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HYDROFLUIDIC SERVOACTUATOR DEVELOPMENT

Holger C. Kent, et al

Honeywell, Incorporated

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EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
FORT EUSTIS, VIRGINIA

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This report has been reviewed by the Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory and is considered to be technically sound.

The purpose of the program was to demonstrate the feasibility of a servoactuator using a hydrofluidic amplifier cascade input stage which replaces the bellows-flapper-nozzle of a conventional servovalve, a fluid feedback transducer, and an actuator. The report is published for the exchange of information and appropriate application.

The technical monitor for this contract was Mr. George W. Fosdick, Technology Applications Division.

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HYDROFLUIDIC SERVOACTUATOR DEVELOPMENT

Final Report

Honeywell Document W0510

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Prepared by

Honeywell Inc.
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FOREWORD

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 (AMRDL), Fort Eustis, Virginia, under Contract DAAJ02-72-C-0017. The technical monitor on this program was Mr. George Fosdick.

The objective of this program was to design and develop a hydrofluidic servoactuator, with increased reliability and reduced cost, by utilizing hydrofluidic amplifiers and fluid feedback to replace the conventional servovalve flapper-nozzle first-stage and mechanical feedback. The work effort presented was conducted over a period from 18 November 1971 to 31 October 1972.

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LIST OF SYMBOLS

ACD	-	Apparatus Control Division
d	-	Spool dia., in.
db	-	Decibel - 20 log (output/input)
K_A	-	Actuator gain, in./in.^{3}
K_{FB}	-	Feedback gain, psi/in.
K_V	-	Spool valve gain, in.^{3}/sec/psi
K_1	-	Preamplifier gain, psi/psi
K_2	-	Power amplifier gain, psi/psi
ℓ	-	Depth of capillary, in.
ℓ_P	-	Length of capillary, in.
ℓ_r	-	Restricted length of spool, in.
N	-	Spool threads per inch
P_F	-	Feedback transducer supply pressure, lb/in.^{2}
P_{FB}	-	Feedback transducer output pressure, lb/in.^{2}
P_S	-	Servoactuator supply pressure, lb/in.^{2}
P_{SYS}	-	Control system supply pressure, lb/in.^{2}
S	-	LaPlace operator
T	-	Time constant, sec
X	-	Actuator displacement, in.
ω	-	Natural frequency, rad/sec
ζ	-	Damping ratio
ΔP	-	Pressure differential, lb/in.^{2}
μ	-	Viscosity, lb/sec/in.^{2}

SECTION I

SERVOACTUATOR DESIGN

The program consisted of the design and development of a hydrofluidic servoactuator laboratory model utilizing high-pressure fluidic amplifiers to replace the flapper-nozzle of a conventional servovalve, and designed to meet the performance requirements of a UH-1 helicopter.

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 which drives the spool valve directly. Fluidic summing is provided in the feedback loop, eliminating the mechanical feedback linkage used on present devices. Figure 1 shows a circuit schematic of the system. Design and performance requirements are based on the performance requirements of a UH-1 helicopter and the utilization of MIL-H-5606 hydraulic fluid.

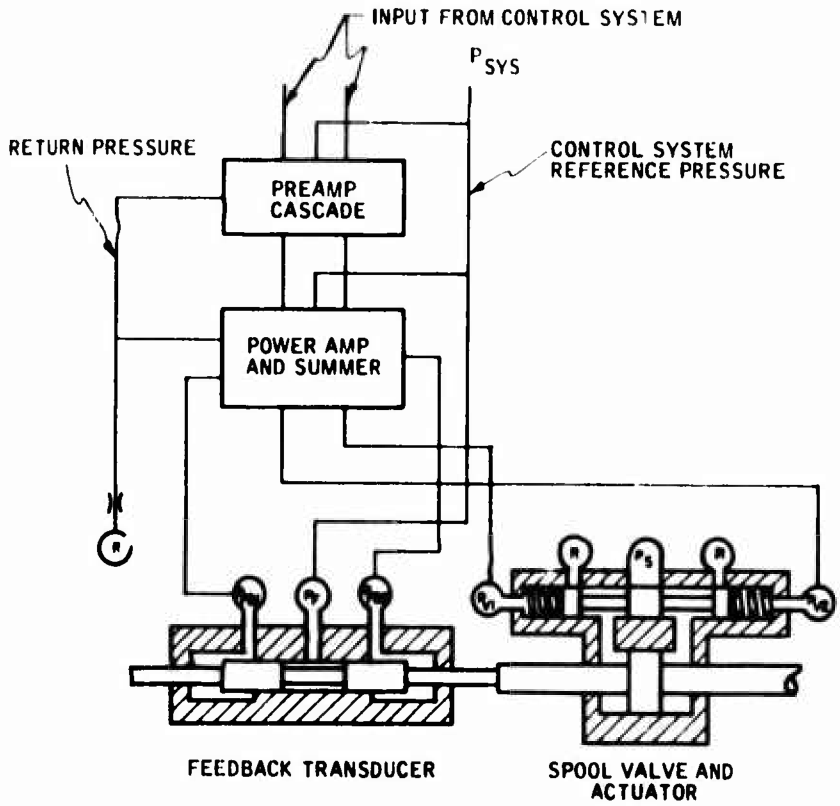

Figure 1. Circuit Schematic.

The following performance requirements were used as design goals for the series servoactuator:

Quiescent control pressure	6^{+9}_{-2} psig above return
Quiescent control flow	0.1 $in.^3$/sec
Input range	±4 psid
Supply pressure	1500 psig
Quiescent power flow	1.5 $in.^3$/sec (0.39 gpm)
Actuator piston stroke	±0.375 in.
Slew rate	10 in./sec
Static gain	0.093 in./psi
Response	
Amplitude ratio	+2 db max, -6 db max attenuation at 10 Hz
Phase lag	90 deg max at 10 Hz

RESPONSE ANALYSIS

The block diagram for the mechanization of the servoactuator is shown in Figure 2 where

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.)

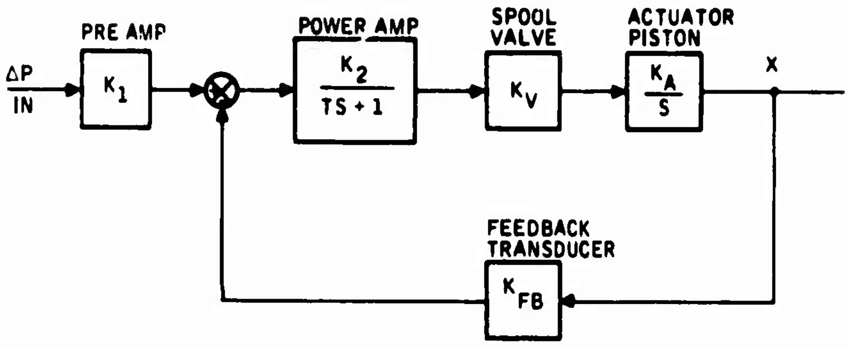

Figure 2. Servoactuator Mechanization.

Typical values for present UH-1 components are as follows:

Spool Valve

K_V	=	0.0066 in.^{3}/sec/psi
Diameter	=	0.187 in.
Spring rate	=	700 lb/in. (400-700 lb/in.)
Gain	=	168 in.^{3}/sec/in.
Capacitance	=	1.09×10^{-6} in.^{3}/psi

Actuator

K_A	=	6.06 in./in.^{3}
Piston area	=	0.165 in.^{2}
Stroke	=	±0.375 in.

Assume K_2 = 10 psi/psi

ω = 94 rad/sec (15 Hz)

T = 0.007 (Approximate lag between amplifier and spool valve)

$$\frac{X}{\Delta P} = \frac{\dfrac{K_2 K_V K_A}{S(TS+1)}}{1+\dfrac{K_2 K_V K_A K_{FB}}{S(TS+1)}} = \frac{K_2 K_V K_A}{S(TS+1) + K_2 K_V K_A K_{FB}} \tag{1}$$

$$\frac{X}{\Delta P} = \frac{K_2 K_V K_A}{TS^2 + S + K_2 K_V K_A K_{FB}} = \frac{\dfrac{K_2 K_V K_A}{T}}{S^2 + \dfrac{S}{T} + \dfrac{K_2 K_V K_A K_{FB}}{T}} \tag{2}$$

The damping ratio becomes

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

$$\delta = \frac{1}{2\omega T} = \frac{1}{2(94)\,(0.007)} = 0.76 \tag{3}$$

and the feedback gain becomes

$$\omega^2 = \frac{K_2 K_V K_A K_{FB}}{T}$$

$$K_{FB} = \frac{\omega^2 T}{K_2 K_V K_A} = \frac{(94)^2\,(0.007)}{10\,(0.0066)\,(6.06)} = 155 \text{ psi/in.} \tag{4}$$

The spool valve gain (K_V) and amplifier gain (K_2) can be varied by changing the spool valve centering spring rate and number of amplifiers, respectively, thus providing for a variation in the feedback gain (K_{FB}). Figure 3 shows the required response envelope and that calculated from the design parameters.

The saturation range requirements of the preamplifier and power amplifier cascades are determined from the maximum slew rate required for the servoactuator.

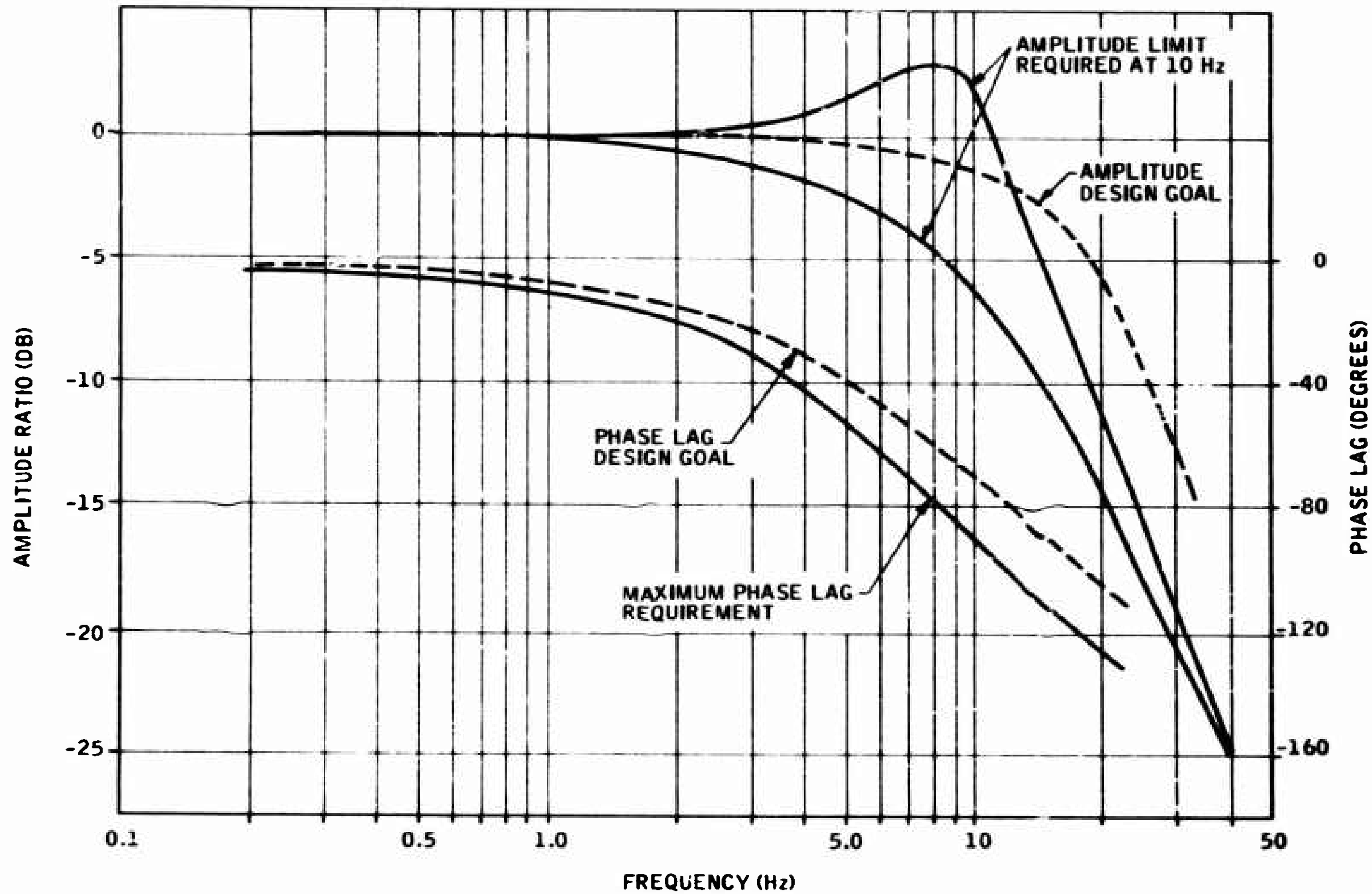

Figure 3. Response Requirement.

$$\text{Maximum flow rate into spool valve} = \frac{\text{Max slew rate}}{\text{Actuator gain } (K_A)}$$

$$= \frac{10 \text{ in./sec}}{6.06 \text{ in./in.}^3}$$

$$= 1.65 \text{ in.}^3\text{/sec}$$

$$\text{Power amplifier range} = \frac{\text{Max flow rate into spool valve}}{\text{Spool gain } (K_V)}$$

$$= \frac{1.65 \text{ in.}^3\text{/sec}}{0.0066 \text{ in.}^3\text{/sec/psi}}$$

$$= 250 \text{ psid}$$

$$\text{Preamplifier range} = \frac{\text{Power amplifier range}}{\text{Power amplifier gain } (K_2)}$$

$$= \frac{250}{10} \cong 25 \text{ psid}$$

SECTION II

HARDWARE DESIGN AND FABRICATION

GENERAL

Hardware design was directed at providing hydrofluidic preamplifier and power amplifier cascades which could be manifolded and mounted directly to the servovalve mounting surface of a servoactuator. The servoactuator (85112010, SN006) is one previously used on the Three-Axis Fluidic Stability Augmentation System Flight Test (USAAMRDL Technical Report 71-34). This actuator was also modified to accept a fluidic feedback transducer. No attempt was made to miniaturize the hardware, with the main emphasis being on the demonstration of feasibility and ease of manufacture. The assembled unit is shown in Figure 4. Figure 5 shows the amplifier manifold removed from the spool valve centering spring housing. Figures 6 and 7 show the main and individual amplifier manifold blocks. The servoactuator schematic is shown in Figure 8.

COMPONENT DESIGN

Amplifier

Amplifier design utilized for this application required a configuration which operated at higher differential pressures (500 psid) than in the past. Based on tests on various amplifiers, the amplifier configuration chosen was one developed by Apparatus Control Division (commercial) of Honeywell Inc. for high-pressure fluidics activity. Two basic amplifiers were used: a single-input amplifier (ACD-1) and a summing amplifier (ACD-2). Both amplifiers have a 0.020- x 0.020-inch power nozzle (Figure 9).

Feedback Transducer

The feedback transducer was designed using a capillary threaded spool for providing a push-pull linear differential pressure output to the summing amplifier. The null flow and pressure level was designed for approximately 600 psi level at a flow of 0.25 in.^{3}/sec and pressure differential of 100 psid to match the requirements of the summing amplifier.

Design parameters were determined by the use of the capillary flow equation

$$P_F - P_{FB} = \frac{3Q_{FB}\,\mu \ell_P\,(2\ell + 2w)^2}{(\ell w)^3} \qquad (5)$$

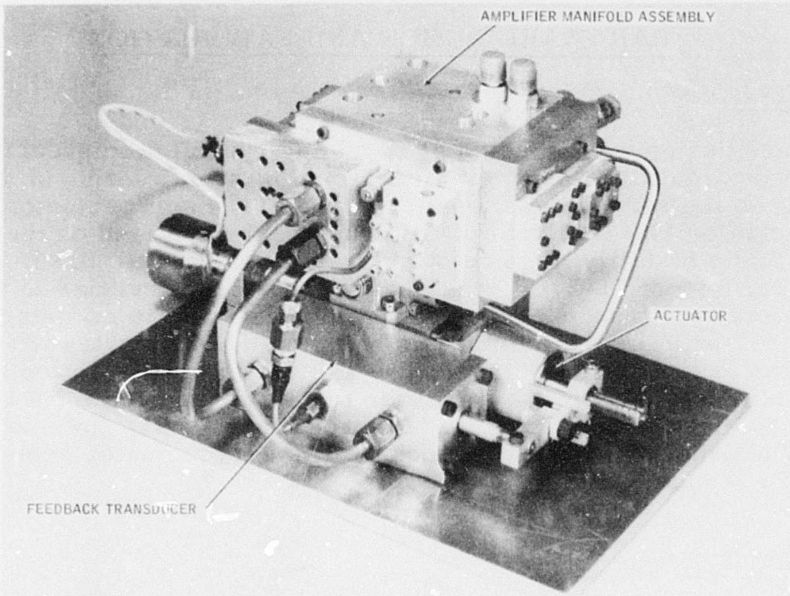

Figure 4. Servoactuator Assembly.

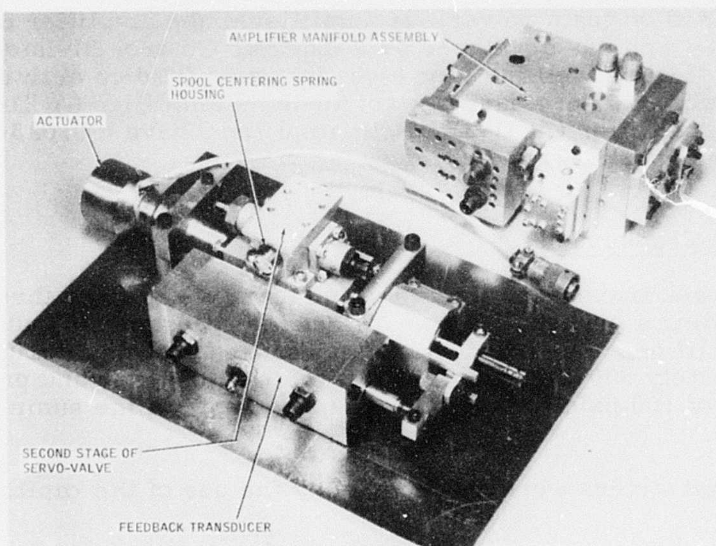

Figure 5. Servoactuator With Amplifier Manifold Removed.

