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HYDROFLUIDIC SERVOVALVE DEVELOPMENT

Holger C. Kent

Honeywell, Incorporated

Prepared for:

Army Air Mobility Research and Development Laboratory

June 1975

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USAAMRDL-TR-75-21

HYDROFLUIDIC SERVOVALVE DEVELOPMENT

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

June 1975

Final Report for Period December 1973 - December 1974

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Prepared for

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

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EUSTIS DIRECTORATE POSITION STATEMENT

The purpose of the program was to investigate and evaluate the use of poppet-type valves in place of the conventional spool and sleeve valves in the servovalve function of hydrofluidic stability augmentation system servoactuators.

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

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1. REPORT NUMBER USAAMRDL-TR-75-21	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) HYDROFLUIDIC SERVOVALVE DEVELOPMENT		5. TYPE OF REPORT & PERIOD COVERED Final Report December 1973 - December 1974
		6. PERFORMING ORG. REPORT NUMBER W0565-FR
7. AUTHOR(s) Holger C. Kent		8. CONTRACT OR GRANT NUMBER(s) DAAJ02-74-C-0015
9. PERFORMING ORGANIZATION NAME AND ADDRESS Honeywell, Inc., Government and Aeronautical Products Division Minneapolis, Minn. 55413		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS 62114A 1T162114A645 01 002 EK
11. CONTROLLING OFFICE NAME AND ADDRESS Eustis Directorate U.S. Army Air Mobility R&D Laboratory Fort Eustis, Va. 23604		12. REPORT DATE June 1975
		13. NUMBER OF PAGES 55
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS. (of this report) Unclassified
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Hydrofluidic Servovalve
Poppet-Type Servovalve

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This report covers the analysis, detail design, fabrication, and evaluation of a poppet-type servovalve to be used in servoactuators for hydrofluidic stability augmentation systems. Initial analysis and design included review of poppet valve principles, various design configurations and preliminary tests of poppet valve operation and flow forces. Evaluation included static, dynamic, and limited environmental tests. Although servovalve performance requirements were not met, it was shown that poppet valving can be mechanized to operate as a servovalve in driving an actuator piston.

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PREFACE

This document is the final report of a program to analyze, design, fabricate, and evaluate a poppet-type servovalve to be used in servo-actuators for hydrofluidic stability augmentation systems. The program was administered by the Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory, under Contract DAAJ02-74-C-0015. This program is part of the Army's effort to improve hydrofluidic servo-valve reliability and performance as applied to hydrofluidic stability augmentation systems of helicopters. The work presented started December 1973 and was completed December 1974.

This program was conducted under the technical management of Mr. G. W. Fosdick. Technical consultation was provided by R. F. Rasmussen of TRI-TEC Associates, Minneapolis, Minnesota.

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INTRODUCTION

The objective of the Hydrofluidic Servovalve Development program was to develop a servovalve for increased reliability and performance of hydrofluidic stability augmentation system application. The servovalve was to interface with a Government-furnished servoactuator from Contract DAAJ02-72-C-0051, or similar actuator.

The program called for development of a poppet-type servovalve to replace the present spool-sleeve configuration. Special emphasis was to be directed at the following servovalve associated problem areas: 1) contamination sensitivity, 2) dynamic performance, 3) high cost, and 4) fabrication complexity.

The program consisted of a 10-month effort which included analysis and detail design, fabrication and evaluation of a hydrofluidic poppet-type servovalve. Evaluation included static, dynamic, and limited environmental tests. Evaluation was performed using a Honeywell-fabricated actuator and test block which replaced the unavailable Government-furnished actuator.

REVIEW OF BASIC PRINCIPLES

The use of a poppet valve for servovalve applications in a fluidic control system requires careful study of the three basic elements:

a) Poppet valving elements

b) Means for driving with fluidic devices

c) Means for applying actuator position feedback

Because each of these elements has specific characteristics which naturally impose restrictions on their application, it is important that each be understood thoroughly before system integration is attempted.

POPPET VALVE

There are probably more poppet-type valves produced than any other kind, due primarily to the success of this design as applied to internal-combustion engines. However, its application in liquid-flow control is just as popular because of its inherent insensitivity to contamination. High-pressure, low-volume pumping applications are examples: where relatively clean water is used in pressure washing applications, the poppet-type valve is by far the most popular style of valving element for pump suction and discharge valves, relief valves, unloader valves, and chemical selector and proportioning valves. Wherever the contamination levels of ordinary tap water create a problem, the popped valve has served admirably.

