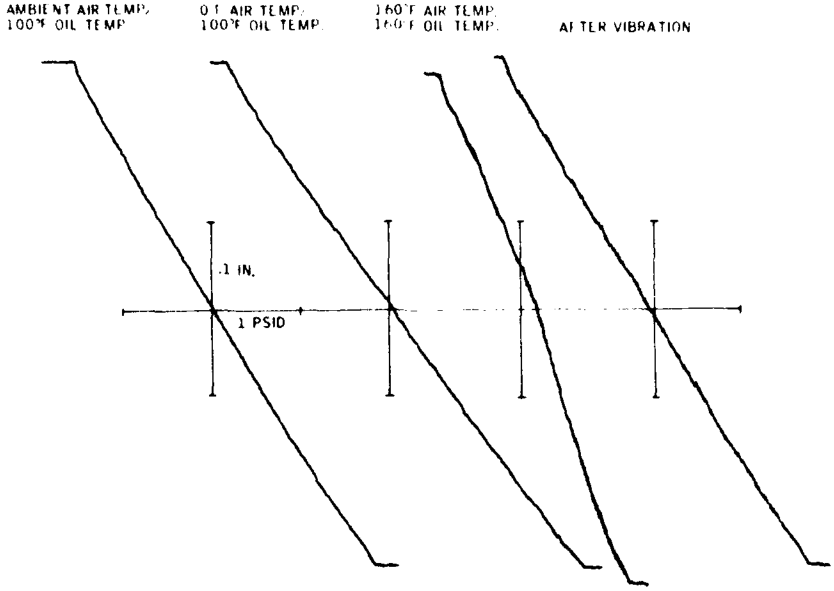

Figure 26. Servoactuator Gain Curves (Final Configuration).

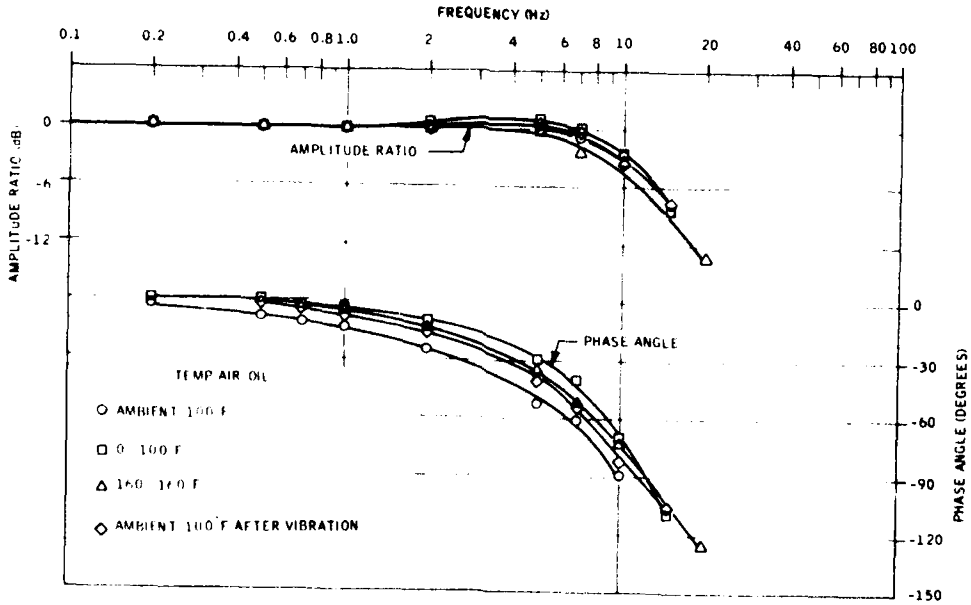

Figure 27. Servoactuator Frequency Response (Final Configuration).

ENVIRONMENTAL TESTS

Vibration Test Per Test Plan

A vibration test was run per Procedure I, Curve B of Figure 514-1 of MIL-STD-810B. During exposure, the null of the actuator was monitored, and no shift was observed. In addition, the servoactuator was exposed to Procedure I, Curve M and was subsequently performance tested. The results of this performance test are included in Table 2. Performance did not change (within instrumentation accuracy)

Temperature Test Per Test Plan

The temperature test included exposure to -20°F air temperature with 100°F oil temperature. Null and null hunting were observed and the results are included in Table 2.