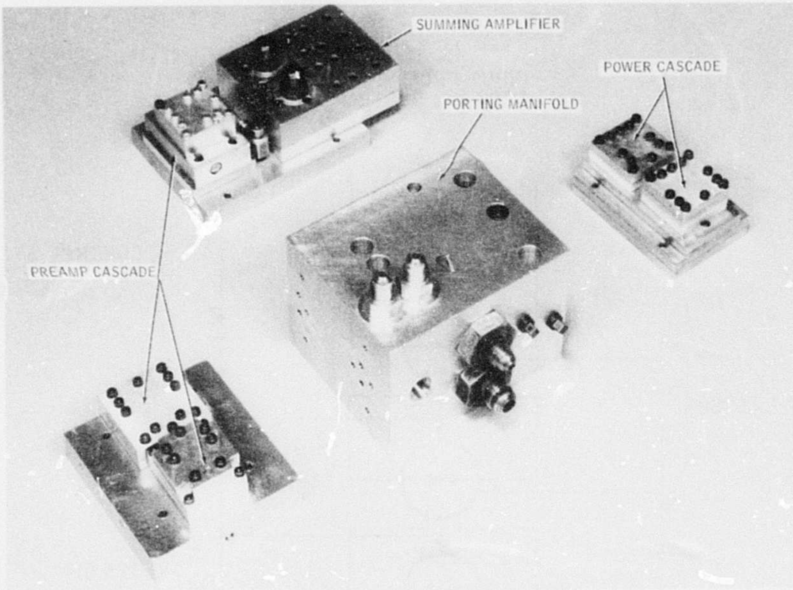

Figure 6. Amplifier Manifold Block.

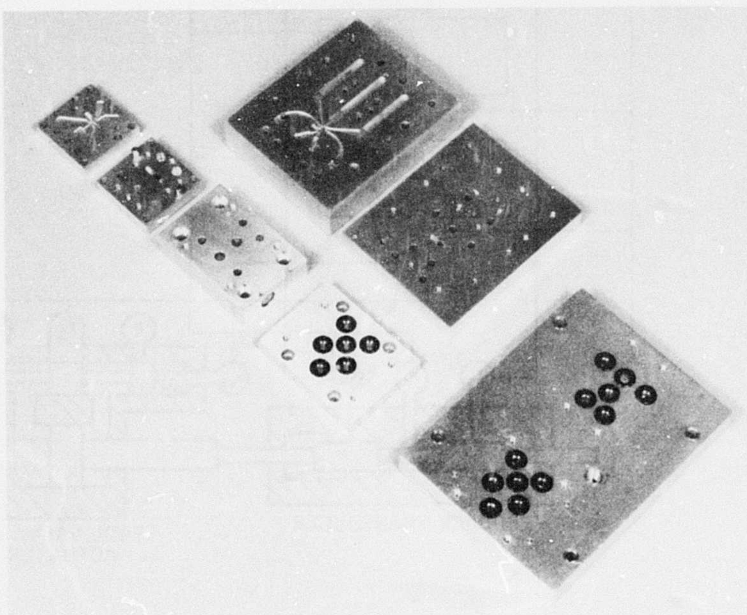

Figure 7. Typical Amplifier Manifolding.

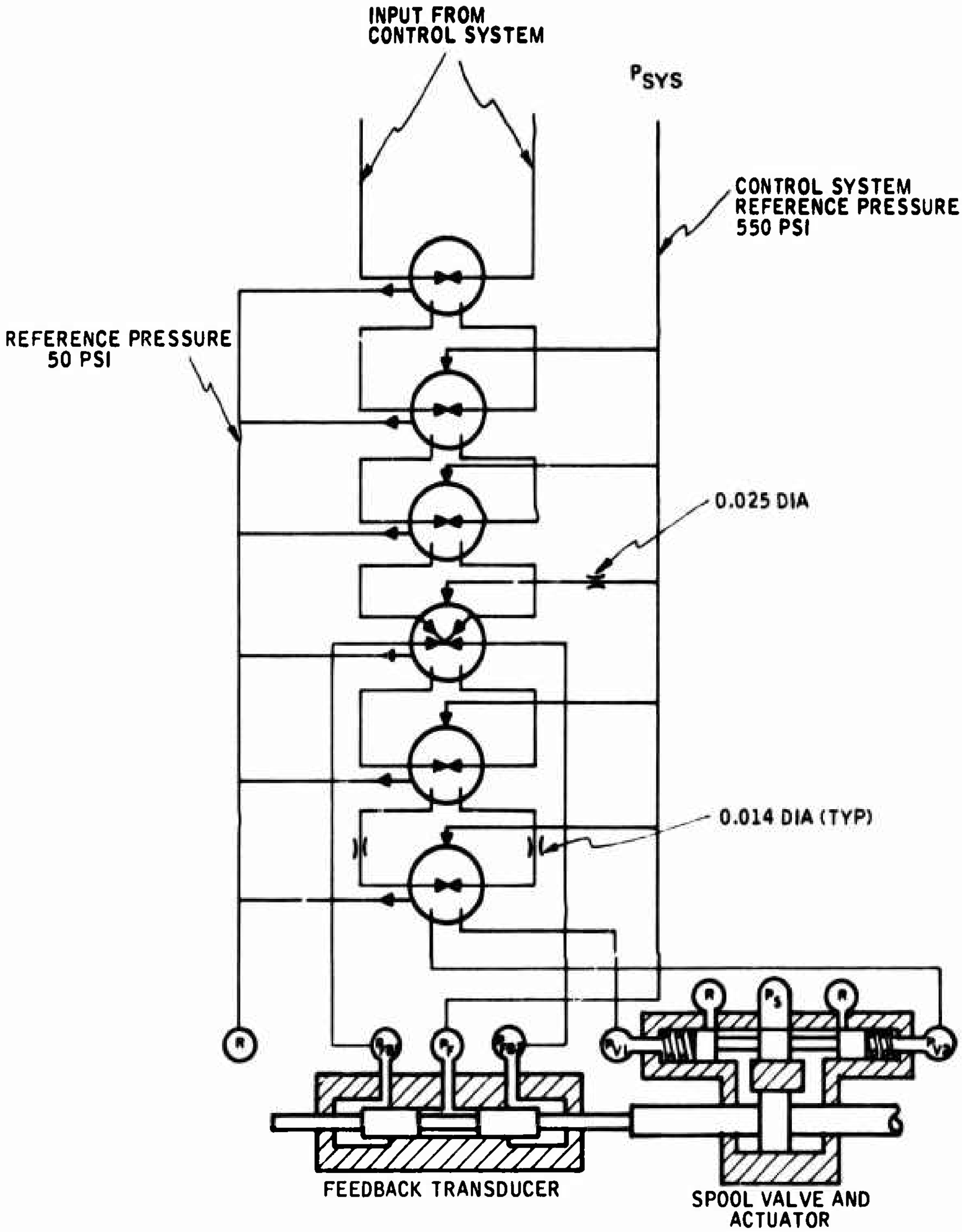

Figure 8 Servoactuator Schematic.

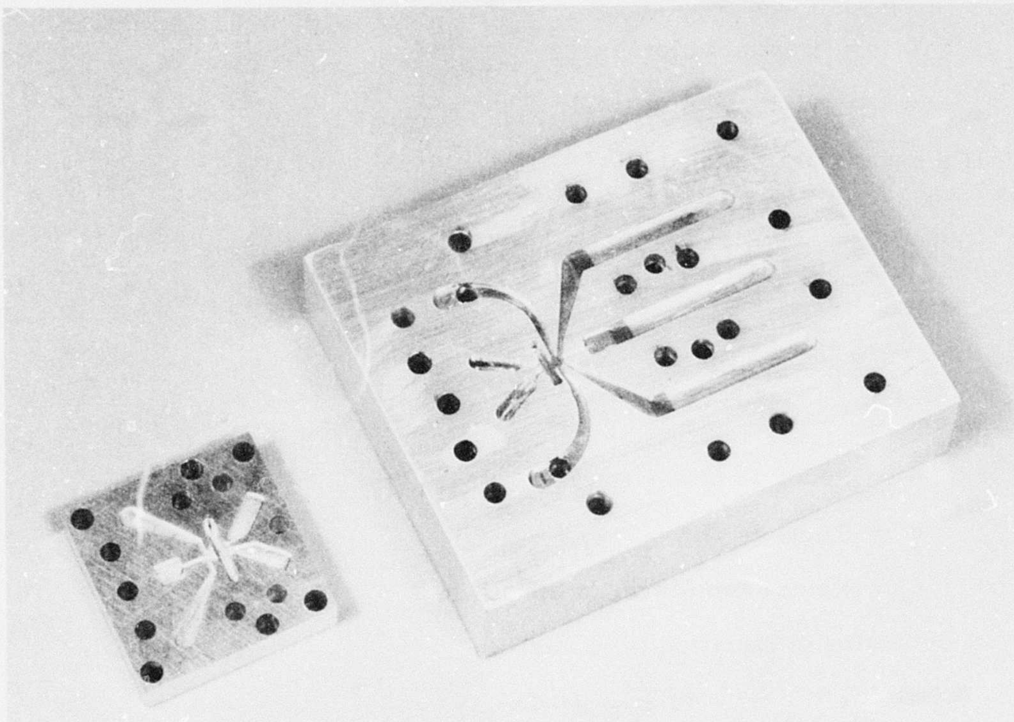

Figure 9. Hydrofluidic Amplifiers.

where

P_F = Supply pressure (lb/in.^{2})

P_{FB} = Output pressure (lb/in.^{2})

Q_{FB} = Flow (in.^{3}/sec)

μ = Viscosity (lb-sec/in.^{2})

ℓ_P = Length of capillary (in.)

ℓ = Depth of slot (in.)

w = Width of slot (in.)

If a square slot is used,

$$\ell = w$$

and

$$P_F - P_{FB} = \frac{48\, Q_{FB}\, \mu \ell_P}{w^4} \tag{6}$$

$$\ell_P = \pi\, dN\ell_r$$

where

d = Spool diameter (in.)

N = Threads /in.

ℓ_r = Restricted length of spool (in.)

Substituting in Equation (6) gives

$$P_F - P_{FB} = \frac{48\, Q_{FB}\, \mu \pi\, dN\ell_r}{w^4} \tag{7}$$

The groove size required to accommodate the amplifier flow at null is determined from

$$w^4 = \frac{48\, Q_{FB}\, \mu \pi\, dN\ell_r}{P_F - P_{FB}} \tag{8}$$

For

$Q_{FB} = 0.25 \text{ in.}^3/\text{sec}$

$\mu = 0.161 \times 10^{-5} \text{ lb-sec/in.}^2$

$d = 0.375 \text{ in.}$

$N = 20$

$\ell_r = 0.5 \text{ in.}$

$P_F = 1000 \text{ psi}$

$P_{FB} = 600 \text{ psi}$

$$w = \sqrt[4]{\frac{48\,(0.25)\,(0.161 \times 10^{-5})\,(3.14)\,(0.375)\,(20)\,(0.5)}{1000 - 600}}$$

$$w = \sqrt[4]{0.057 \times 10^{-5}} = 0.0275 \text{ in.}$$

With a selected feedback gain (K_{FB}) of 100 psi/in., the gain for each half of the spool will be 50 psi/in.; and for a stroke of ±0.375 in., the pressure differential from null to full stroke will be

$$50\,(0.375) = 18.8 \text{ psi}$$

Therefore, the pressure at the summing amplifier (P_{FB}) will be 18.8 psi less, or 581 psi. Correspondingly, the flow through the amplifier will be less. To determine the threads per inch (N) required for this position, we have

$$N = \frac{(P_F - P_{FB})\, w^4}{48\, Q_{FB}\, \mu \pi d \ell_r} \qquad (9)$$

where

$P_{FB} = 581 \text{ psi}$

$Q_{FB} = 0.225 \text{ in.}^3/\text{sec}$

$\ell_r = 0.375 + 0.5 = 0.875 \text{ in.}$

$$N = \frac{(1000 - 581)(0.027)^4}{48(0.275)(0.161 \times 10^{-5})(3.14)(0.375)(0.875)}$$

$$N = 0.000124 \times 10^{+5} = 12.4$$

A sketch of the feedback transducer is shown in Figure 10. The assembled unit and the spool are pictured in Figures 5 and 11.

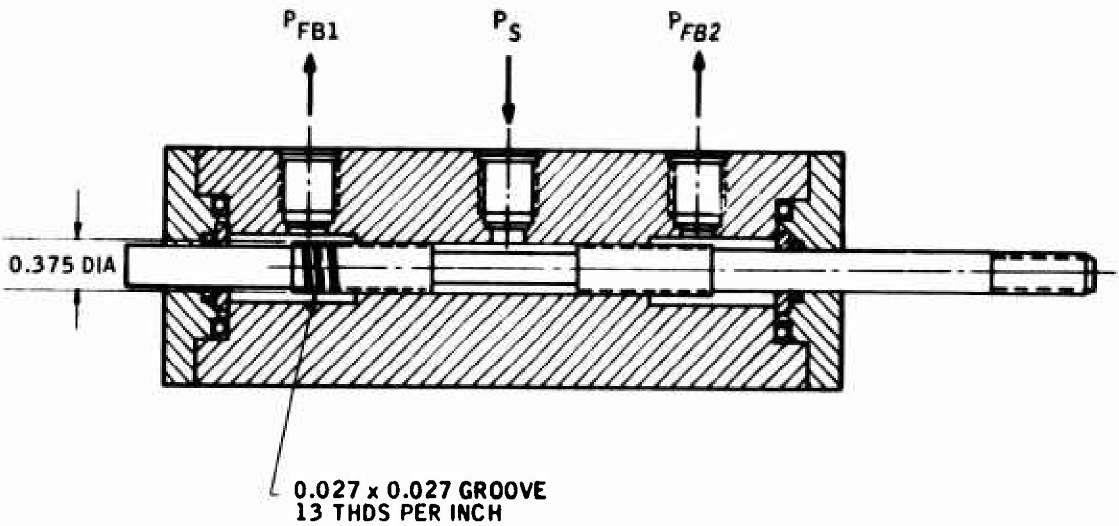

Figure 10. Feedback Transducer.

Servovalve and Actuator

The servovalve and actuator are units used previously on the Three-Axis Fluidic Stability Augmentation System Flight Test Program (USAAMRDL Technical Report 71-34). The first-stage flapper nozzle section is removed from the servovalve, *with only* the spool valve section utilized and driven by the power amplifier output. Two extra sets of spool valve centering springs with lower spring rates (140 lb/in. and 230 lb/in.) were fabricated for increasing the forward loop gain. Figure 5 shows the servovalve, actuator and feedback transducer as a subassembly. An exploded view of the spool valve and centering springs is shown in Figure 12.

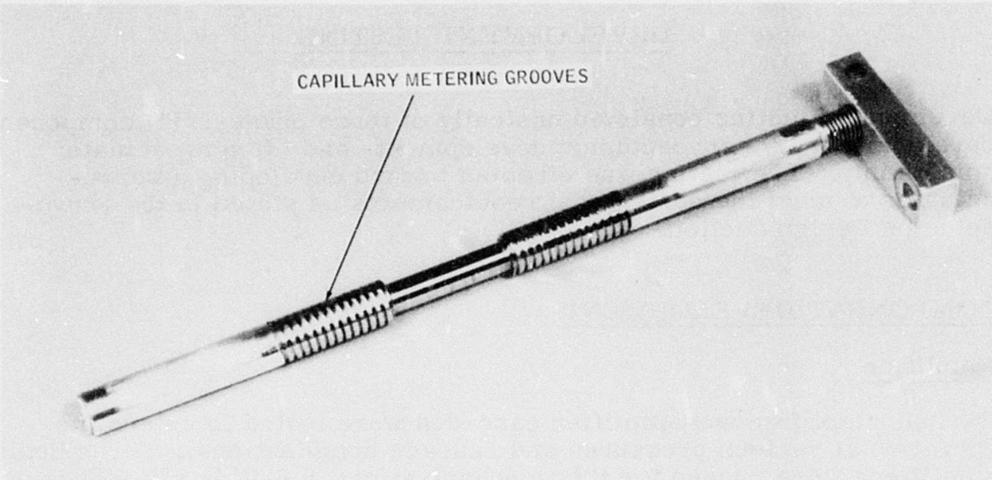

Figure 11. Feedback Transducer Spool.