This insensitivity is quite easy to understand when the detail design of a poppet valve is closely examined (Figure 1).

Note that the flow path is directed immediately over the relatively sharp edge of the valve seat. If the edge were infinitely sharp, there would be no place for contaminants to be trapped; they would be forced out of the valve opening by the wedging effect in either direction.

When the poppet valving elements are used without stems, guides or actuators, they can serve as simple check valves (a frequent practice), but if they are used as flow control elements with a limited amount of operating force, some means must be used to compensate for the inherent hydrostatic imbalance of the configuration. The simplest solution is to add a "balance piston" (Figure 2) whose area is equal to that of the metering opening.

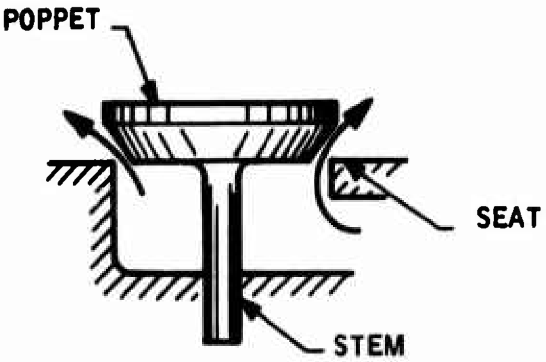

Figure 1. Basic Poppet Valve

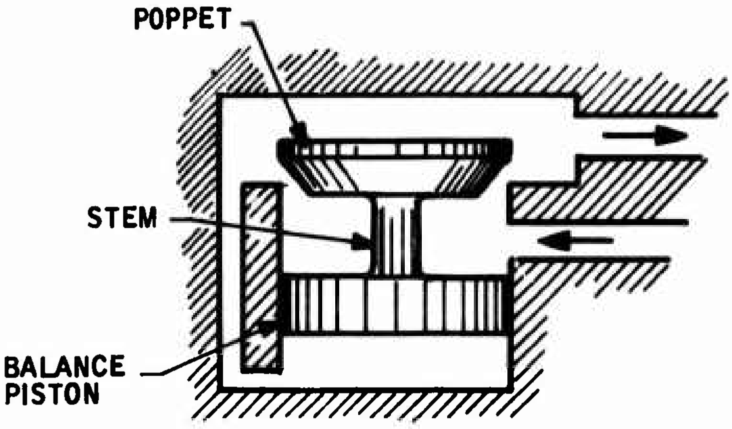

Figure 2. Balanced Poppet Valve

In this arrangement, the hydrostatic forces of the poppet valve elements are in perfect balance; inlet and outlet pressures may vary as loads demand, and little or no effect on the valve-operating force should be experienced. Flow forces and other dynamic forces on the valve element must be given careful consideration, however, and it is the intent of this study program to show that such forces are manageable.

In many cases, it is inconvenient to operate even a balanced poppet valve with the operating linkage submerged in one of the flow lines directly connected to the poppet. In such instances, an "isolated" poppet valve (Figure 3) is convenient.

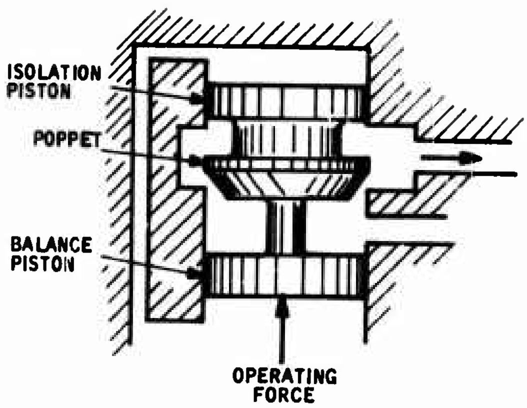

Figure 3. Isolated Poppet Valve

In this arrangement, a second, or "isolation", piston is added to completely isolate the poppet flows from passages containing the operating elements. Note that the valve is in complete hydrostatic balance.

FLUIDIC DRIVING ELEMENTS

The original poppet valve proposal considered the use of turbine-type "buckets" as a means of generating valve operating forces with a fluidic command signal. These forces were then proposed to be used to operate a nozzle-flapper valve whose output could be used to position a poppet valve (Figure 4).