The second step of the temperature test included stabilizing the air temperature at 0°F and the oil temperature at 100°F. A full performance test was run, and these results are also tabulated in Table 2, and gain and response are plotted in Figures 26 and 27.

Finally, the servoactuator was exposed to 160°F air and 160°F oil temperature. Performance tests were run, and the results again are included in Table 2, Figure 26, and Figure 27.

SUPPLEMENTARY TESTS

One additional test was run to demonstrate the compatibility of the hydrofluidic actuator with a hydrofluidic Stability Augmentation System (SAS). Both the actuator and the hydrofluidic SAS that it was connected to were developed on Contract DAAJ02-72-C-0051. Only the servovalve was different. A frequency response test was run, and this data is shown in Figure 28. It can be observed that the servoactuator and SAS system show no sign of peaking at 2 Hz as observed with the standard servovalve. The response of the hydrofluidic actuator provides a much more desirable total response, and as a result, the system output noise is reduced dramatically. In the earlier test run with a bellows input servovalve, the system noise was 0.039 inch peak to peak at the actuator output compared with 0.005 inch peak to peak with the hydrofluidic servovalve. These tests were run at normal operating conditions of room ambient air and 120°F oil temperature.

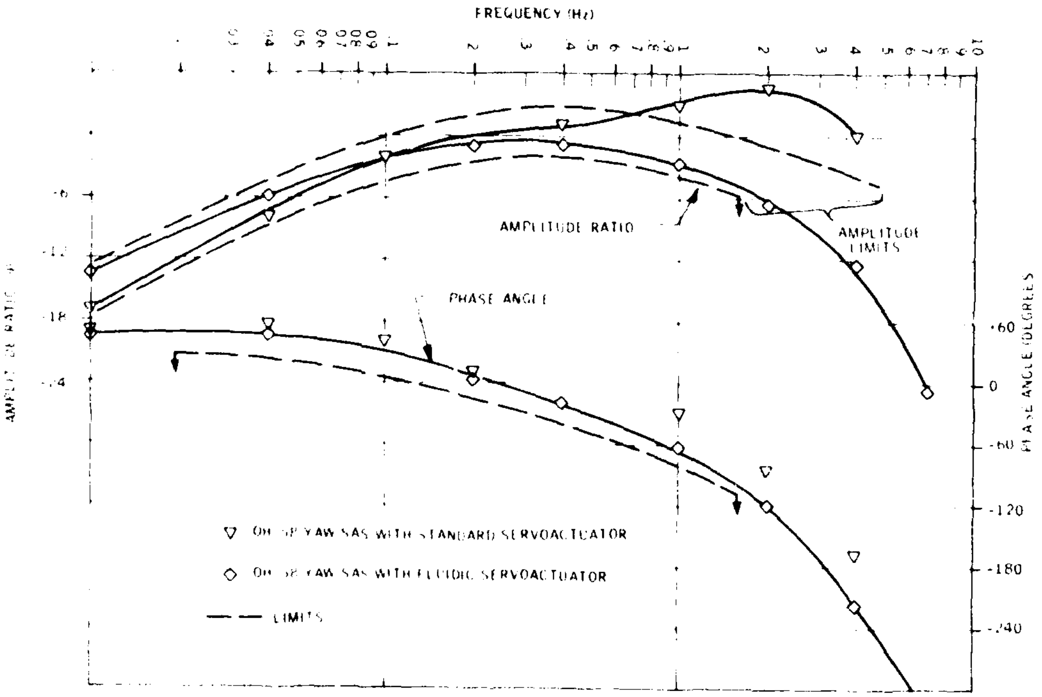

Figure 28. OH-58 Yaw SAS Frequency Response.

SECTION 5
CONCLUSIONS

The final performance of the servoactuator was very close to satisfying the design goals. Where slight variations from the goals are noted, such as frequency response and maximum ram velocity, redesign could readily eliminate these problems. The only significant problem is the temperature sensitivity of the flapper-nozzle feedback transducer. It was not possible to fully analyze this problem within this program, but a redesign, with attention given to the viscous effect type pathways in the feedback transducer, would eliminate all or most of the changes in feedback gain.