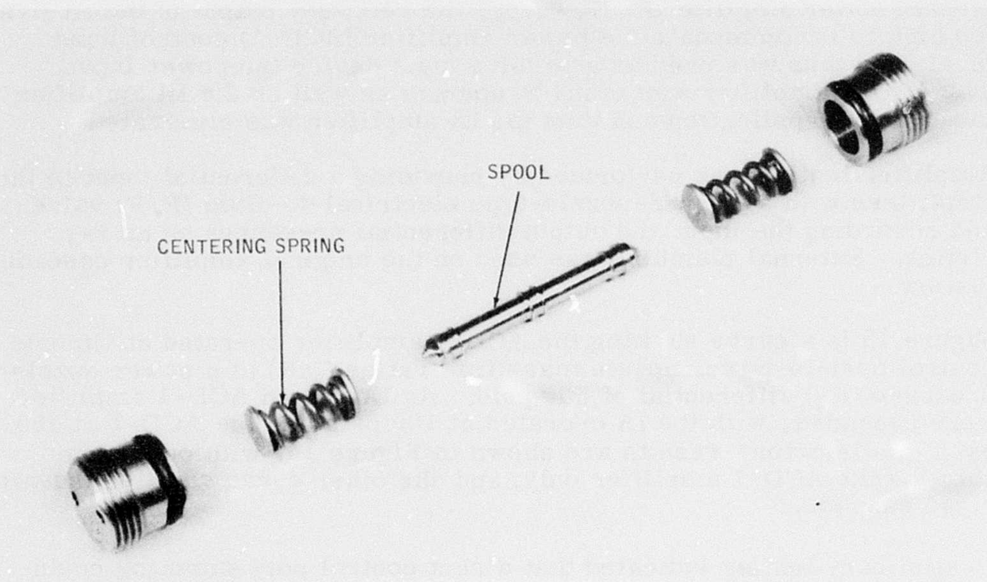

Figure 12. Servovalve Spool and Centering Spring.

SECTION III

DEVELOPMENT TESTING

Development testing consisted basically of three phases: (1) component development, (2) servoactuator development, and (3) servoactuator noise study. The testing was directed toward developing a servoactuator to meet the design goal requirements as stated in the Servoactuator Design section.

COMPONENT DEVELOPMENT

Amplifier

Various amplifier and amplifier cascades were tested to determine operation at various pressures and cascade combinations. Three basic amplifiers were chosen for this mechanization: a 0.015- x 0.020-inch power nozzle amplifier (IA), to be used mainly as an interface amplifier between the input and the cascade; a 0.020- x 0.020-inch power nozzle amplifier (ACD-1); and a 0.020- x 0.020-inch power nozzle summing amplifier (ACD-2). Final configuration, however, utilized only the ACD-1 and ACD-2. Tests were run using the IA amplifier as a lower pressure (100 psig) interface amplifier between the input and the downstream power amplifiers. However, the recovery output of the IA was too high to accommodate the power amplifier (ACD-1) control input level, and thus was used as a passive input device (no power input). The ACD-1 amplifier was found to operate as well as the IA amplifier in a passive application, and thus the IA amplifier was eliminated.

Amplifier testing was performed by providing a differential input to the amplifiers with a flapper-nozzle-type electrical-to-fluid (E/F) valve and recording the input and output differential pressures on an x-y plotter. External plumbing was used on the original amplifier cascade testing.

Figure 13 is a curve showing the ACD-1 amplifier operated at various control port-to-power nozzle mass flow ratios (MR) at a power nozzle pressure (P_S) differential of 500 psid. An IA and an ACD-1 amplifier were cascaded, with the IA operated at 100 psig and the ACD-1 at 500 psig. Satisfactory results are shown in Figure 14, with one curve showing the ACD-1 amplifier only, and the other curve showing the two-stage cascade.

Preliminary testing indicated that direct control port summing could be utilized; however, the adverse effect of gain reduction and input interaction increased the difficulty of providing sufficient input range to obtain the required output stroke without an excessive forward loop gain. Therefore, the summing amplifier (ACD-2) was incorporated to

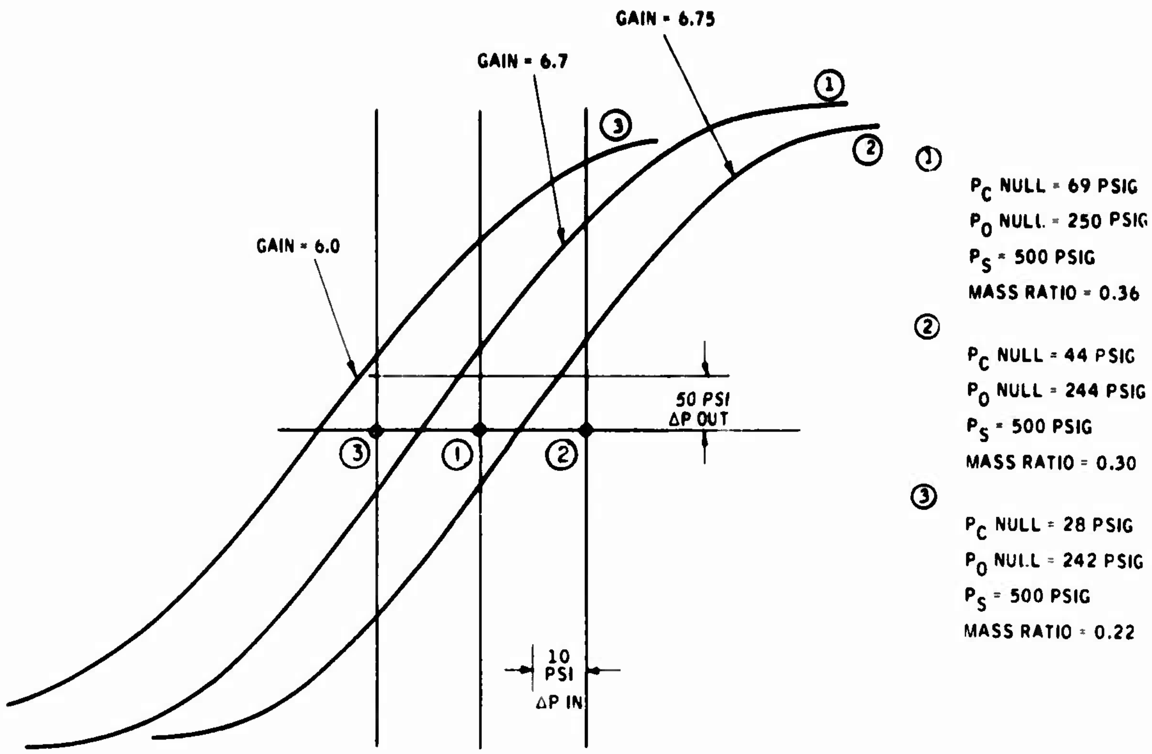

Figure 13. 0.020 x 0.020 Power Amplifier (PA), Dead Ended Output - MIL-H-5606 Hyd. Fluid at 100°F.

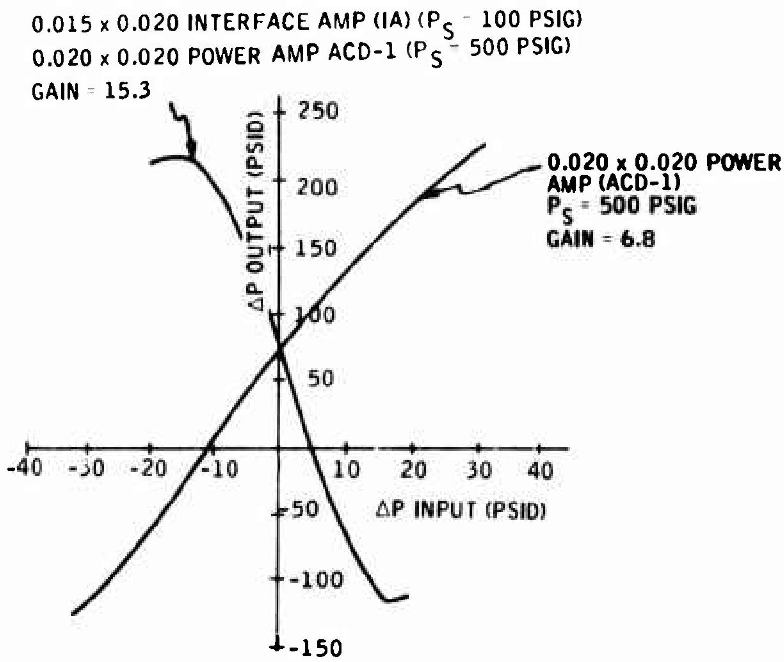

Figure 14. IA and ACD-1 Cascade, Dead Ended Output - MIL-H-5606 Hyd. Fluid at 100°F.

facilitate the summing function. Shown in Figure 15 are curves of the ACD-1 and ACD-2 cascade at 500 psig supply pressure with input command in either set of summing control ports.

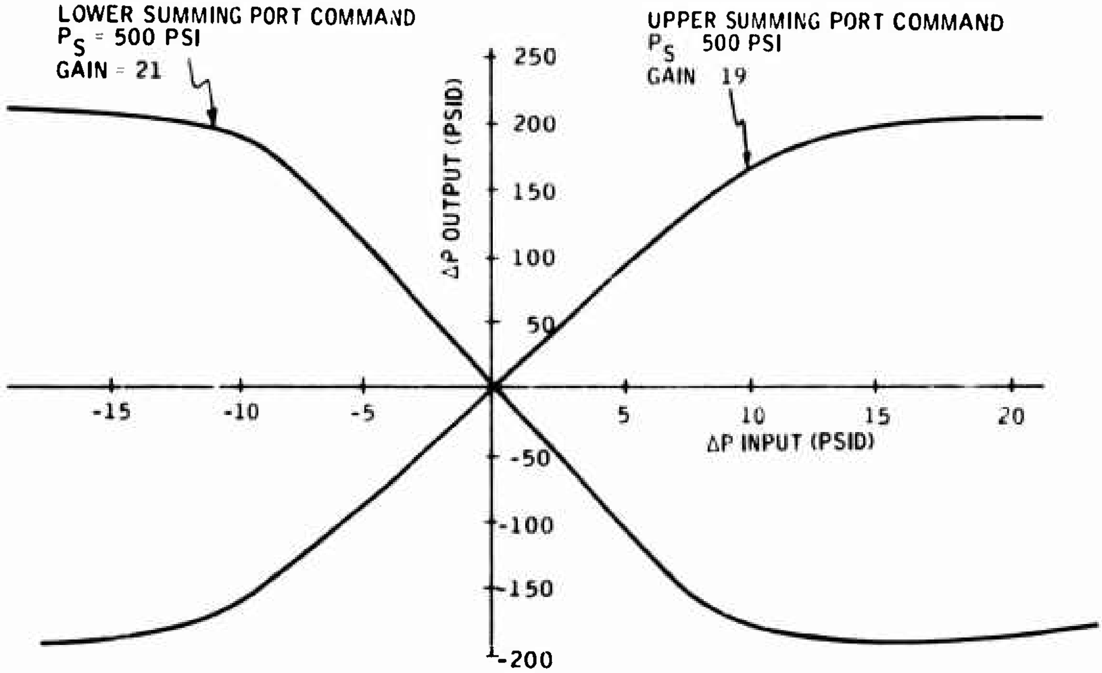

Figure 15. SA and PA Cascade, Dead Ended Output - MIL-H-5606 Hyd. Fluid at 100°F; ΣQ_{FB} on Summing Port = 0.54 in.^{3}/sec.

To determine amplifier gain effects with supply pressure, two ACD-1 amplifiers were cascaded together and operated at various supply pressures. Figure 16 shows the results with supply pressures from 500 to 200 psig.

A four-stage cascade consisting of IA, ACD-1, ACD-2, and ACD-1 amplifiers was tested as a typical cascade. The IA was used as a passive element to reduce control pressure level into the second stage. Figure 17 shows the results, with curve 1 the first two stages and curve 2 the total cascade.

From the amplifier test data obtained, a preliminary amplifier circuit and power supply configuration for interfacing to the servoactuator was selected and is shown in Figure 18. The last ACD-1 amplifier was added to provide a higher forward-loop gain and thus reduce the feedback gain to facilitate adequate input range for full-stroke operation.

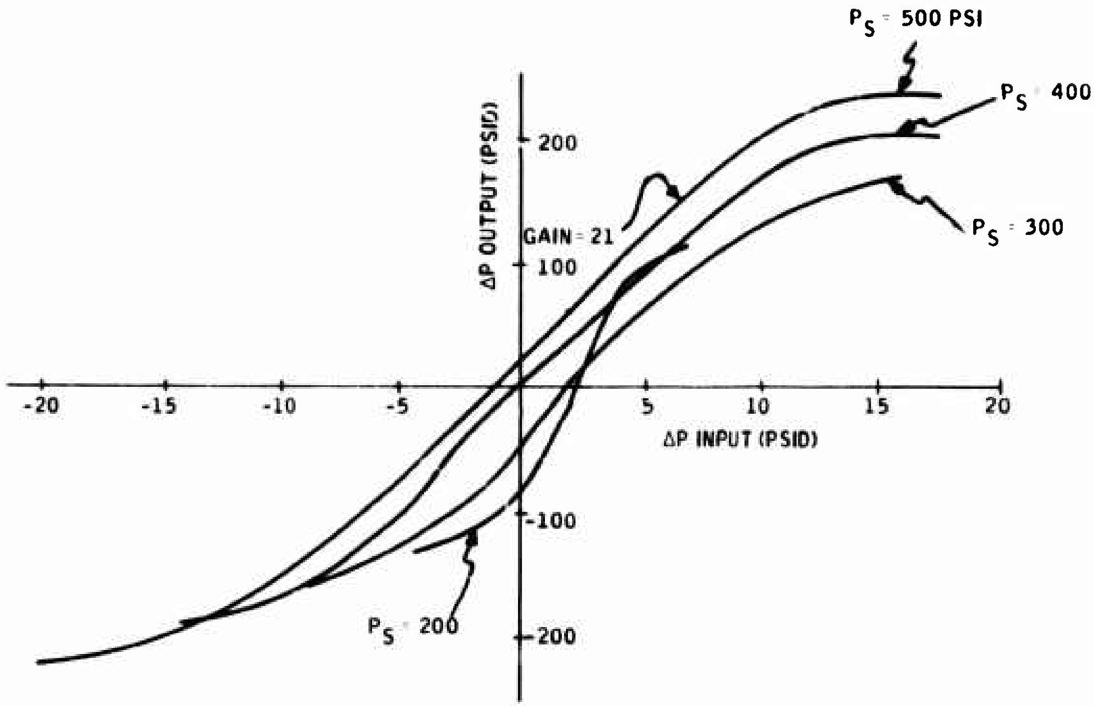

Figure 16. Two PA Cascade, Dead Ended Output on Second Stage - MIL-H-5606 Hyd. Fluid at 100°F.

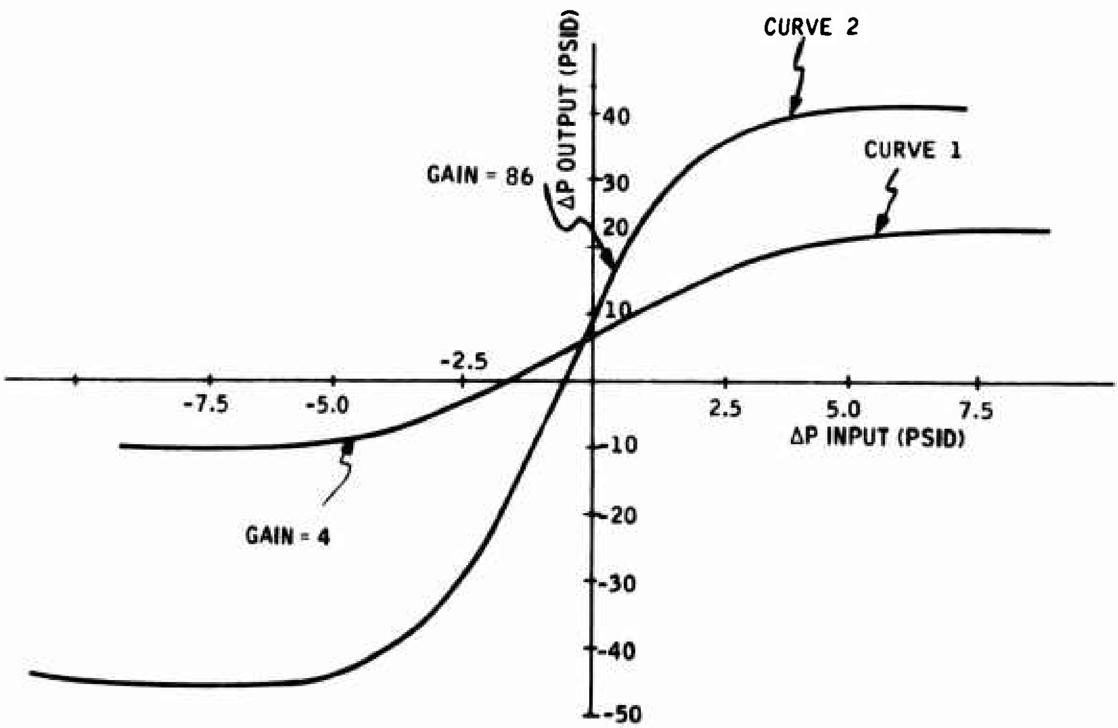

Figure 17. Four-Stage Cascade, Last Stage Dead Ended - MIL-H-5606 Hyd. Fluid at 100°F.