In this arrangement, the differential flow rates from the command amplifier are converted to a force on the valve drive arm by the fluid momentum and directional change. This force (neglecting the forces produced by the velocity and position feedback springs for the time being) is, in turn, used to create differential pressures in the nozzle-flapper valve. The output of the nozzle flapper valve moves the poppet beam and positions the poppets which control fluid flow to the cylinder.

A simpler arrangement, if the poppet operating forces are small enough, is illustrated in Figure 5.

In this arrangement, output from the fluidic command amplifier is used to pressurize a pair of driver bellows. Force output from the bellows is applied to a lever which is directly coupled to a pair of poppets. Output from the poppets drives the servocylinder. Note that no nozzles, flapper or flow control orifices are used; obviously, this arrangement is limited to applications where output flow requirements are minimal.

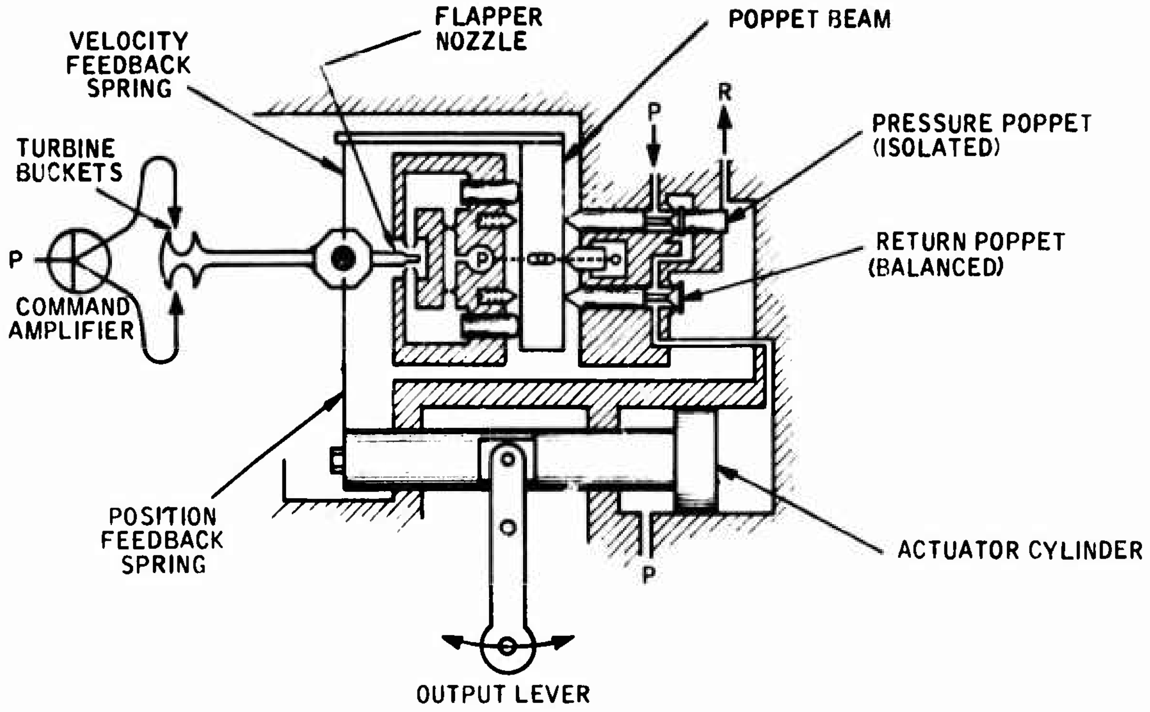

Figure 4. Two-Stage, Turbine Bucket-Driven Poppet Valve

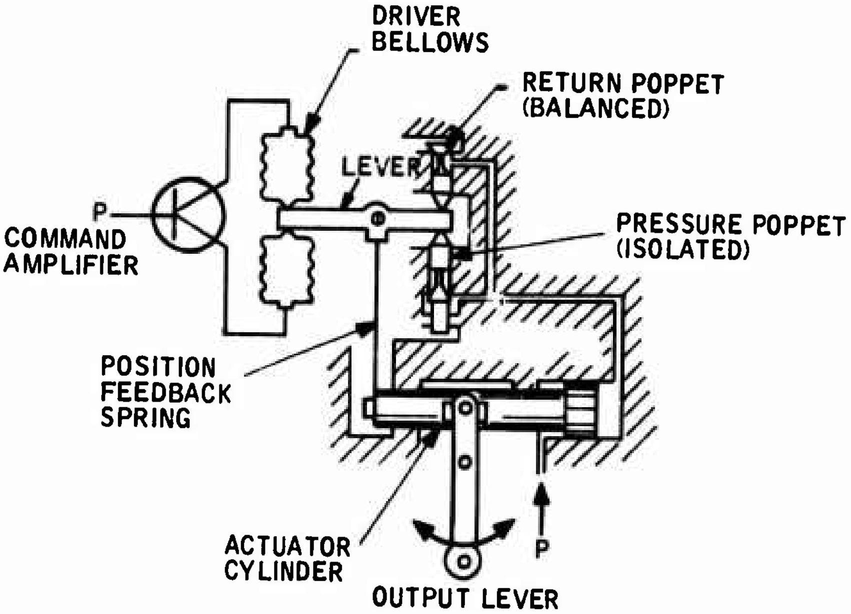

Figure 5. Single-Stage, Bellows-Driven Poppet Valve

POSITION FEEDBACK SYSTEM