A flight test of the OH-58 yaw SAS using this servovalve should be considered to further prove its compatibility. A part of such a program should be a redesign of the feedback transducer to minimize sensitivity to viscosity.

APPENDIX A
TEST PLAN FOR A HYDROFLUIDIC SERVOVALVE UTILIZING FLUIDIC AMPLIFIER FIRST STAGE AND FLUIDIC FEEDBACK SUMMING

1.0 SCOPE

This test plan outlines the procedures to be followed for conducting the performance tests on a hydrofluidic servovalve, utilizing fluidic amplifier first stage and fluidic feedback summing. These tests shall consist of static, dynamic, and limited environmental tests to determine a servoactuator performance with the servovalve. The actuator will be provided as Government-furnished property from Contract DAAJ02-73-C-0056 (or equivalent).

2.0 APPLICABLE DOCUMENTS

MIL-H-5440 E	Hydraulic System, Aircraft Types I and II, Design, Installation, and Data Requirements for
MIL-G-5514 F	Packings, Installation and Gland Design, Hydraulic, General Specification for
MIL-H-5606 G	Hydraulic Fluid, Petroleum Base, Aircraft and Ordnance
MIL-P-25732 B	Packing, Preformed, Petroleum Hydraulic Fluid Resistant 275°F
MIL-STD-810B	Environmental Test Methods

3.0 TEST REPORTS

3.1 Performance test reports shall be of standard Honeywell AEX format.

3.2 DCAS witnessing will not be required for the testing.

4.0 TEST ITEM

A servoactuator assembly 30004830 used on Contract DAAJ02-73-C-0056 will be used as a test bed for the servovalve tests. A fluidic amplifier driven by an electrical-to-fluidic interface valve will provide the initial input signal for testing of the servovalve and servoactuator. Tests will also be run using the fluidic controller output presently on the servoactuator.

5.0 PERFORMANCE REQUIREMENTS

The performance requirements shall be used as design goals for the servovalve tests. They are specified in the servovalve specification of Appendix B; titles and paragraph numbers are listed below:

Leakage	3.2.1.2
Flow Gain	3.2.1.3.1
Pressure Gain	3.2.1.3.2
Null	3.2.1.3.3
Linearity	3.2.1.3.4
Threshold	3.2.1.3.5
Neutral Cyl. Port Pressure	3.2.1.3.6
Dynamic Response	3.2.1.4
Servoactuator Performance	3.2.1.5

Temperature	3. 2. 1. 6. 1
Vibration	3. 2. 1. 6. 2
Proof Pressure	3. 2. 2. 2

6. 0 STANDARD TEST CONDITIONS

6. 1 Standard test conditions are defined as:

Hydraulic Fluid:	MIL-H-5606
Supply Pressure:	600 ± 50 psig
Return Pressure:	0 to 50 psig
Signal Reference Pressure:	100 ± 5 psig
Signal Quiescent Pressure:	5 ± 1 psig above the reference pressure level
Hydraulic Fluid Temperature:	100 ± 10°F
Ambient Temperature:	80 ± 10°F

6. 2 The following standard instrumentation will be used, as appropriate, for the testing:

Turbine flowmeter with signal conditioner
Counter for flowmeter readout
Servoactuator position readout equipment
Bafco frequency response analyzer
Low-frequency function generator
Pressure transducers and gages
X-Y plotter
Thermocouples and temperature potentiometer
Sanborn recorder

7.0 TESTING

7.1 General

Except for proof pressure, all tests shall be conducted with the servovalve on the servoactuator. For those tests which require measurements of the servovalve outputs, an adaptor block shall be used between the valve and the actuator.

Tests not adequately described in the Servovalve Specification (Appendix B) with the requirements are clarified below.

7.2 Test Schedule

Testing shall be conducted in the approximate sequence of the listing below (specification paragraphs are noted).

7.3 Leakage (3.2.1.2)

7.4 Flow Gain (3.2.1.3.1)

7.5 Pressure Gain (3.2.1.3.2)

7.6 Null (3.2.1.3.3)

7.7 Linearity (3.2.1.3.4)

7.8 Threshold (3.2.1.3.5)

7.9 Neutral Cylinder Port Pressure (3.2.1.3.6)

7.10 Dynamic Response (3.2.1.4)

7.11 Servoactuator Performance (3.2.1.5)

7.11.1 Gain (3.2.1.5.1)

7.11.2 Linearity (3.2.1.5.2)

7.11.3 Threshold (3.2.1.5.3)

7.11.4 Stall Load (3.2.1.5.4)

7.11.5 Saturation Velocity (3.2.1.5.5)

7.11.6 Null Bias (3.2.1.5.6)

7.11.7 Null Shift (3.2.1.5.7)

The temperature test portion of this procedure shall be performed just once during the temperature test of paragraph 7.13 below.