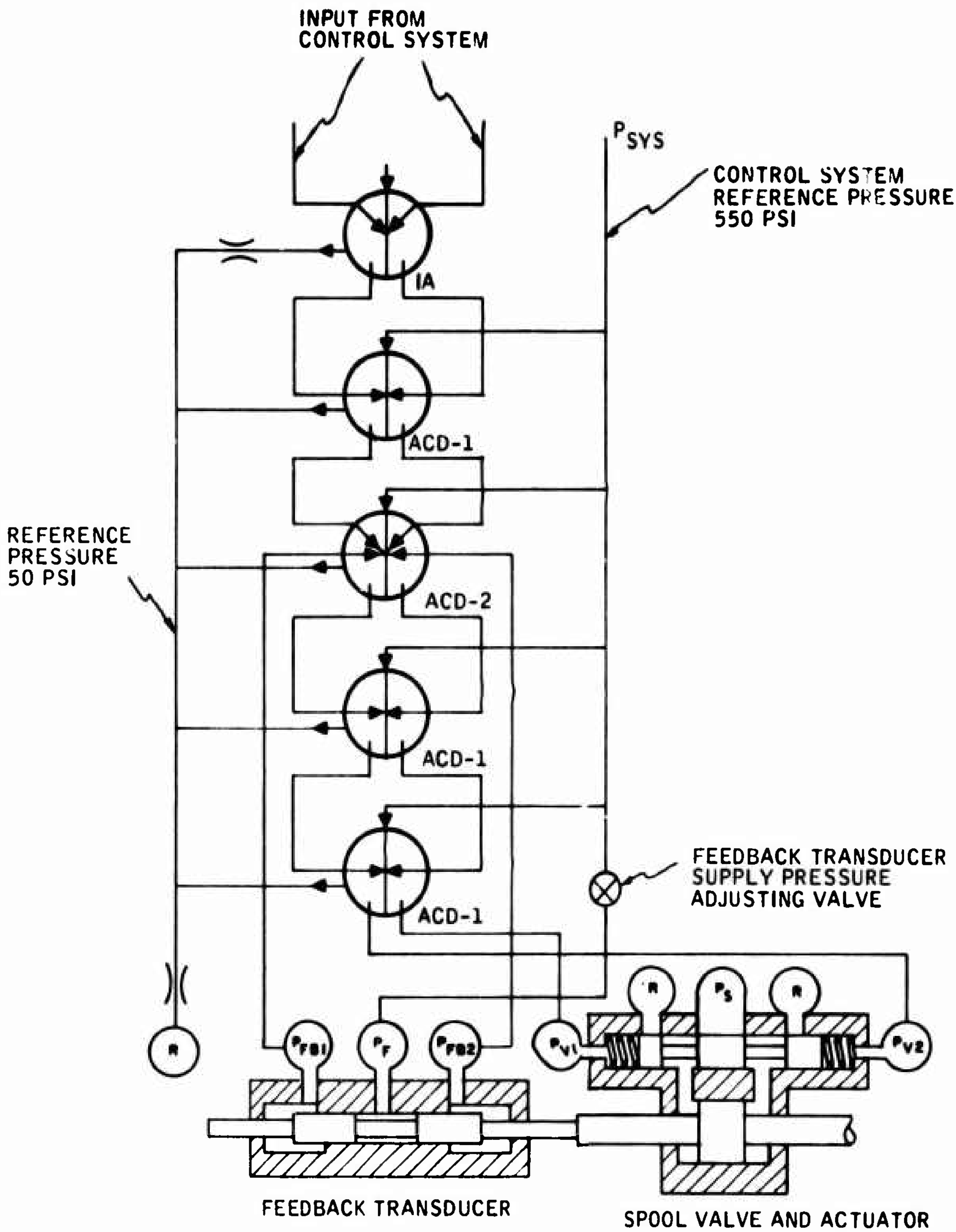

Figure 18. Preliminary Amplifier Circuit and Power Supply Schematic.

All amplifiers are powered from the control system reference pressure of 550 psig except the first stage, which will operate as a passive-type element with zero supply pressure. Thus, the servo actuator system requires no added flow above that of the control system, except for the spool valve flow.

Feedback Transducer

The feedback transducer operation was determined by checking the differential pressure output versus spool position. Figures 19 and 20 show the gain at 100 and 200 psig supply pressure, respectively, with the output loaded into the summing amplifier (ACD-2) operating at 500 psig supply pressure. The required stroke is ±0.375 inch.

SERVOACTUATOR DEVELOPMENT

Interface testing was done using an interface manifold between the amplifiers and the servoactuator. External lines were run between the feedback transducer output and the summing amplifier. Figure 21 shows the test setup used in the servoactuator system testing.

Initial Configuration Tests

Initial performance of the servoactuator system is shown in the output gain curve of Figure 22 and the dynamic response in Figure 23. The output gain is shown to be 0.046 in./psi and output stroke 0.58 in. as compared to the required gain of 0.093 in./psi and stroke of 0.75 in., which indicates that the forward loop gain should be increased and feedback gain reduced to provide added gain and range.

The initial response results, Figure 23, indicated the response to be considerably lower than the loop characteristics indicated; also, the amplitude ratio of -11 db at the 90-degree phase lag frequency was not characteristic of a second-order system of approximately -3 db for a damping ratio of 0.7. Various gain adjustments were made by changing amplifier gain, spring rates of the spool valve centering springs, and feedback transducer gains, without satisfactory results.

A check was made of the feedback output response with respect to the system input and compared to the system output response as shown in Figure 23 and indicates a feedback lead, which effectively results in a system lag. The feedback output response was also checked versus actuator stroke, for two different feedback gains, and found to have a

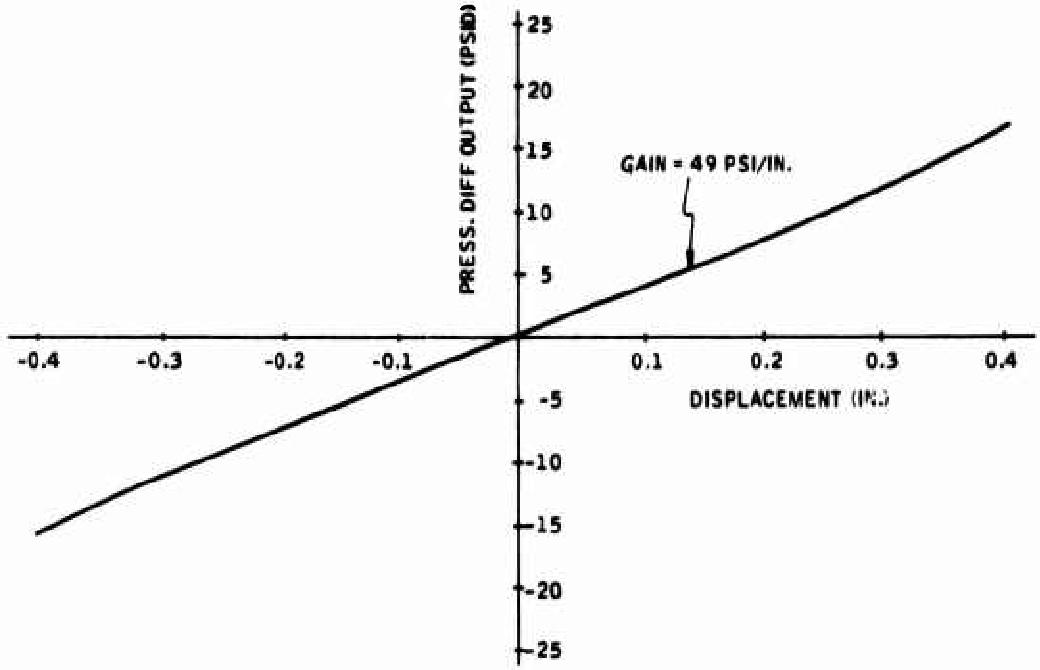

Figure 19. Feedback Transducer Output Gain - Summing Amplifier Load at P_{SYS} = 500 psi, P_F = 100 psi.

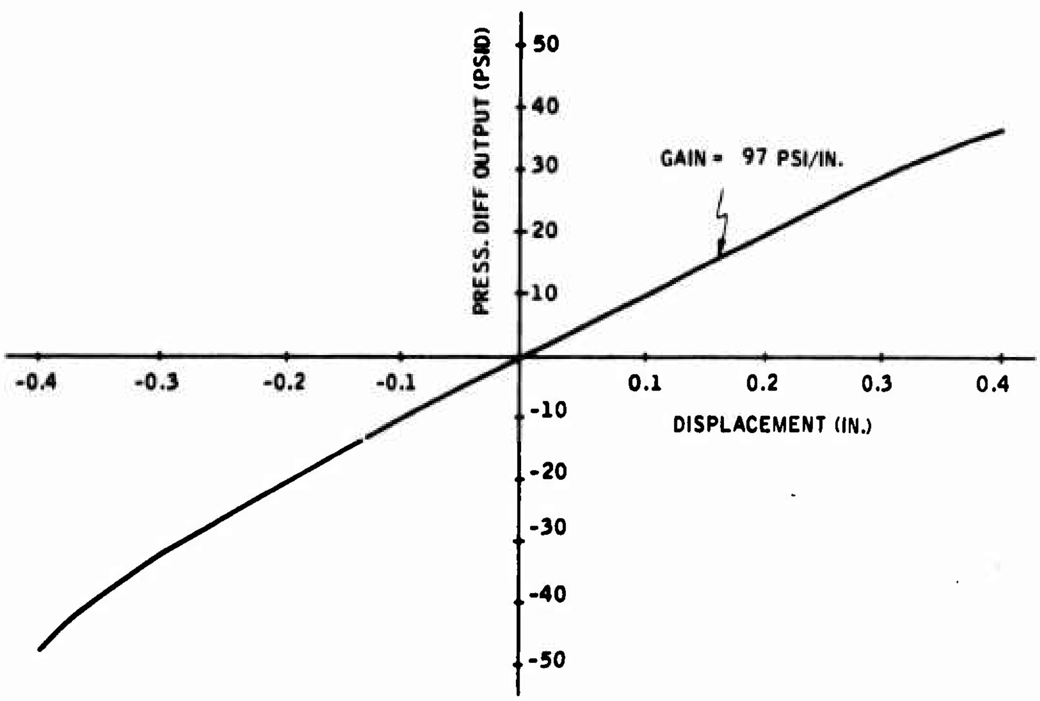

Figure 20. Feedback Transducer Output Gain - Summing Amplifier Load at P_{SYS} = 500 psi, P_F = 200 psi.

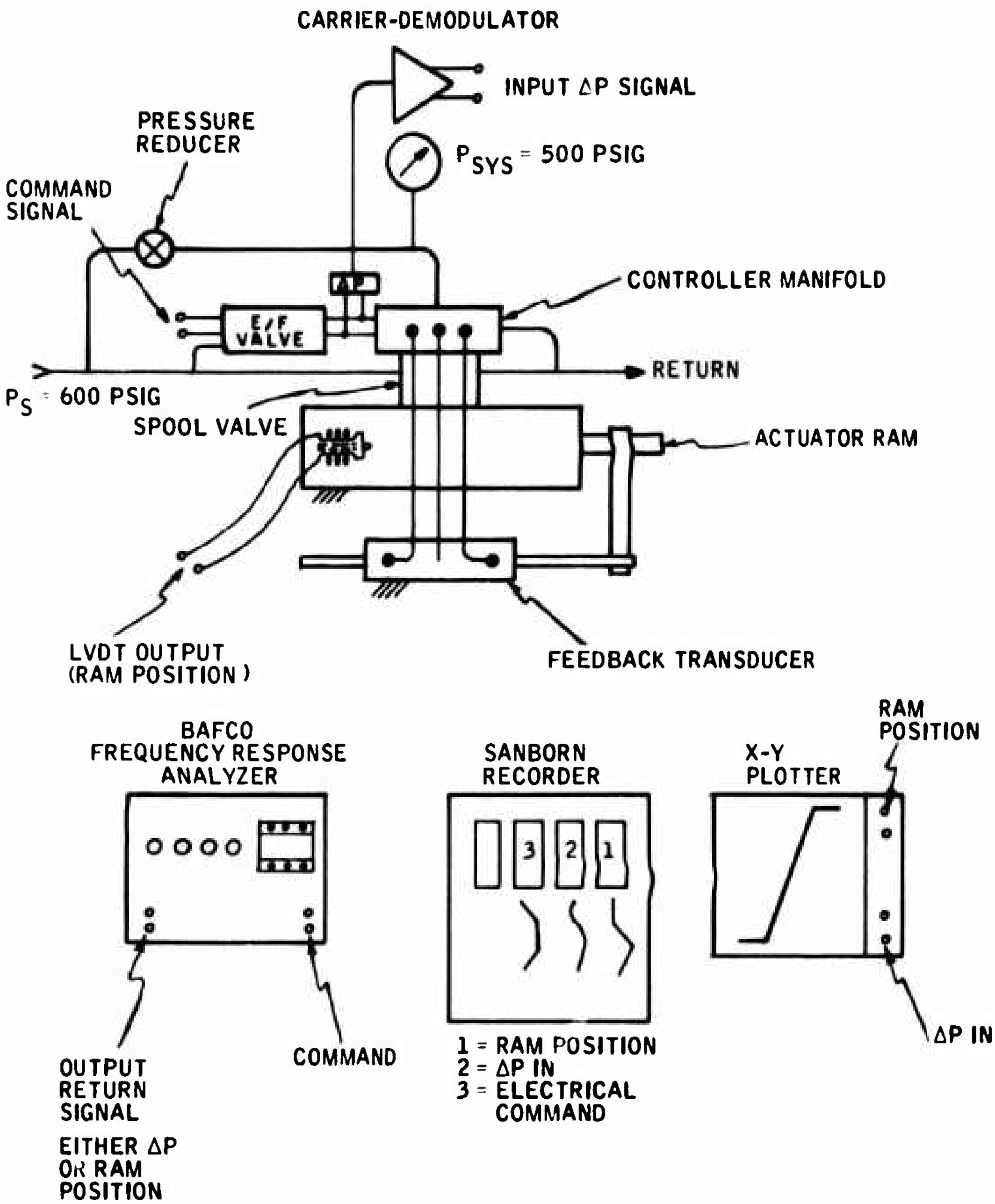

Figure 21. System Test Setup.

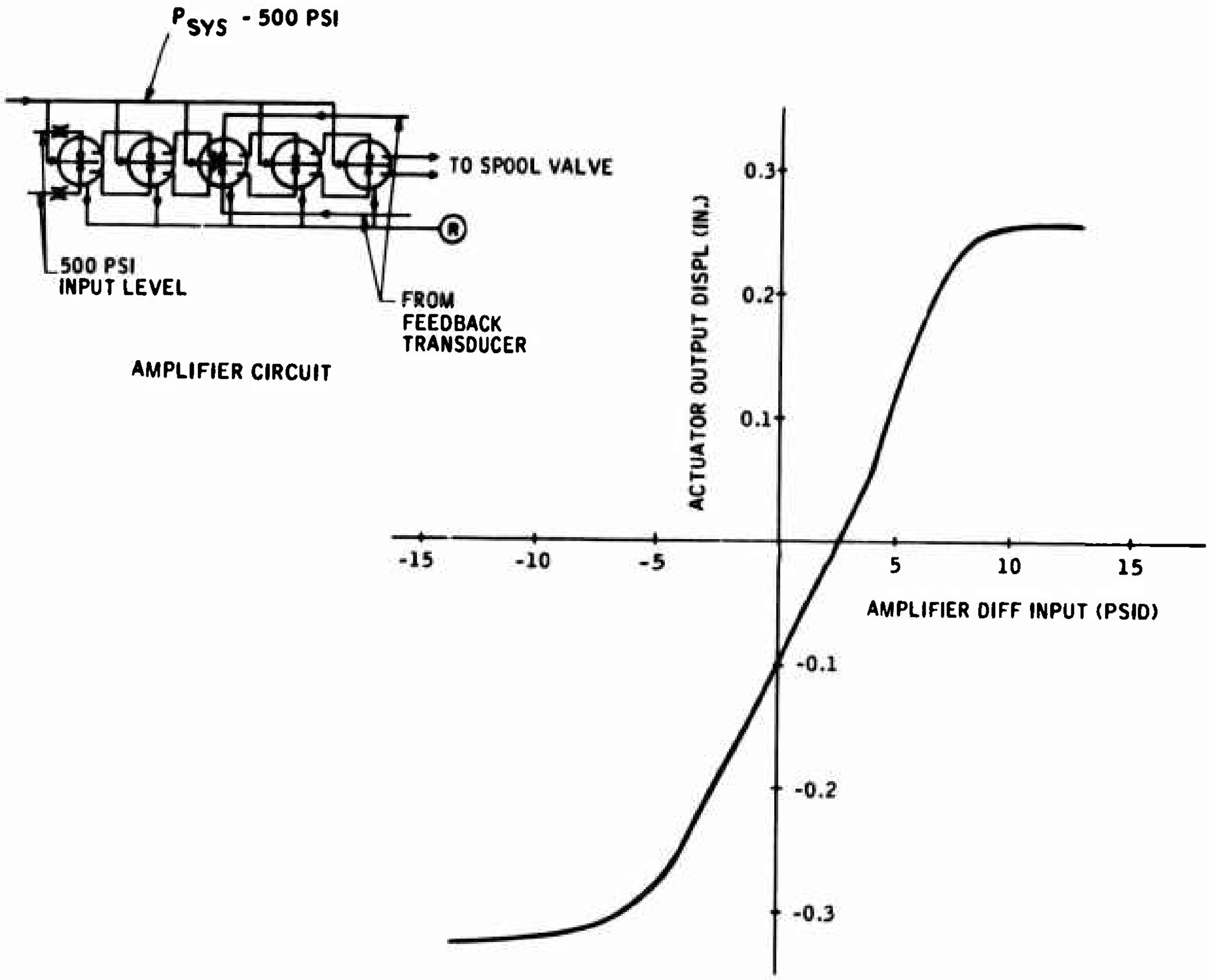

Figure 22. Actuator Output Versus Amplifier Cascade Input Pressure.

lead as shown in Figure 24. This lead was found to be caused by a slight decrease in the feedback spool diameter at the seal ends, causing a pumping action due to the change in volume. The decreased spool diameter provided cutter relief for the capillary metering grooves and also served as a stop. Modifying the spool by eliminating the decreased diameter, as shown in Figure 25, corrected the response problem. Two sets of spool valve springs, 140 lb/in. and 230 lb/in., were tried in an effort to increase the valve gain, allow reduced feedback gain and provide increased output range of the system. Figure 26 shows the response before and after the modification using the 140 lb/in. spool valve centering springs.