To complete the function of a servoactuator, some method must be provided to supply actuator cylinder position feedback information. For fluidic-controlled poppet-valve actuators, there are two basic methods: one uses a position feedback spring to develop a position-proportional force, and the second uses a fluidic transducer to produce a position-proportional differential flow. The latter might appear in block diagram form as shown in Figure 6.

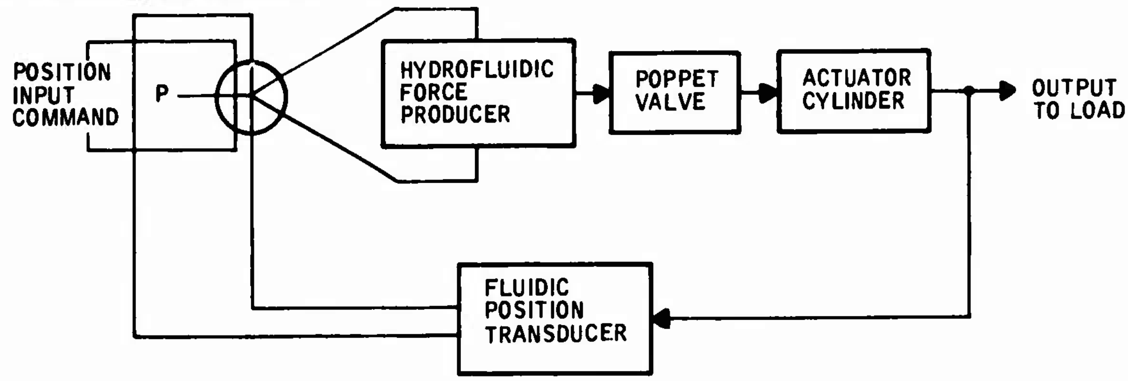

Figure 6. Fluidic Feedback System

Note that this configuration performs the feedback loop closure with non-mechanical fluidic elements; flow from the feedback transducer is summed with flow from the position command source, and the difference (error) is amplified to provide the servo gain that is required. While a direct-drive poppet configuration is shown, a two-stage arrangement using a nozzle-flapper or a jet-pipe first stage could be used as well. The force producer can be either a bellows, a bucket, or a valve device, so long as it is compatible with the characteristics of the amplifier.

However, most contemporary hydrofluidic servoactuators use a somewhat different form of positional feedback. Although it is probably a carryover from two-stage electrohydraulic servovalve design, the principle is well tested and proven from thousands of installations in aircraft, spacecraft and industry. A representative block diagram is shown in Figure 7.