7.11.8 Dynamic Performance (3.2.1.5.8)

7.11.8.1 Amplitude Ratio and Phase Angle (3.2.1.5.8.1)

7.11.8.2 Null Hunting (3.2.1.5.8.2)

7.12 Proof Pressure (3.2.2.2)

7.13 Temperature (3.2.1.6.1)

Mount the servoactuator in a temperature chamber and stabilize at -20°F ambient. Energize the system and check for null (3.2.1.5.7) and null hunting (3.2.1.5.8.2).

With the system de-energized, stabilize the ambient temperature at 0°F. Energize and record servoactuator performance (above paragraph 7.11). Increase the ambient and oil temperature to 160°F and repeat the performance tests.

7.14 Vibration (3.2.1.6.2)

Mount the servoactuator on the vibration driver so that it can be vibrated in each of the three mutually perpendicular axes. Energize the actuator with the input signal ports at normal pressure level. Run a vibration scan for 15 minutes in each of the three axes according to Procedure I, Curve B (2g), Figure 514-1 of MIL-STD-810B. Check Null Bias per DS paragraph 3.2.1.5.6.

Remove the pressure and return lines from the actuator and cap off the fittings with the actuator full of oil. Run the vibration scans according to Curve M (5g).

Repeat the servoactuator performance tests (above paragraph 7.11).

APPENDIX B

SPECIFICATION FOR A HYDROFLUIDIC SERVOVALVE UTILIZING FLUIDIC AMPLIFIER FIRST STAGE AND FLUIDIC FEEDBACK SUMMING

1.0 SCOPE

This specification establishes the requirements for a fluidic input hydromechanical servovalve to be used in a hydrofluidic stability augmentation system (HYSAS). The servovalve shall utilize fluidic amplifiers to drive the spool valve in place of the bellows-driven flapper-nozzle configuration. Mounting configuration and interfacing shall be compatible with the servoactuator of Contract DAAJ02-73-C-0056, or similar.

1.1 Classification

The servovalve described herein shall be classified as experimental. Performance and configuration requirements shall be design goals and shall be compatible with the servoactuator requirements of Contract DAAJ02-73-C-0056.

2.0 APPLICABLE DOCUMENTS

The following documents of the issue in effect on the date of this specification shall form a part of this specification to the extent specified herein. In the event of conflict between a referenced document and this specification, this specification shall be considered a superceding requirement.

MIL-H-5440E	Hydraulic System, Aircraft Types I and II, Design, Installation and Data Requirements for
MIL-G-5514F	Packings, Installation and Gland Design, Hydraulic, General Specification for
MIL-H-5606G	Hydraulic Fluid, Petroleum Base, Aircraft and Ordnance
MIL-P-25732B	Packing, Preformed, Petroleum Hydraulic Fluid Resistant 275°F
MIL-STD-810B-4	Environmental Test Methods

3.0 REQUIREMENTS

3.1 Item Definition

The subject item shall be a two-stage, four-way flow-control servovalve that provides control flow (at constant load) which is proportional to the applied input signal pressure. The first stage shall be a fluidic amplifier mechanization with the output driving a spool valve. Flow from the servovalve is used to drive a hydraulic actuator. Feedback shall be provided from the actuator output piston position in the form of a fluidic signal which is summed into the amplifier circuit.