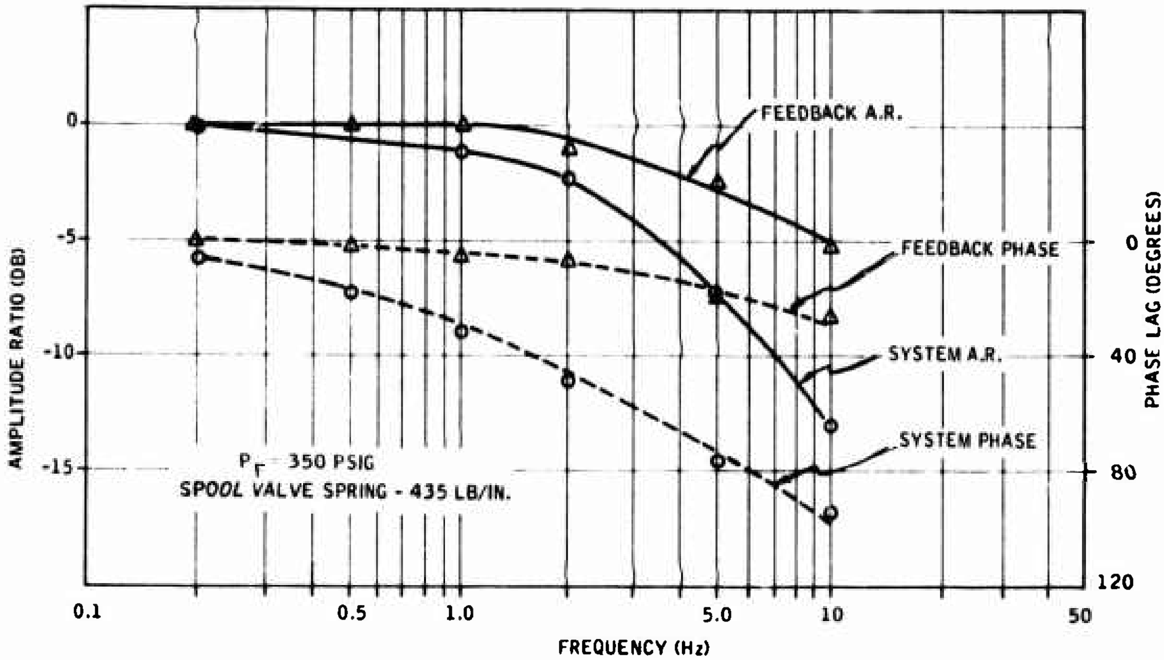

Figure 23. Feedback and System Response With System Input.

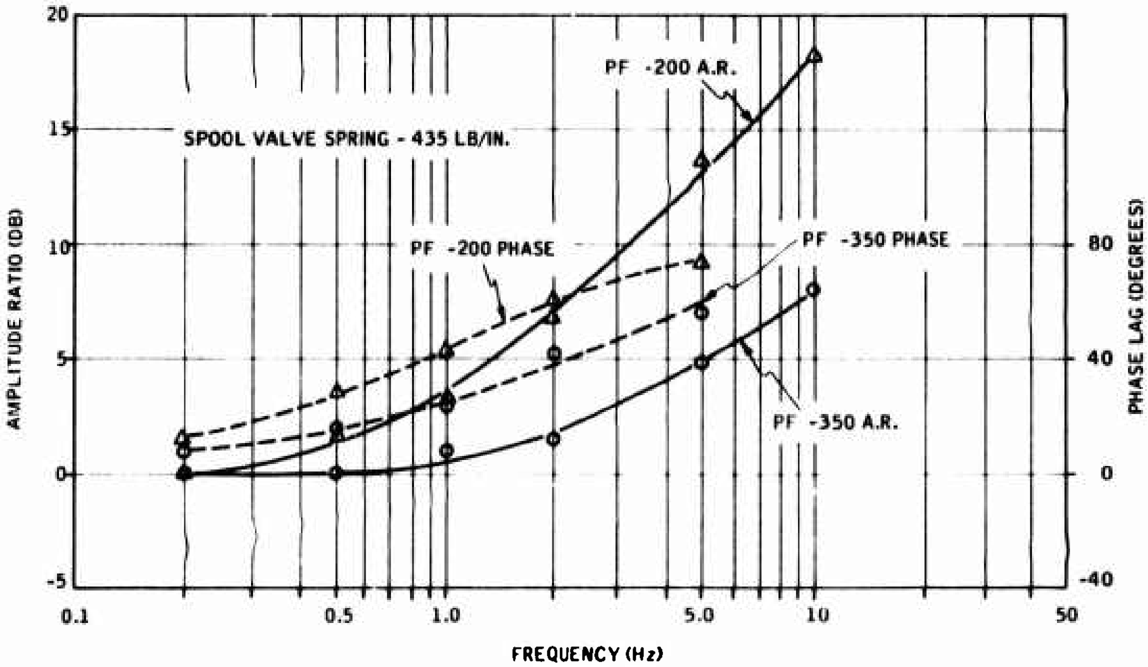

Figure 24. Feedback Response Versus Actuator Stroke for Feedback Transducer Supply Pressure (P_F) of 200 and 350 psig.

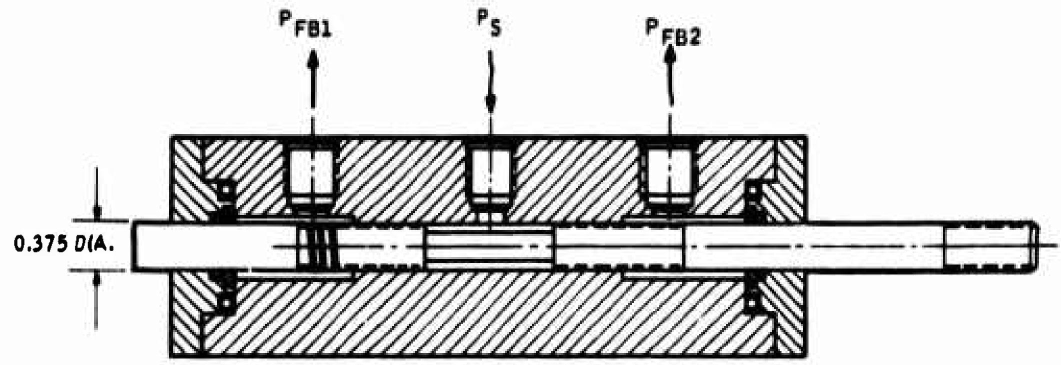

Figure 25. Feedback Transducer With Modified Spool.

To decrease the response to within the required band, as shown in Figure 3, the spool valve centering springs were changed to 230 lb/in. and the forward loop gain was reduced. Also, the preamp gain was increased to increase the range. The resulting response, shown in Figure 27, has a 90-degree phase lag and -6-db amplitude ratio at 14 Hz. The system gain of the actuator output versus amplifier input is 0.063 in./psi, as shown in Figure 27.

Final Configuration Tests

In the final circuit configuration the system output gain was increased by adding an amplifier in the forward loop and resistors in the last-stage amplifier output and in the summing amplifier supply pressure port. The feedback gain was decreased by decreasing the supply pressure to the feedback transducer. The frequency response for this circuit is shown in Figure 28 and has a phase lag of 90 degrees and a -5-db amplitude ratio at 15 Hz. Output gain and noise level are shown in Figure 29 to be 0.075 in./psi and 4 percent of total stroke, respectively.

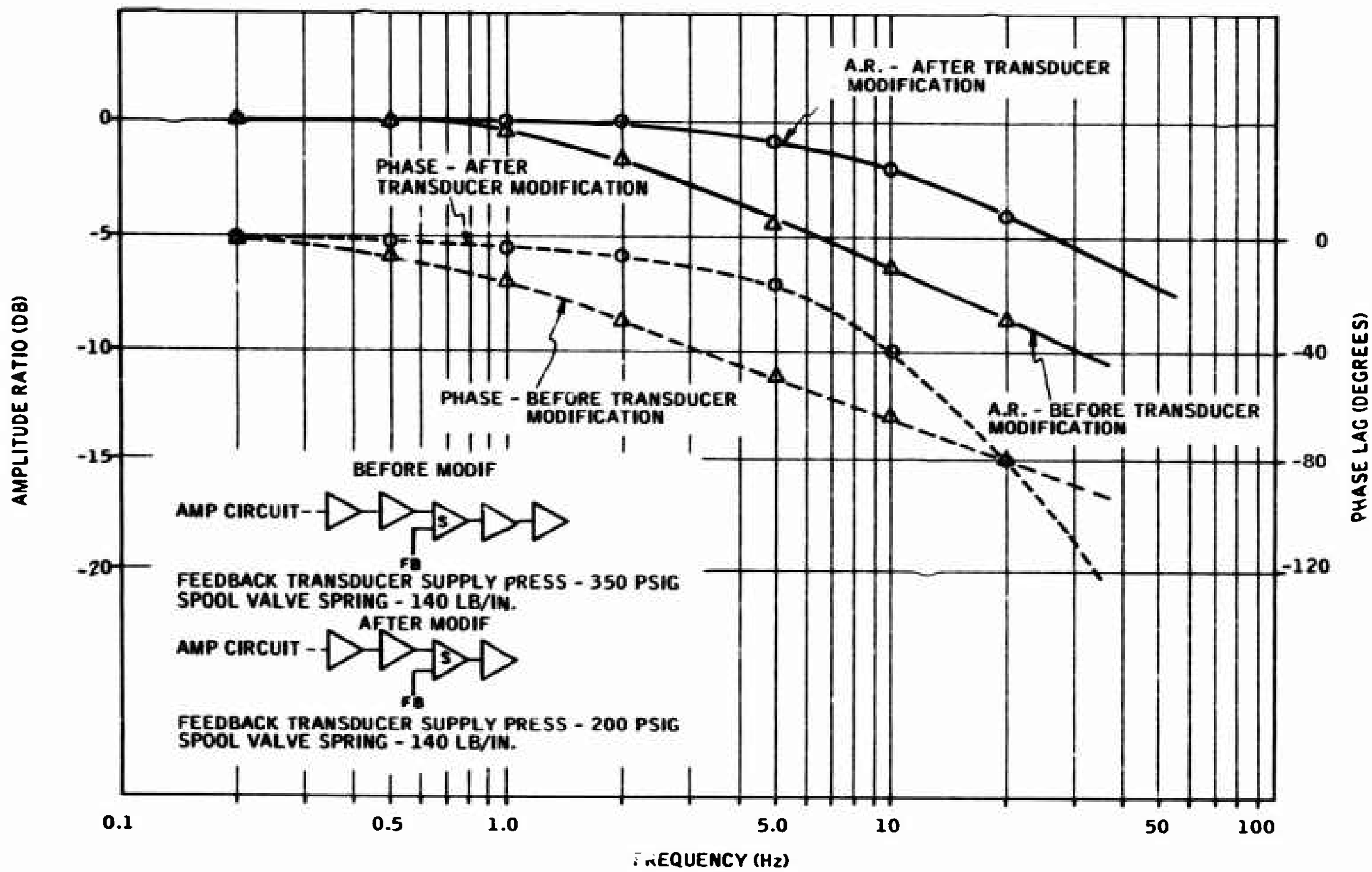

Figure 26. System Response Before and After Feedback Transducer Modification.

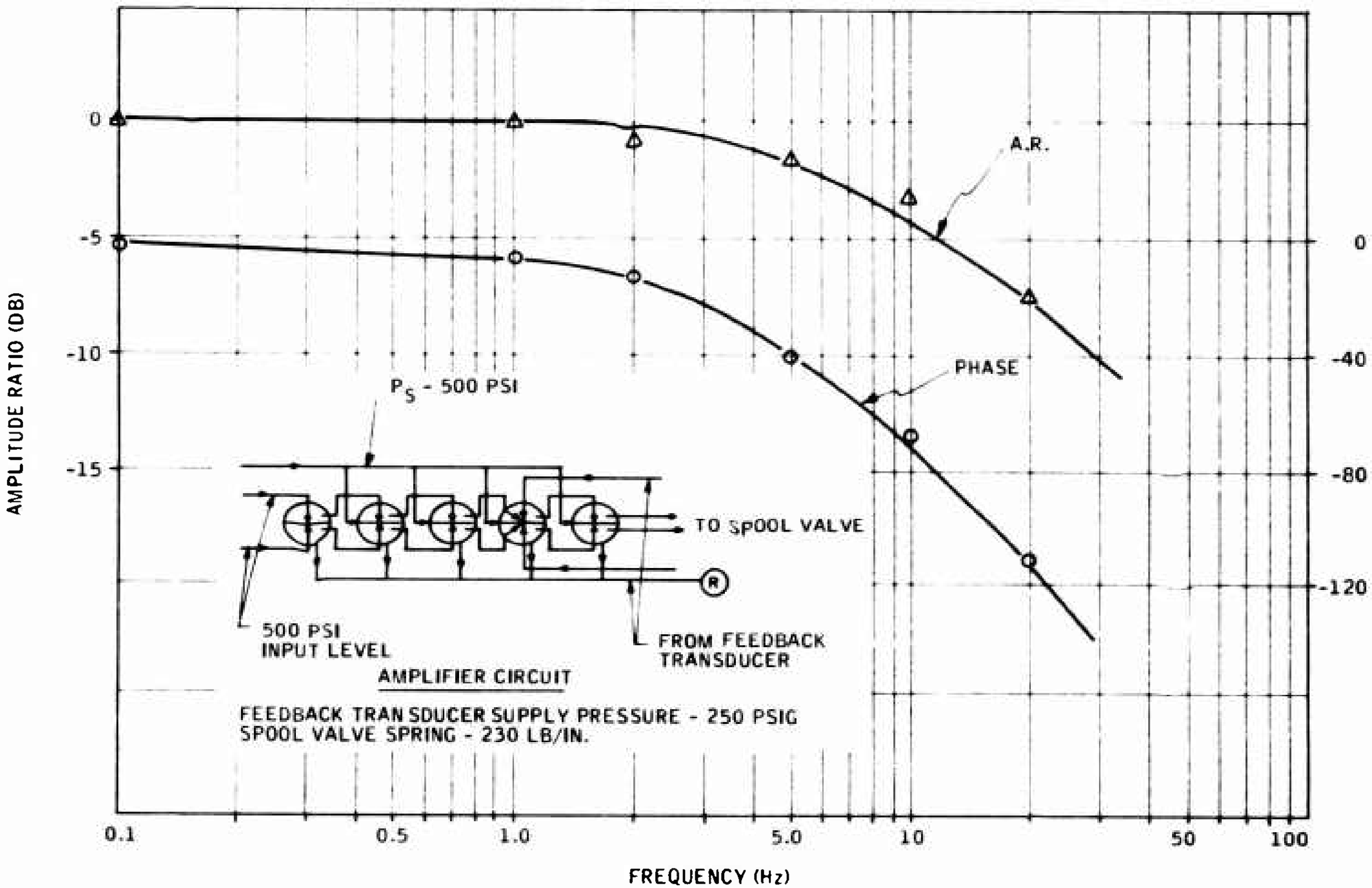

Figure 27. System Response -- Feedback Transducer Supply Pressure, 250 psig.

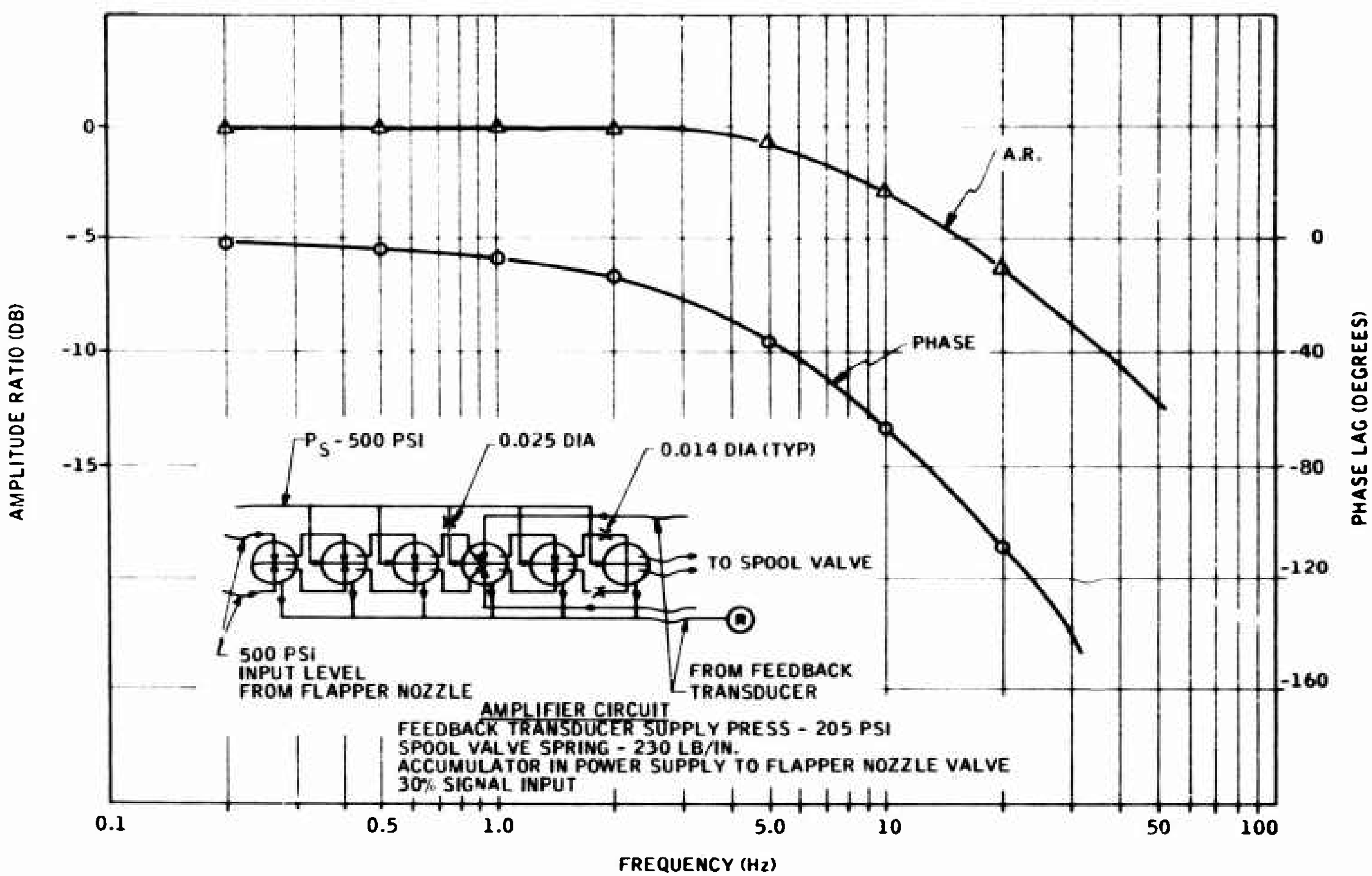

Figure 28. System Response -- Feedback Transducer Supply Pressure, 205 psi.