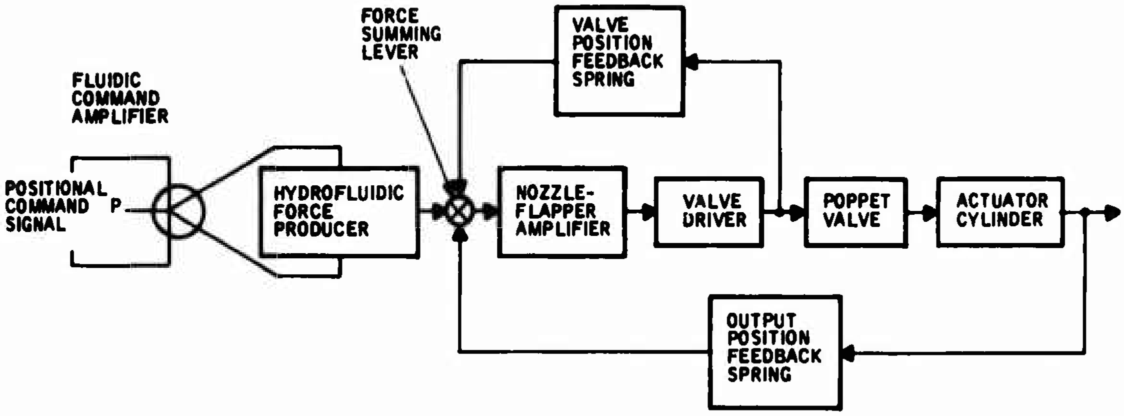

Figure 7. Spring Feedback System

This arrangement, also represented by the configuration shown in Figure 7, used feedback springs in two places -- one for feedback of the output position and the second for velocity compensation in the form of valve position. Both are desirable if maximum performance is to be achieved, but the use of both requires adequate driving power in the form of force producer (bellows or buckets) output. This latter limitation often restricts overall valving system selection.

INITIAL STUDIES AND TESTING

Before a prototype valve design could be attempted, studies of poppet valve flow/force characteristics were deemed advisable. While a limited effort was made to obtain information on poppet valve research, little reliable data was obtained. Consequently, studies were limited to two areas: 1) analytical and design studies of the feedback system capable of operating within the contractual constraints, and 2) analytical and empirical studies of the basic poppet valve.

FEEDBACK SYSTEM DESIGN

The methods of obtaining and summing feedback with the command signal as shown in the previous section have many variations. In the interest of simplicity, and as a result of preliminary calculations, a decision was made to make a single-stage poppet valve with spring force feedback the initial objective. Once gains and other parameters could be defined, testing of appropriately sized poppets would verify the selected approach. A block diagram of the selected approach is shown in Figure 8.

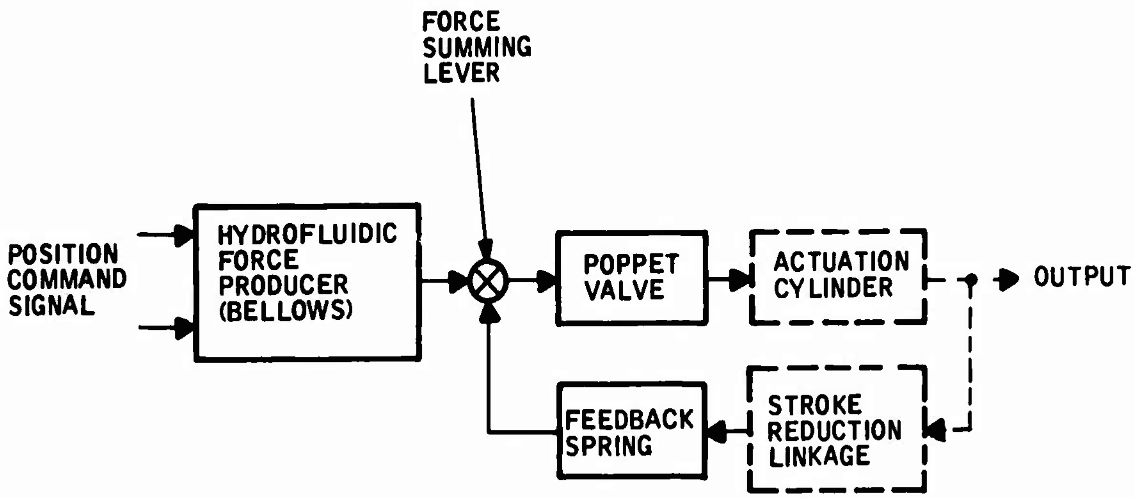

Figure 8. Block Diagram, Selected Approach

The contract had previously specified a particular actuator cylinder to be used, and this defined the following parameters:

- Valve mounting dimensions
- Fluid flow rates and pressure
- Feedback stroke and spring parameters
- Dynamic response and internal gain limits
- Force producer capacitance

A design study was initiated to determine the optimum arrangement of elements for a valve fitting the requirements. Because of the constraints imposed by the mounting requirements, only the configurations shown in Figure 9 were considered.