3.1.1 Block Diagram

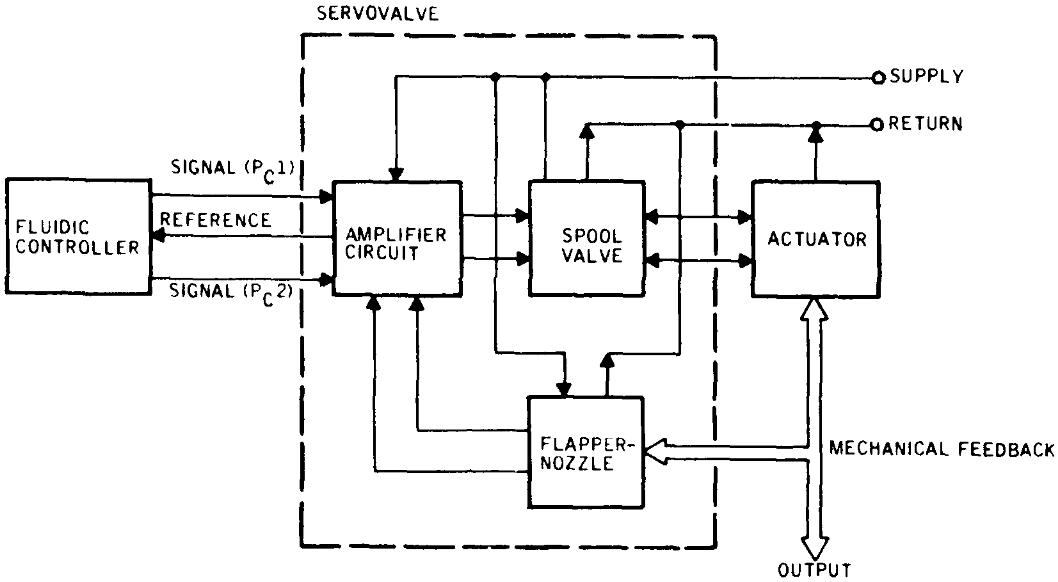

Figure B-1. Servovalve Block Diagram.

3.1.2 Servovalve Features and Physical Interface

The servovalve shall incorporate the following features and physical interface.

(a) Housing

(b) Provision for mounting to and interfacing with the servo-actuator assembly 30004830 used on Contract DAAJ02-73-C-0056.

(c) The interface specified in (b) shall include hydraulic supply, return and cylinder ports plus a fluidic reference port and two fluidic signal ports.

(d) Position feedback mechanism to be driven by the piston of the servoactuator of (b).

(e) Polarity shall be such that when a differential pressure is applied with $P_{c1} > P_{c2}$, the servoactuator shall retract (-).

3.1.3 Hydraulic Supply

The subject equipment shall be designed for operation in a Type I (-65° to +160°F), Class 1500 psi hydraulic system conforming to MIL-H-5440.

3.2 Characteristics

3.2.1 Performance

3.2.1.1 Rated Test Conditions

The servovalve shall be tested under the following conditions unless otherwise specified.

Hydraulic Fluid:	MIL-H-5606
Supply Pressure:	600 ± 50 psig
Return Pressure:	0 to 50 psig
Signal Reference Pressure:	100 ± 3 psig above return pressure
Signal Quiescent Pressure:	5 ± 1 psig above the reference pressure level
Hydraulic Fluid Temperature:	100 ± 10°F
Ambient Temperature:	80 ± 10°F

3.2.1.2 Leakage

3.2.1.2.1 External Leakage

There shall be no evidence of external leakage, other than a slight wetting at seals insufficient to form a drop.

3.2.1.2.2 Internal Leakage

The servovalve internal leakage from power supply to return shall not exceed 1.0 cis at rated test conditions and any value of fluidic signal within the rated control range.

3.2.1.3 Static Performance

3.2.1.3.1 Flow Gain

The servovalve no-load flow gain shall be 6.0 cis ±10% per psid of signal pressure. Rated flow shall be ±0.30 cis +10% with ±0.05 psid signal pressure applied to the input of the preamplifier.

3.2.1.3.2 Pressure Gain

To be determined (TBD)

3.2.1.3.3 Null

3.2.1.3.3.1 Null Bias

The null bias shall not exceed ±0.2 psid signal pressure. Null bias is defined as hydraulic null where both cylinder pressures are equal.

3.2.1.3.3.2 Null Shift

3.2.1.3.3.2.1 <u>With Temperature Variation</u>

The null shift (from the null bias at standard test conditions) shall not exceed ±0.20 psid signal pressure over the temperature range of -20° to 180°F.

3.2.1.3.3.2.2 <u>With Supply Pressure Variation</u>

The null shift (from the null bias at standard test conditions), with a variation of supply pressure of ±20%, shall not exceed ±0.04 psid signal pressure.