The frequency and output gain shown are considered satisfactory for this program and indicate the capability of attaining the required response. Further feedback and/or amplifier gain adjustments can be made to more closely match the performance to the design goals, if desired.

Noise was reduced from 8.5 percent to the 4 percent shown in Figure 29 by providing an accumulator in the power supply line to the flapper-nozzle signal generator to reduce the command signal noise. Further noise reduction was attempted by varying feedback gain and modifying the manifold porting, without significant improvement. More comprehensive noise study is covered in the following Servoactuator Noise Study of this section.

Temperature Tests

Temperature effect on response was checked with oil temperature controlled from 70°F to 180°F. Response, gain, and output noise were relatively constant from 110°F to 180°F as shown by the response curves of Figures 28 and 30 and the output-noise curves of Figures 29, 31, and 32. The amplifier circuit used for Figures 28 through 32 was the same except that the 0.014-in.-diameter orifices used in the amplifier output stage of Figure 28 were changed to 0.025 in. diameter for Figures 30, 31, and 32. Temperature effects became more prevalent at the lower temperatures, where at 70°F (the lowest temperature run) the output gain was 0.375 in./psi (2 psi input for full stroke) and the noise was 25 percent. No investigation was made into the low-temperature sensitivity; however, the oil viscosity effect on the feedback transducer gain (decrease) may be a cause for the high output gain and noise.

Servoactuator Noise Study (SNS)

After obtaining satisfactory performance from the servoactuator in terms of response, gain, and linearity, it was determined that a noise reduction investigation would be needed. Noise of ±8 percent of the actuator stroke was observed, which is considered to be in excess of the desirable level.

This investigation was aimed at locating the cause of the noise, and reducing or eliminating it, if possible, within the total system of servoactuator constraints. These constraints included the response and pressure requirements of the servo loop.

The noise study began with an evaluation of the noise caused by each amplifier. Some data was already available, but none with the amplifiers mounted on the experimental manifold. Data was then taken on

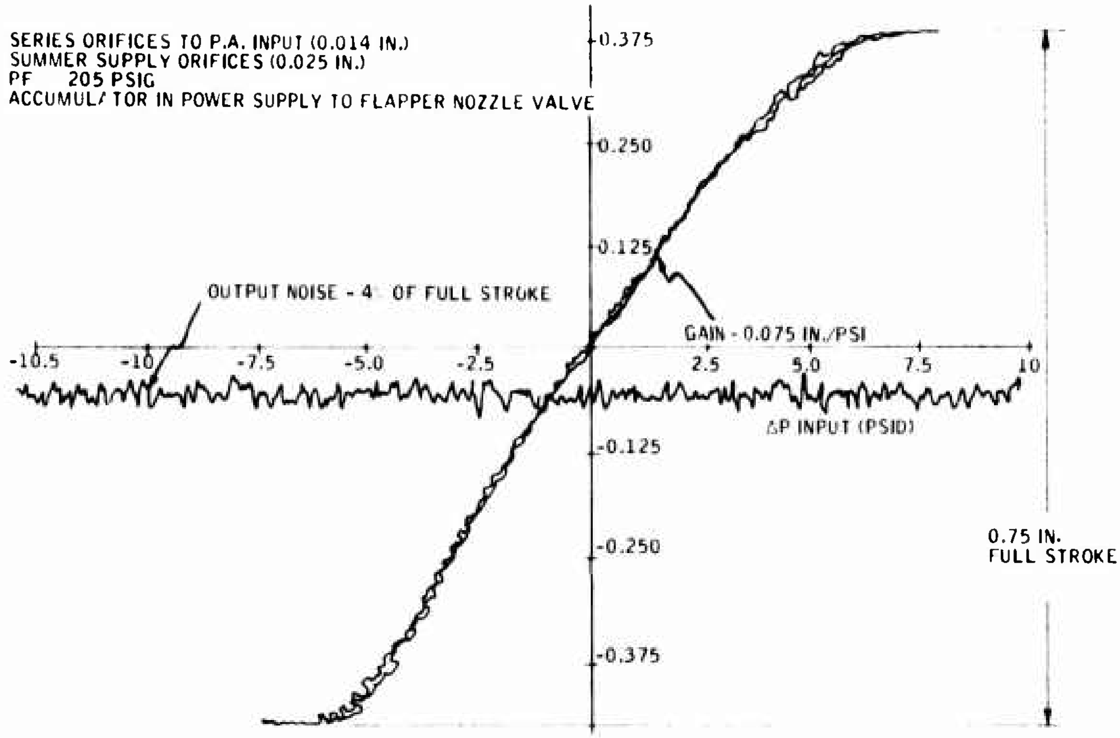

Figure 29. System Output Gain and Noise at 110°F.

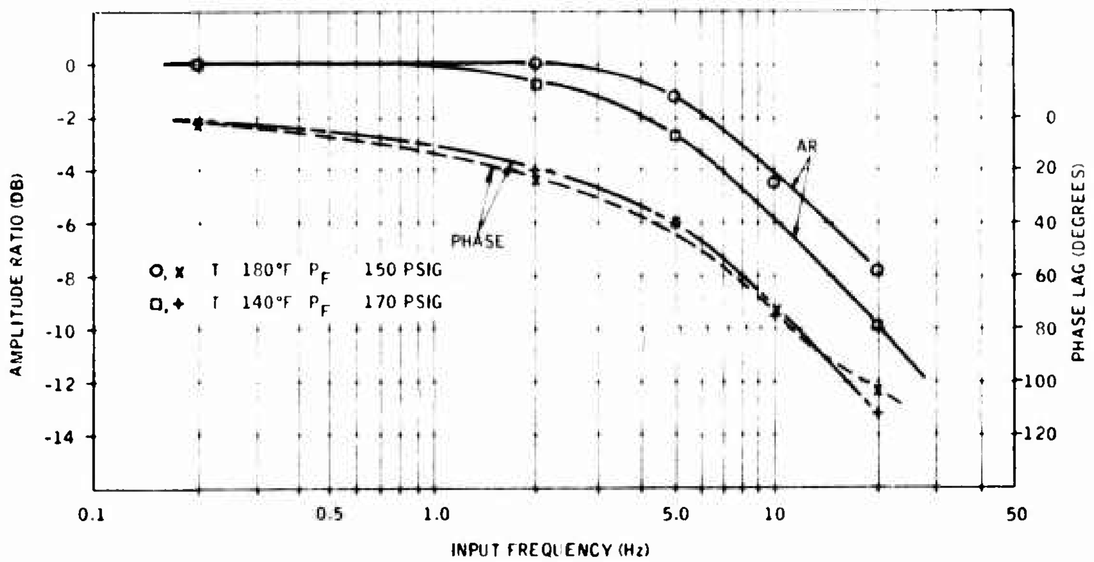

Figure 30. System Response With Feedback Transducer Valve Setting Unchanged Through Temperature Tests.

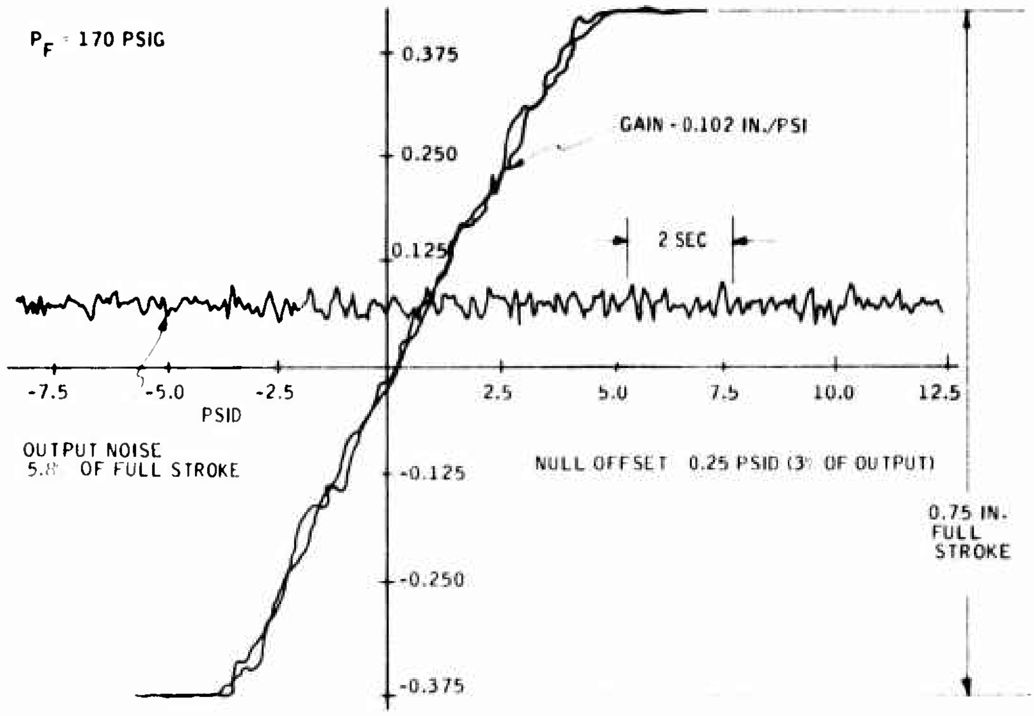

Figure 31. System Output Gain and Noise at 140°F.

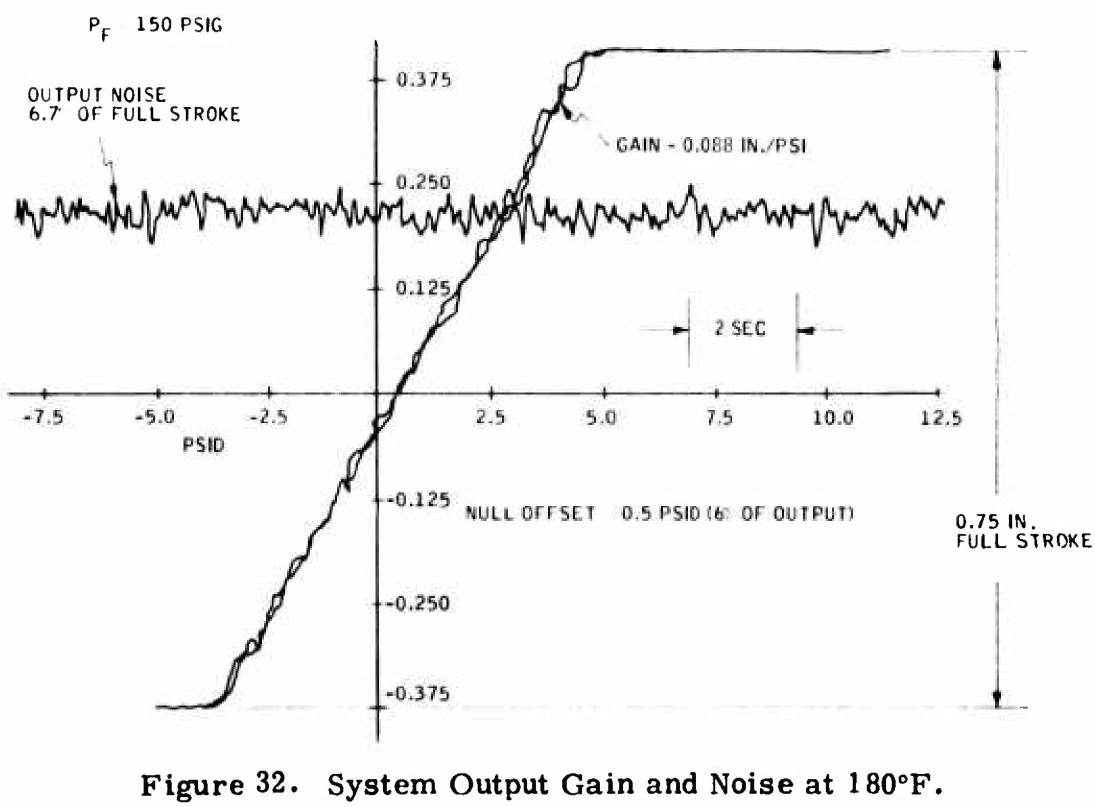

Figure 32. System Output Gain and Noise at 180°F.

individual amplifiers and cascades to determine if the noise could be caused by a single amplifier, possibly operating under adverse conditions.

The first tests were run on the servoactuator preamp. Gain curves are shown in Figure 33 (1, 2 and 3). These amplifier cascades are the same ones operating under the same conditions as used in the final performance tests of the servoactuator (Figure 33-2). Figure 33-1 shows the input-versus-output plot of the input stage only. This model is an ACD-1 (built by Apparatus Control Division of Honeywell) amplifier operating with a blocked power supply. Figure 33-2 shows the gain curve of the first two stages, and Figure 33-3 shows the gain curve of all three stages before the summing amplifier. Stages two and three are operating with a power supply pressure P_S = 500 psid. These curves show that all stages are contributing to the noise but that the first stage is creating more noise than the other two. For this reason it was decided to try a few other amplifiers or conditions for the first stage to reduce the noise contribution of the first stage. In Figure 34, curves 1 and 2 show the gain curves of the input amplifier and then the input amplifier plus two stages, respectively. These curves show the performance of the best alternative approach to a preamp that was found. The noise in this case is less than on the comparable curves (Figure 33-1 and -2); however, the gain is significantly less. To bring the gain up again, another stage would have to be added with a resulting increase in noise and power consumption. The indicated result would be no improvement in noise but an increased power consumption.

The preamp as tested in the final performance tests of the servoactuator is considered to be the best one that could be built with presently available amplifiers.

The amplifiers in the forward loop were evaluated by the same method as above; again it was concluded that with the present amplifiers, noise had been reduced as far as possible without changing the power supply pressure to the amplifiers.

Reduction of the amplifier power supply is a basic and attractive method for reducing amplifier noise. In this case one of the servoactuator constraints restricts the level of the power supply. The differential pressure required to move the control valve spool a particular distance is established by the end area of the spool and the spring rate of the springs at the end of the spool. This valve spool and its springs are shown in Figure 12. A desired maximum ram velocity of 10 in./sec dictated the maximum required control valve position and therefore the differential pressure needed to position the spool. Several sets of control valve springs were tried. The lowest rate springs (140 lb/in. each) were tried again to evaluate actuator noise. The low rate springs

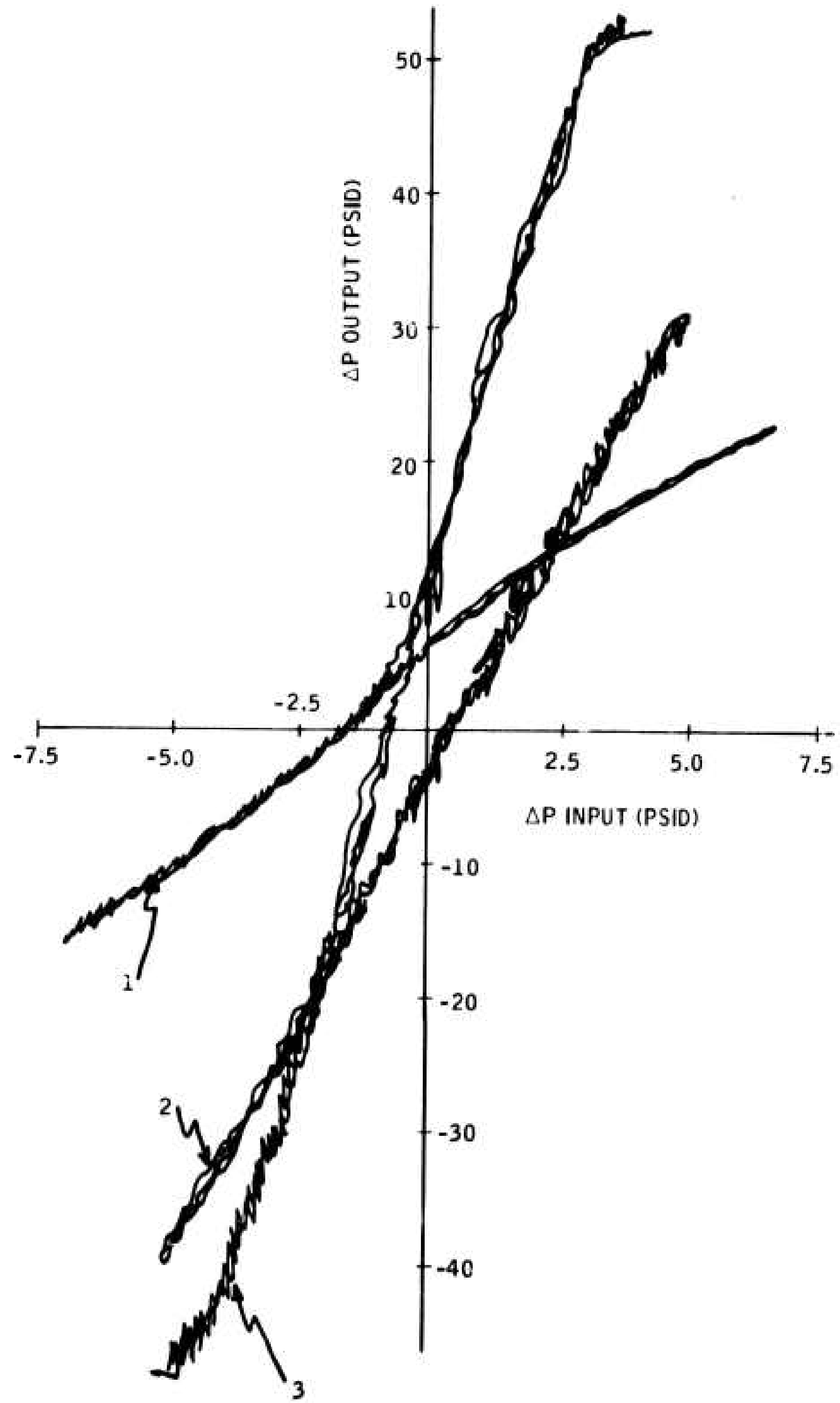

Figure 33. Input Cascade (ACD Passive Amplifier).