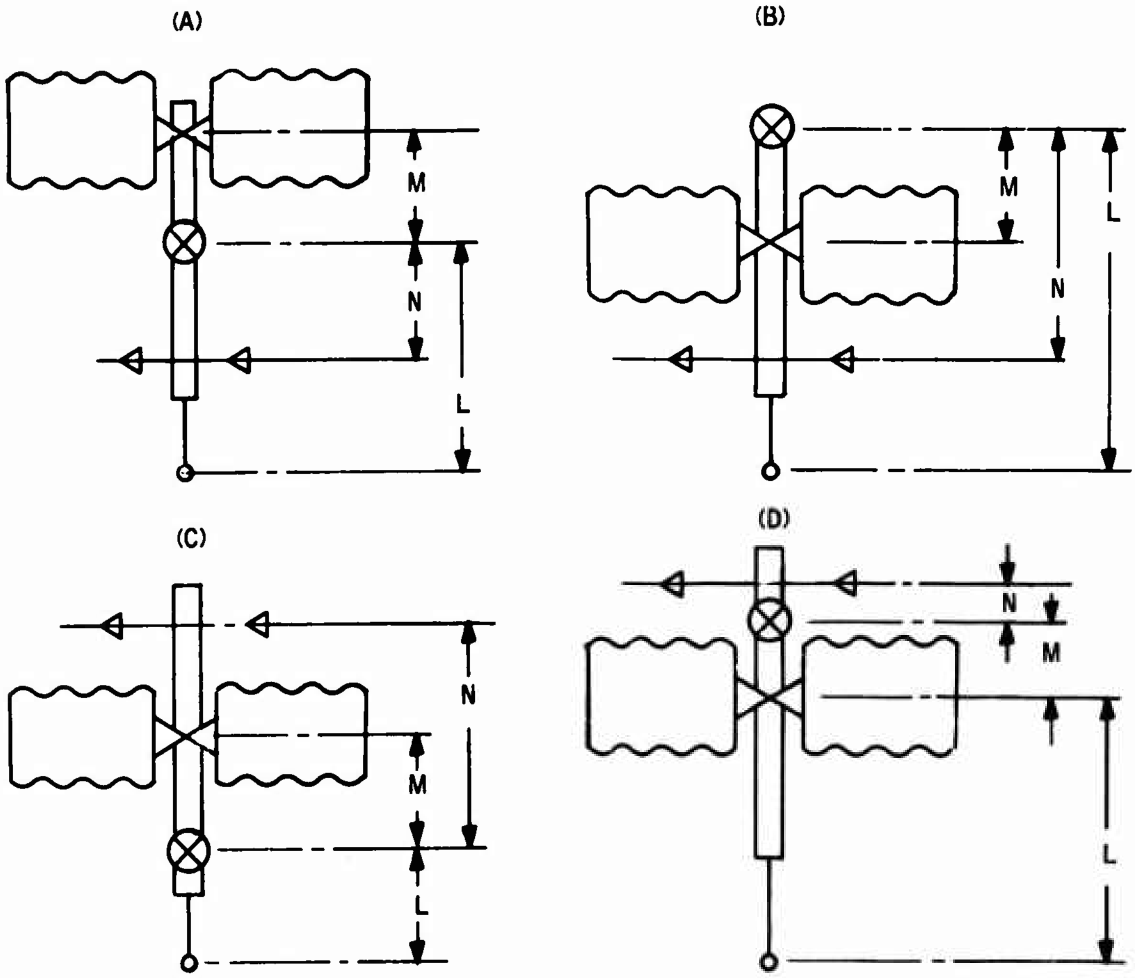

Figure 9. Pivot Location Configurations

These four arrangements were evaluated on the basis of:

- Ability to achieve functional dimensions
- Bellows sizing within achievable limits
- Poppet sizing such that stable operation would result
- Ability to be mass balanced for vibration resistance

From this study configuration (A) was selected, and dimensions were established on the basis of the schematic as shown in Figure 10.