3.2.1.3.3.2.3 <u>With Return Pressure Variation</u>

The null shift (from the null bias at standard test conditions), with a variation of return pressure of 50 psig, shall not exceed 0.04 psid.

3.2.1.3.3.2.4 <u>With Signal Quiescent Pressure Level</u>

The null shift (from the null bias at standard test conditions), with a variation in signal quiescent pressure level (above a signal reference pressure), shall not exceed 0.04 psid per psid signal pressure between 4 psi to 6 psi.

3.2.1.3.4 <u>Linearity</u>

All the gain curve test points shall fall within a ±0.12 psid band of the best straight line through these test points.

3.2.1.3.5 <u>Threshold</u>

The threshold shall not exceed 0.02 psid signal pressure as measured from the pressure gain curve.

3.2.1.3.6 Neutral Cylinder Port Pressure

Neutral cylinder pressure shall be 50% ±15% of supply pressure.

3.2.1.4 Dynamic Response

The nominal servovalve transfer function with signal input of ±0.20 ±0.04 psid applied to the preamplifier input shall be as shown below. The servovalve dynamic response shall be within the limits shown in Figure B-2.

$$\frac{Q}{P} = \frac{6.0 \quad (157)^2}{S^2 + 2\,(0.9)\,(157)\,S + (157)^2} \quad \frac{\text{cis}}{\text{psid}}$$

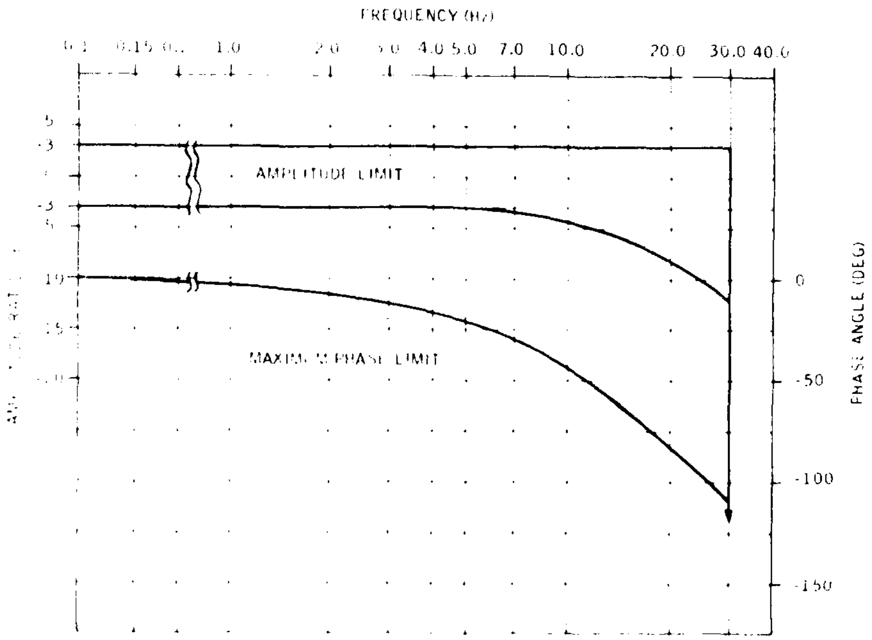

Figure B-2. Servovalve Frequency Response.

3.2.1.5 Servoactuator Performance

3.2.1.5.1 Gain

No load stroke for ±2 psid input shall be ±0.300 ±0.015 inch.

3.2.1.5.2 Linearity

All test points shall fall within a ±0.015 inch band of the best straight line drawn through the position curve. This requirement also includes the hysteresis.

3.2.1.5.3 Threshold

The threshold shall not exceed 0.02 psid input signal.

3.2.1.5.4 Stall Load

The stall load shall be greater than 80 pounds.

3.2.1.5.5 Saturation Velocity

The servoactuator shall be capable of moving at 1.0 to 2.4 in/sec minimum.

3.2.1.5.6 Null Bias

The null bias shall not exceed ±0.03 inch, with the input signal at null.