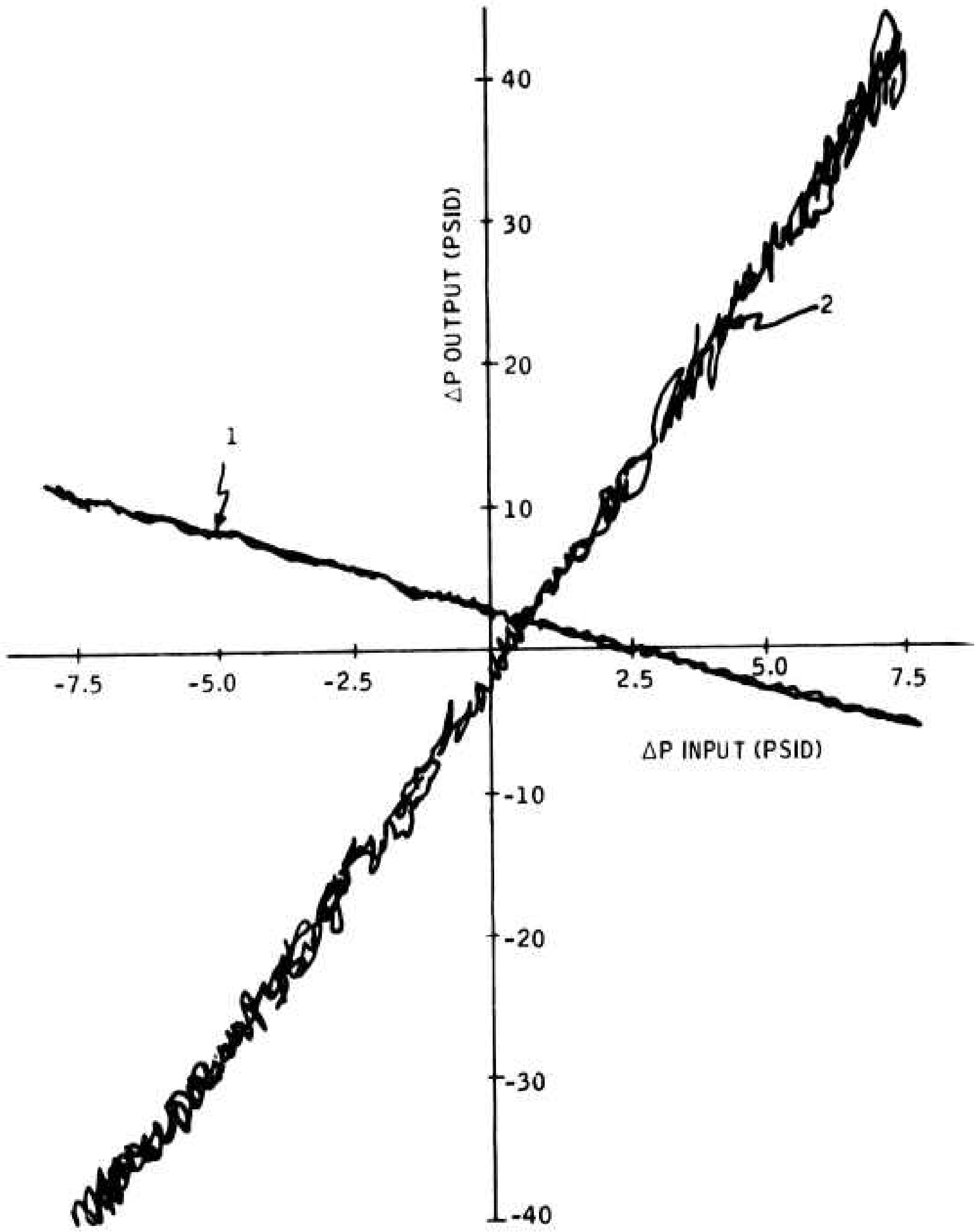

Figure 34. Input Cascade (0.015 x 0.020 Input Amplifier).

made it possible to operate the amplifiers at 300 psid instead of the original 500 psid. As shown in Figure 35 (curves 1, 2, 3, 4, and 5), the noise of the full cascade of amplifiers decreased with power pressure and, in addition, the gain increased. The high-frequency components of the noise were reduced; however, the low-frequency noise was as strong as ever.

An amplifier was being developed on an internal development program which showed promise for a high-pressure application with lower noise level. This amplifier was built in a high-pressure version and used for the final tests on this program. An electroformed amplifier of this type is shown in Figure 36. The design of the amplifier can be perceived by the external shape of the plated surface. This amplifier was only available in a 0.025- x 0.025-in. power nozzle size with a single set of control input ports. A summing amplifier would be needed for this application for a final design. Figures 37 through 39 show the input-versus-output relationship of the three amplifiers used. The power supply used, 200 psid, with these amplifiers gives an output range about the same as the old amplifiers with 300 psid power supply. To obtain ±10 in./sec ram velocity with the 140 lb/in. springs in the control valve, about ±110 psid is needed from the power amplifier. As shown in Figure 39, the amplifier output range approaches ±150 psid; therefore, sufficient range is available. The amplifier noise is obviously improved over previous curves shown of the old amplifier design noise; however, there are still improvements to be made, as observed in the nonlinearity shown in Figure 38. These problems could probably have been solved if more time had been available, since a nonlinearity of this type is usually caused by adverse amplifier loading.

Since the amplifiers showed such significant improvement, it was felt that they should be hooked up with the actuator, at least in a breadboard circuit. The actuator, control valve, feedback potentiometer, and amplifier cascade were connected as shown in the block diagram of Figure 40. A three-stage cascade of new amplifiers was assembled and connected to the actuator with hard plumbing as short as possible. A circuit schematic is shown in Figure 41. The springs in the control valve were 140 lb/in. each, with the amplifier power supply 200 psi above the reference pressure, which was maintained at 50 psi. The gain of the amplifier cascade in this case was 37.5 psi/psi,and the feedback potentiometer gain was 110 psid/in. This provides a loop gain about 1.5 times the loop gain used in the earlier actuator tests.

The servoactuator was tested for its response and noise. The frequency response is plotted in Figure 42. The amplitude ratio is plotted in decibels and holds within 0.5 db to 10 Hz. This response would satisfy the original requirements. The servoactuator noise can be observed in Figure 43, which is a recording trace of the actuator input and output with a sine wave input function. The stroke is ±0.150 in. or 0.300 in.

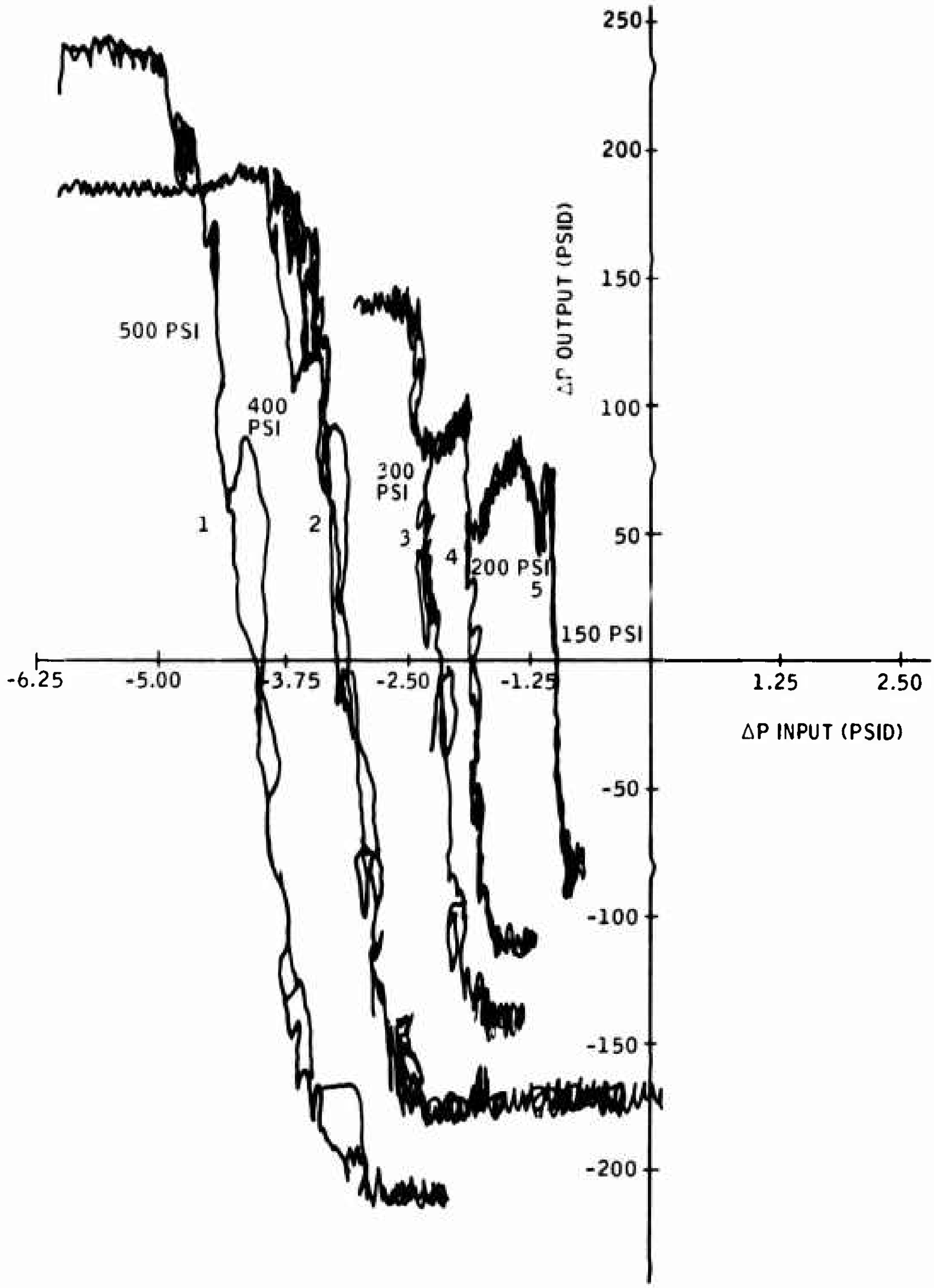

Figure 35. Gain Curve Total Cascade (ACD-1 Type).

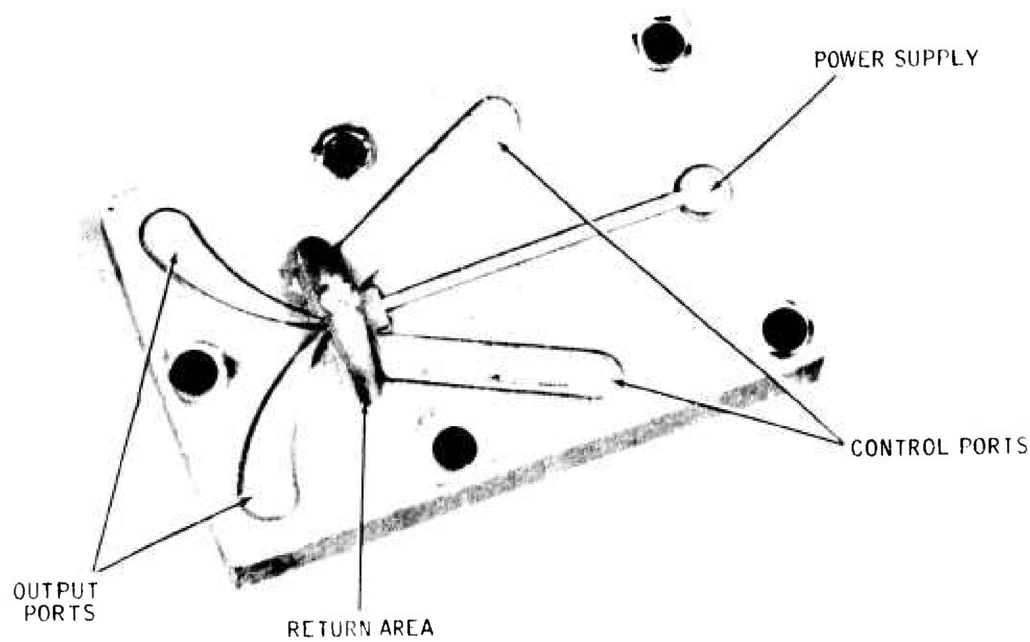

Figure 36. Amplifier, Type FG1005-AA01.

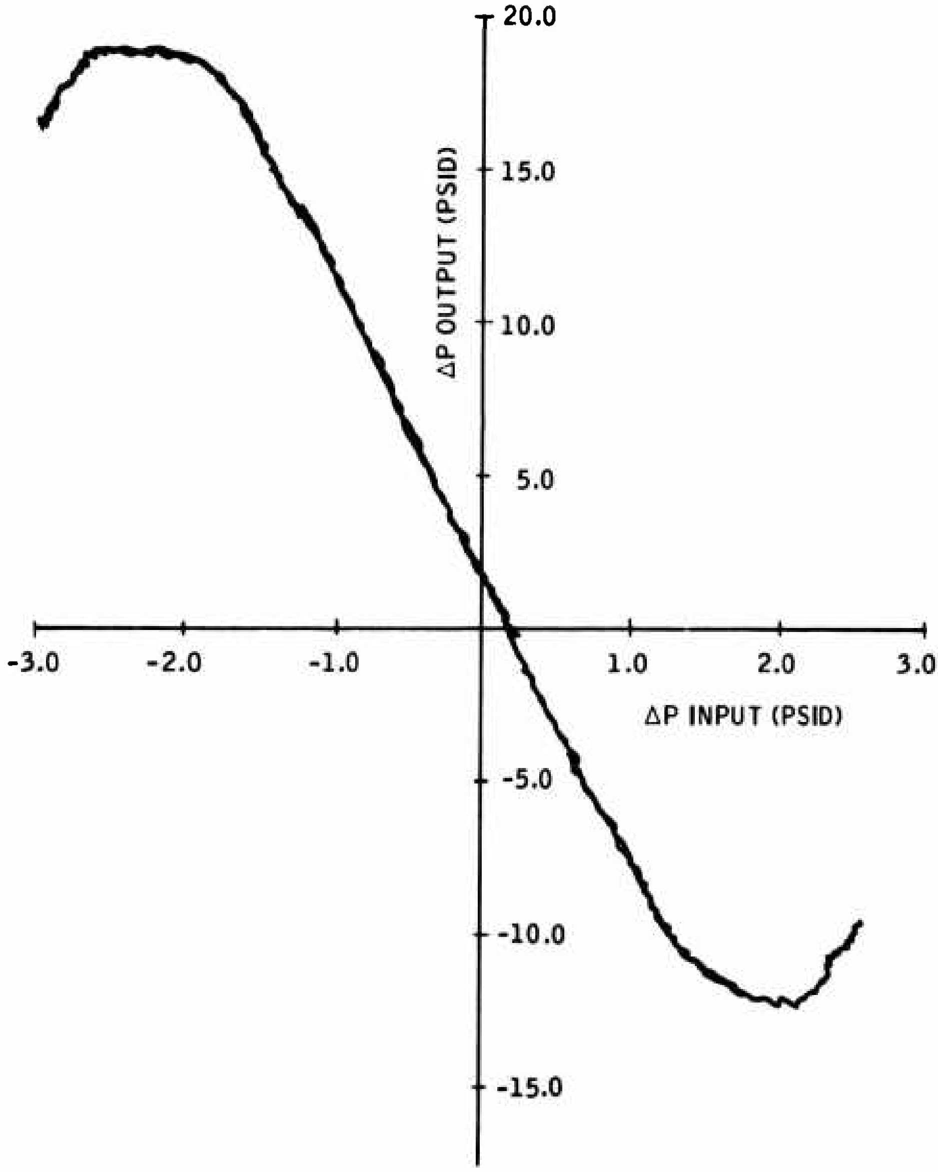

Figure 37. First Cascade Amplifier Gain Curve.