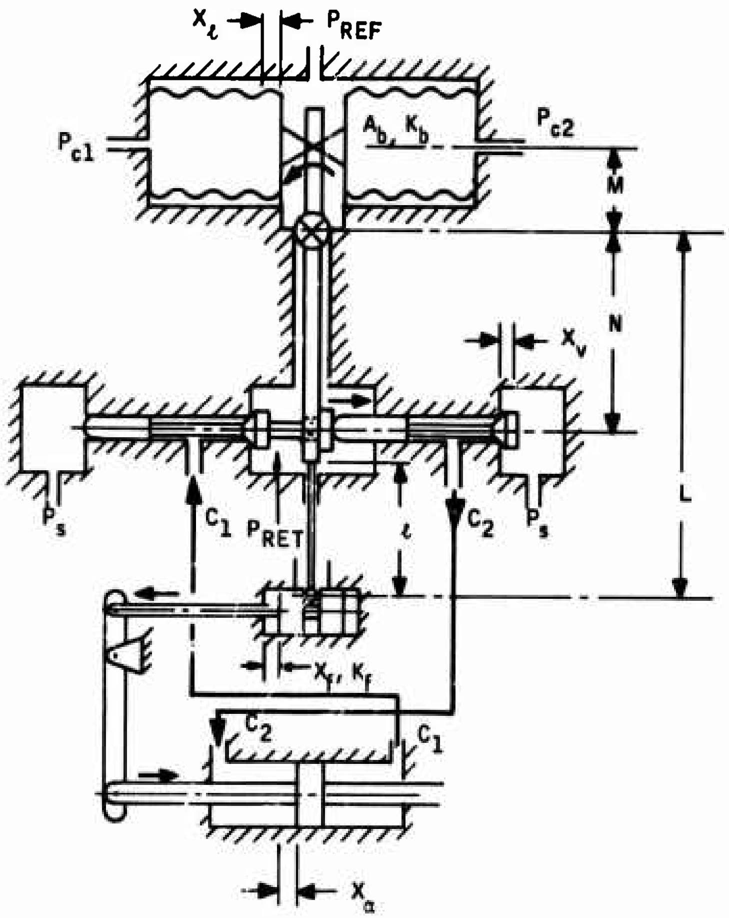

Figure 10. Configuration Schematic

In this figure, only one pair of poppets is shown. The one on the left controls the "cylinder-to-return" flow of cylinder side 1 (C1) and the poppet on the right controls the "pressure-to-cylinder" flow to cylinder side 2 (C2). Because opening of both of these valves simultaneously will only control the velocity of the servocylinder in one direction, a second pair of valving elements is mechanically cross-coupled to the operating lever so that the opposite action takes place when the lever is moved to the left. This is done by mounting the second set of elements alongside those shown; a horizontal section through both sets would appear as shown in Figure 11.

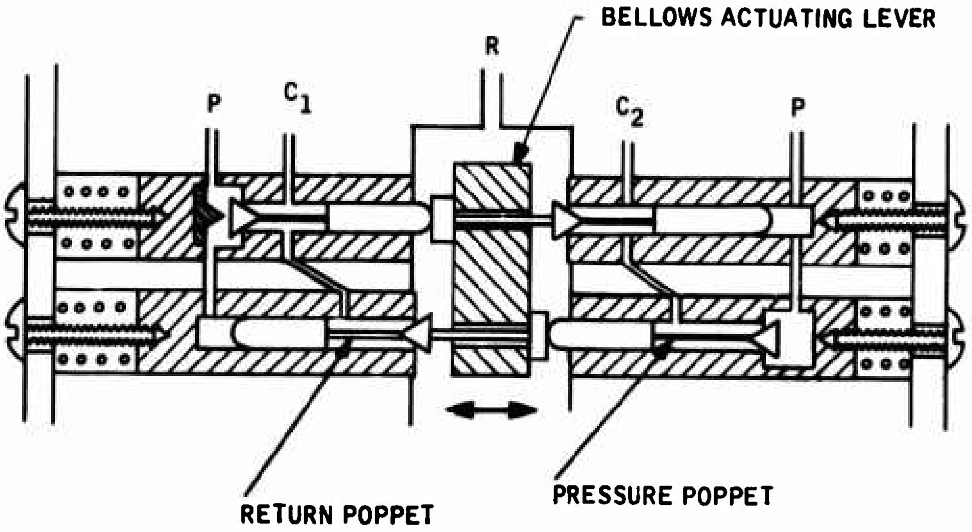

Figure 11. Poppet Orientation

Relationships between the elements of Figure 10 can be expressed in the following analysis: Two conditions must be satisfied; the static, full-stroke, null-torque balance and the dynamic (maximum velocity) torque balance. The static, full-stroke, null-torque balance is

$$P_m A_b M = X_f K_f L \tag{1}$$

where

P_m = maximum ΔP pressure input (psi)

A_b = bellows area (in.^{2})

M = bellows pivot arm (in.)
X_f = feedback spring deflection (in.)
K_f = feedback spring rate (lb/in.)
L = feedback pivot arm (in.)