3.2.1.5.7 Null Shift

The null shift due to temperature, supply pressure, and return pressure variations shall be as follows:

Temperature: < 0.03 in. (-20° to +180°F)
Supply Pressure: < 0.006 in. (600 ± 120 psig)
Return Pressure: < 0.006 in. (0 to 100 psig)
Quiescent Signal Pressure: < 0.006 in. (5 ± 1 psi)

3.2.1.5.8 Dynamic Performance

3.2.1.5.8.1 Amplitude Ratio and Phase Angle

The response of the servoactuator when run at ±0.06 inch shall meet the requirements of Figure B-3. The driving device shall have an output impedance of 80 lb-sec/in^5.

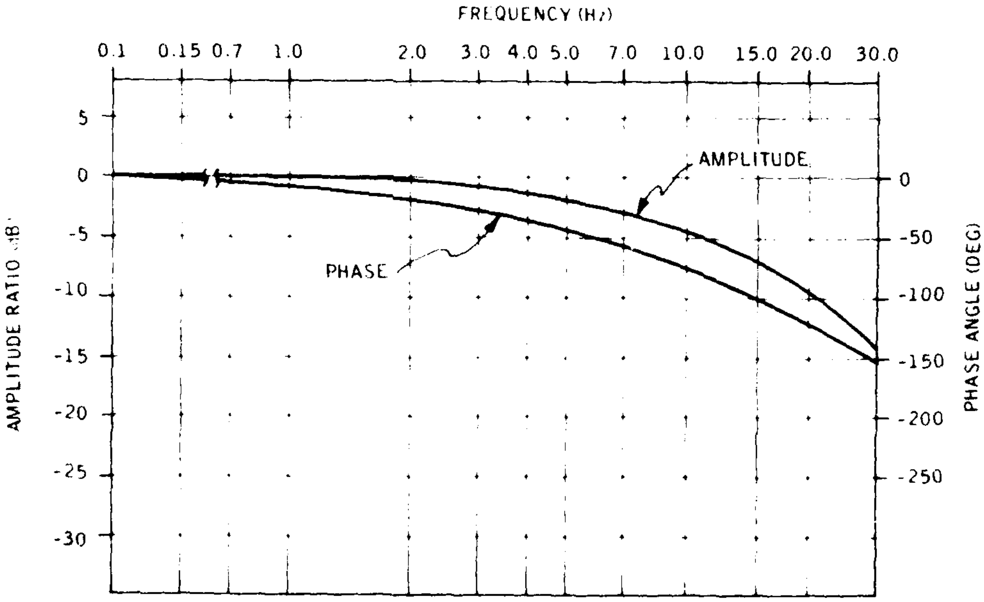

Figure B-3. Servoactuator Frequency Response.

3.2.1.5.8.2 Null Hunting

The hunting of the output shaft shall not exceed 0.018 inch.

3.2.1.6 Environmental Requirements

3.2.1.6.1 Temperature

The servoactuator shall function with degraded performance at temperatures below 0°F.

Below 0°F, the servoactuator shall meet 3.2.1.5.7 and 3.2.1.5.8.2. From 0° to 180°F, the servoactuator shall meet 3.2.1.5.

3.2.1.6.2 Vibration

The servoactuator shall meet the requirements of 3.2.1.5.6 when subjected to Procedure I, Curve B of Figure 514-1 of MIL-STD-810B. The servoactuator shall also withstand the Procedure I, Curve M and meet the requirements of 3.2.1.5 after exposure.

3.2.2 Physical Characteristics

3.2.2.1 Envelope and Interface

The servovalve envelope shall be held to a minimum to be compatible with mounting on servoactuator assembly 30004830 used on Contract DAAJ02-73-C-0056.

3.2.2.2 Proof Pressure

The servovalve shall withstand the following proof pressures without evidence of external leakage or permanent performance degradation.

a) 900 psi applied to the servovalve supply port with (1) the return port and load ports open and (2) with the load ports blocked.

b) 600 psi applied to the servovalve fluidic input ports with the reference port blocked. A differential of 25 psid across the fluidic input ports with the reference port open.

The rate of pressure application shall not exceed 25,000 psi/min.

3.2.2.3 Burst Pressure

The servovalve shall not rupture when subjected to the following burst pressures.

a) 1000 psi applied to the servovalve fluidic input ports with the reference port blocked.

b) 1500 psi applied to the servovalve supply port with the cylinder ports blocked and the return port at 1000 psi.