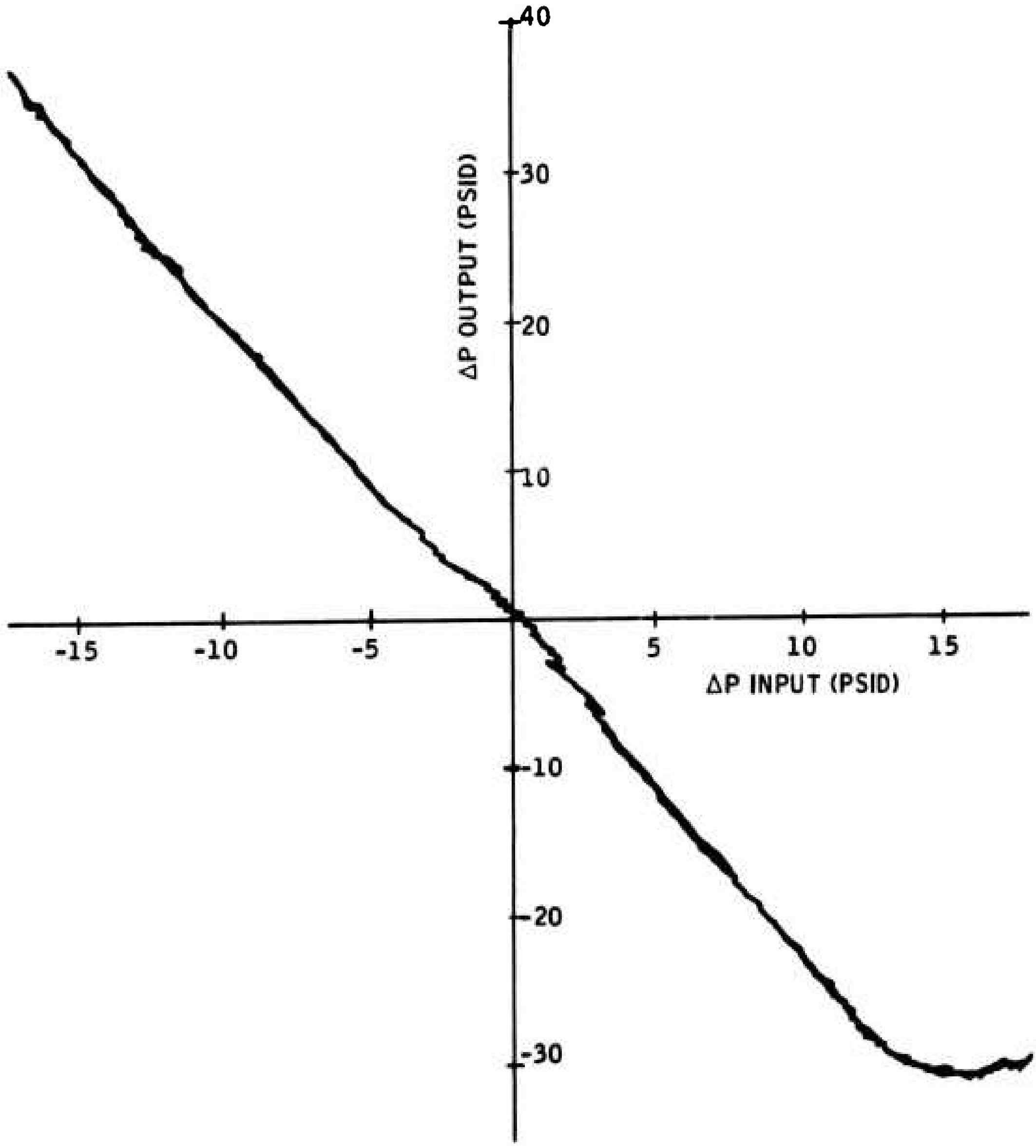

Figure 38. Second Cascade Amplifier Gain Curve.

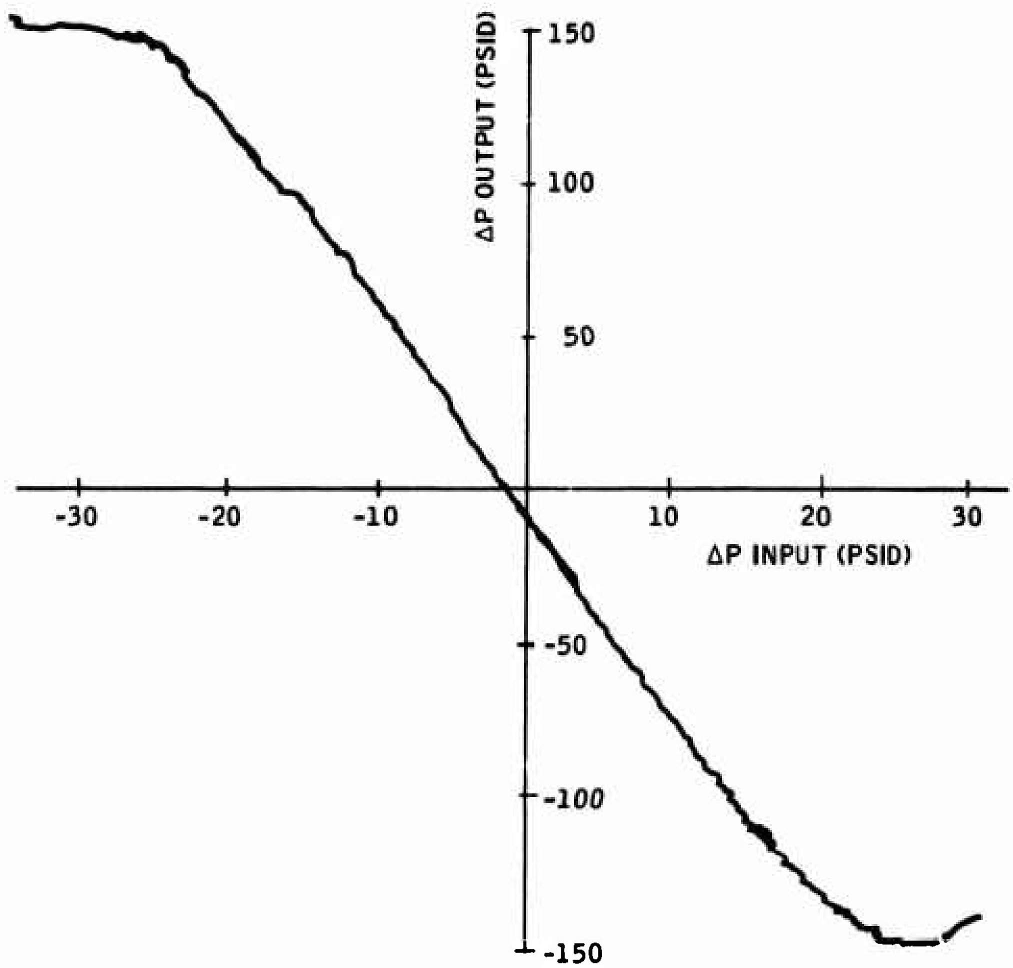

Figure 39. Third Cascade Amplifier Gain Curve.

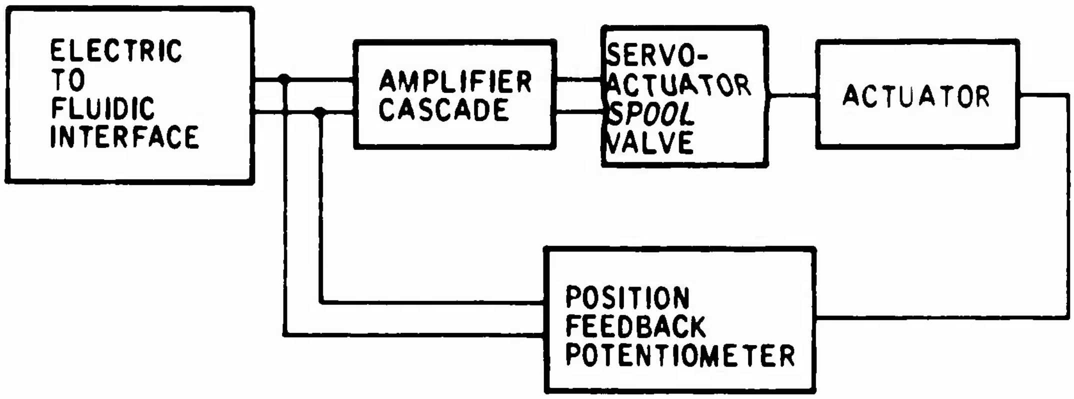

Figure 40. Servoactuator Block Diagram (as run with new high-pressure amplifiers).

peak to peak. The noise is estimated at 2 mm peak-to-peak average while a stroke of 0.300 in. is 42 mm. The percentage of noise is thus calculated as follows:

$$\frac{2}{42} \times 0.300 \text{ in.} = 0.0143 \text{ in.}$$

$$\frac{\text{peak-to-peak noise } 0.0143 \text{ in.}}{\text{full stroke } 0.750 \text{ in.}} \times 100 = 1.9\%$$

The results of these tests obviously must be extrapolated to show the performance of a hydraulic input actuator with the required summing amplifier. The earlier results showed the feasibility of the fluid amplifier approach to a servoactuator; this study has shown that noise can also be reduced significantly with improved amplifier design.

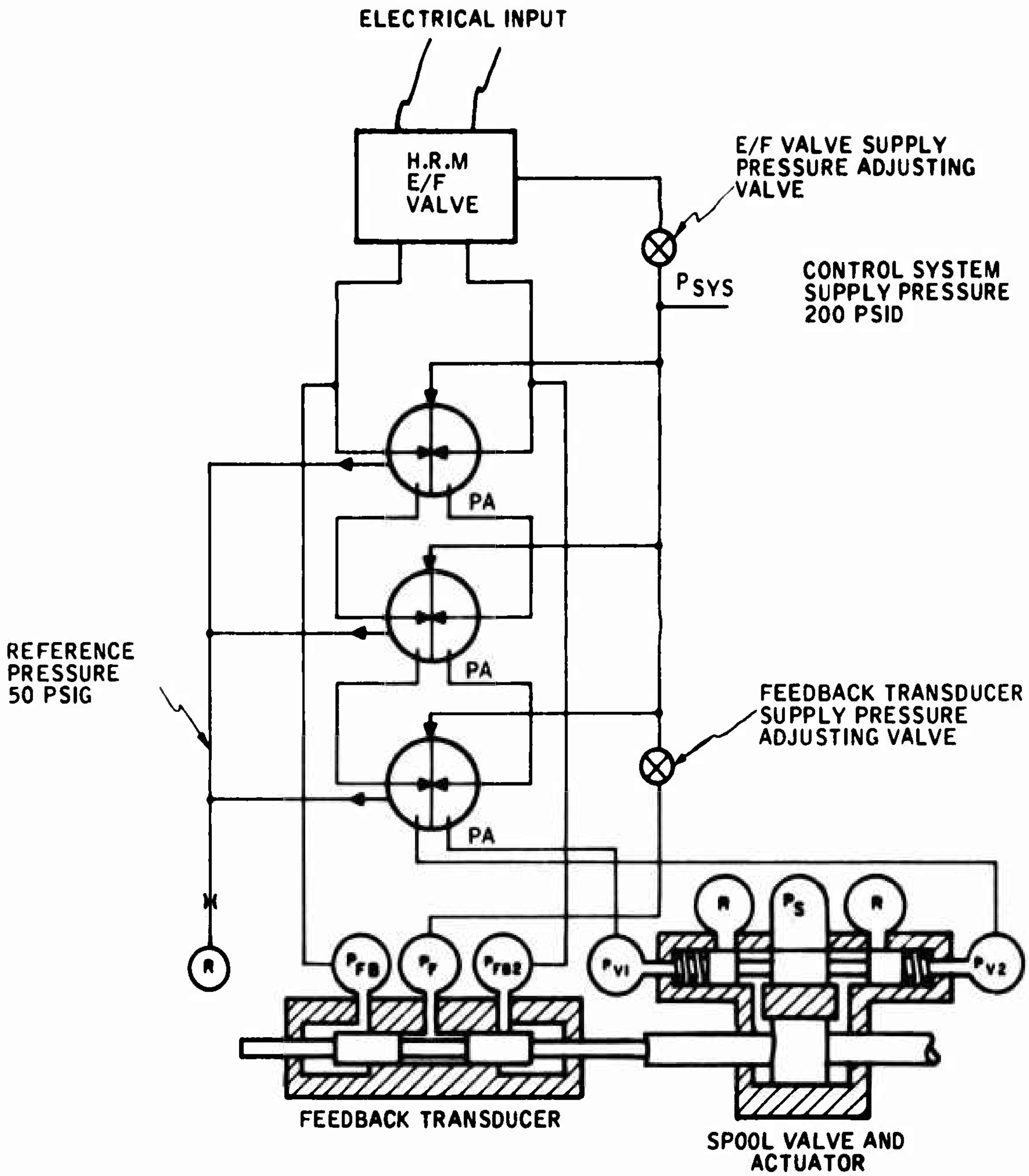

Figure 41. Circuit Schematic.

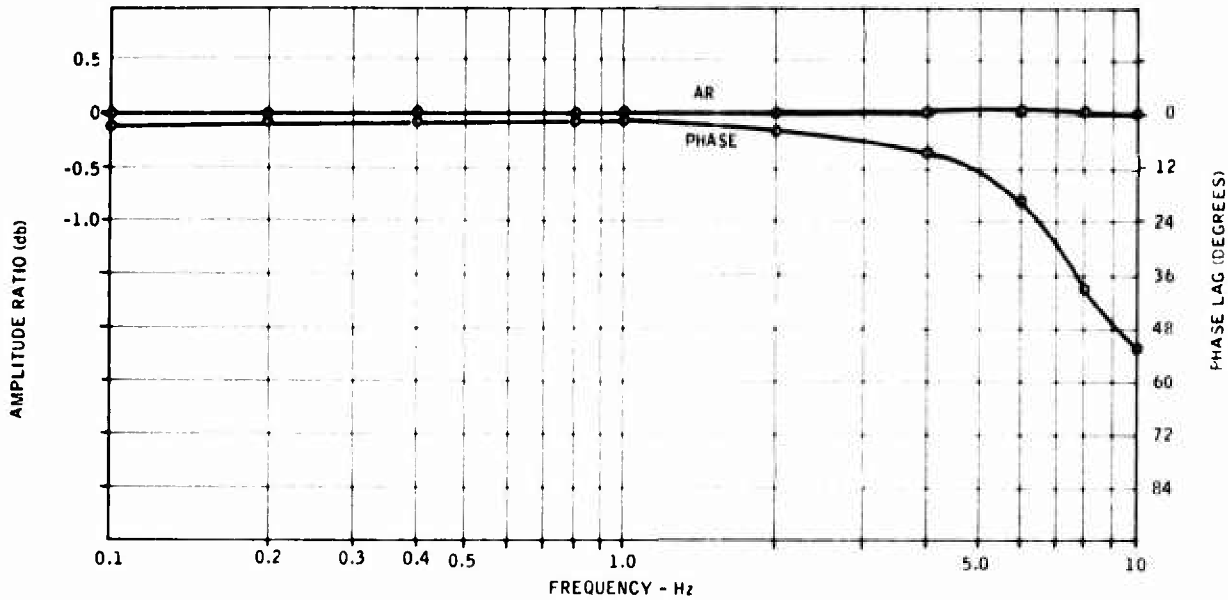

Figure 42. Hydrofluidic Servoactuator With 3-Amp Cascade.

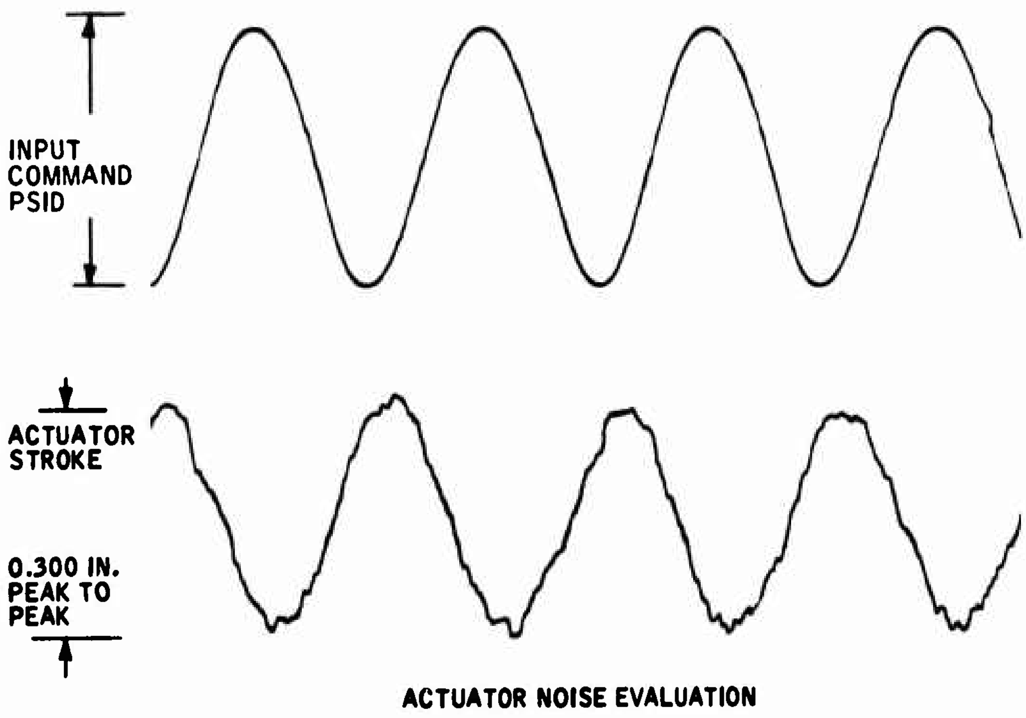

Figure 43. Actuator Noise Evaluation.

SECTION IV
CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

- The servoactuator developed under this program demonstrates the feasibility of using a high-pressure amplifier cascade to drive a conventional spring-centered spool valve and fluid summing in the feedback loop.
- The most efficient feedback summing utilizes a two-input summing amplifier.
- An improved method is needed for interfacing between the high-pressure output signal from the SAS and the servoactuator amplifier control input level. The present fluid amplifier (ACD-1) can be used as a passive element (no supply pressure) for interfacing, but with low efficiency.
- Output noise can be reduced significantly with improved amplifier design.

RECOMMENDATIONS

- An improved method of interfacing between the SAS output and servoactuator input amplifiers should be investigated. This could significantly reduce the number of amplifiers required.
- With feasibility demonstrated, a servoactuator package design program to meet a specific installation envelope is recommended. This should include improved amplifier and feedback transducer manifolding, and utilize the latest amplifier design and fabrication techniques as demonstrated in the Noise Study.