3.2.2.4 Seal Glands

Seal glands shall be in accordance with MIL-G-5514, wherever possible.

3.2.2.5 Seals

Seals shall conform to MIL-P-25732, wherever applicable.

3.2.2.6 Safetying

All threaded ports shall be securely locked or safetied with safety wire, self locking nuts, or other approved methods. Safety shall be applied in accordance with MS33540.

3.2.2.7 Driving Fluidic Amplifier

The output impedance of the driving amplifier will be 80 lb-sec/in.^{5}

3.2.2.8 Environmental Requirements

3.2.3.1 Temperature

The servovalve shall meet the requirements of 3.2.1 when subjected to temperature from 0° to 180°F. From -20° to 0°F the servovalve shall function with degraded performance and shall meet the requirements of 3.2.1.2.

3.2.3.2 Vibration

The servovalve shall perform satisfactorily when subjected to Procedure I, Curve B of Figure 514-1 of MIL-STD-810B and also after being subjected to Procedure I, Curve M of the same specification.

4.0 QUALITY ASSURANCE PROVISIONS

4.1 Design Performance Evaluation

Evaluation shall be performed as required to demonstrate conformance with the following requirements.

Leakage	3.2.1.2
Flow Gain	3.2.1.3.1
Pressure Gain	3.2.1.3.2
Null	3.2.1.3.3
Linearity	3.2.1.3.4

Threshold	3.2.1.3.5
Neutral Cylinder Port Pressure	3.2.1.3.6
Dynamic Response	3.2.1.4
Servoactuator Performance	3.2.1.5
Proof Pressure	3.2.2.2
Temperature	3.2.1.6.1
Vibration	3.2.1.6.2

LIST OF SYMBOLS

A_S	Area of spool valve end
b	Flex pivot spring thickness, in.
C	Distance to farthest fiber, in.
C_D	Coefficient of discharge
cis	in.^{3}/sec
D_A	Effective orifice diameter of amplifier, in.
dB	Decibel - 20 log (output/input)
D_N	Diameter of nozzle, in.
D_O	Diameter of bridge orifice, in.
D_S	Diameter of flapper-nozzle spring, in.
E	Modulus of elasticity
f	Maximum deflection of spring, in.
G	Torsional modulus
G_V	Spool valve gain, $\frac{\text{in.}^3/\text{sec}}{\text{in.}}$
h	Flex pivot spring width, in.
I	Moment of inertia, in.^{4}
K_A	Actuator gain, in./in.^{3}
K_{FB}	Feedback gain, psi/in.
K_{POS}	Actuator position gain, in./psi
K_S	Spool valve springs - spring rate, lb/in.
K_V	Spool valve flow-pressure gain, in.^{3}/sec/psi
K_1	Preamplifier gain, psi/psi
K_2	Power amplifier gain, psi/psi

L	Flapper-nozzle gap, in.
ℓ	Flex pivot spring length, in.
M	Torque at maximum deflection, in.-lb
P_N	Nozzle pressure level, lb/in.^{2}
P_R	Flex pivot safe shear torque, in.-lb
P_S	Supply pressure, lb/in.^{2}
psi	lb/in.^{2}
psid	lb/in.^{2} differential
psig	lb/in.^{2} gage
$P_{1,2}$	Flapper-nozzle pressures, lb/in.^{2}
$Q_{A1,2}$	Flow into summing ports, in.^{3}/sec
$Q_{N1,2}$	Nozzle flow, in.^{3}/sec
Q_{SM}	Maximum flow into spool valve, in.^{3}/sec
Q_T	Total flapper-nozzle circuit flow, in.^{3}/sec
$Q_{1,2}$	Orifice bridge flow, in.^{3}/sec
S	Laplacian operator
S_S	Stress in spring wire, lb/in.^{2}
S_V	Material safe shear stress, lb/in.^{2}
T	Time constant, sec
X	Actuator displacement, in.
x	Flapper-nozzle motion, in.
Y	Length of flapper spring, in.
ω	Natural frequency, rad/sec

δ Damping ratio

ΔP Pressure differential, $lb/in.^2$

ΔP_M Maximum flapper-nozzle differential pressure, $lb/in.^2$

ΔP_{PO} Power amplifier range, psid

ΔP_{PR} Preamplifier range, psid

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