The dynamic torque balance equation is

$$P_\ell A_b M = K_b X_\ell M + K_f X_f L + F_v N \tag{2}$$

Substituting $X_f = X_\ell\ L/M$ and $N = M\ X_v/X_\ell$

$$P_\ell A_b M = K_b X_\ell M + K_f X_\ell \frac{L^2}{M} + F_v M \frac{X_v}{X_\ell} \tag{3}$$

$$P_\ell A_b = K_b X_\ell + K_f X_\ell \frac{L^2}{M^2} + F_v \frac{X_v}{X_\ell} \tag{4}$$

where

P_ℓ = bellows ΔP pressure input for maximum velocity (psi)
K_b = bellows spring rate (lb/in.)
X_ℓ = bellows deflection for maximum velocity (in.)
F_v = poppet valve force (lb)
X_v = poppet displacement for maximum velocity (in.)

Solving Equations (1) and (4) for K_f and solving for L/M we have

$$\frac{L}{M} = \frac{X_f P_\ell A_b - K_b X_\ell X_f - F_v \frac{X_v}{X_\ell} X_f}{X_\ell P_m A_b} \tag{5}$$

The valve force (F_V) is comprised of the valve flow reaction forces, friction forces, and installed spring forces, if required. Since the reaction and friction forces are difficult to predict, they are neglected for initial calculation purposes. To allow for these valve forces, the bellows chosen will have a spring rate less than that calculated (more convolutions). Therefore, Equation (5) becomes

$$\frac{L}{M} = \frac{X_f P_\ell A_b - X_\ell X_f K_b}{X_\ell P_m A_b} \tag{6}$$

Now a relationship between L/M and the bellows spring rate (K_b) and area (A_b) can be determined. The bellows input pressure (P_ℓ) and deflection (X_ℓ) for maximum actuator output velocity are determined from the servo-actuator requirements

$$P_\ell = \frac{V_m P_m}{K_\ell X_a} \qquad (7)$$

where

V_m = actuator output maximum velocity = 2 in./sec

K_ℓ = servoactuator loop gain = 70 sec^{-1}

X_a = actuator travel = 0.3 in.

P_m = 2 psid

$$P_\ell = \frac{2(2)}{70(0.3)} = 0.190 \text{ lb/in.}^2$$

The bellows deflection (X_ℓ) is limited by the required bellows capacitance (C_b) and the bellows area (A_b)

$$X_\ell = \frac{P_\ell Cb}{A_b} \qquad (8)$$

$$X_\ell \frac{0.190(0.0006)}{0.50} = 0.000228 \text{ in.}$$

where

P_ℓ = 0.190 lb-in.^{2}

C_b = 0.0006 in.^{3}/psi

A_b = 0.50 in.^{2}

From Equation (6), L/M was determined for various values of K_b and A_b until a value of L/M was attained which satisfied the physical and operational requirements. For this case,

$$\frac{L}{M} = 7.06$$

where

X_f = 0.01875 in.

P_ℓ = 0.190 psi

A_b = 0.500 in.^{2}

X_ℓ = 0.000228 in.
K_b = 40 lb/in. (2 conv.)
P_m = 2 psi

The poppet valve displacement (X_V) required for maximum actuator velocity for a poppet diameter (D) is

$$Q = 103A\sqrt{P_s} \tag{9}$$

where

Q = maximum flow through poppet = $A_a V_a$ (in.^{3}/sec)
A_a = area of actuator (in.^{2})
V_a = velocity of actuator (in./sec)
A = poppet curtain area = $\pi D X_v$ (in.^{2})
P_s = supply pressure (psi)

$$X_v = \frac{Q}{103\pi D\sqrt{P_s}} = 0.000757 \text{ in.} \tag{10}$$

where

Q = $A_a V_a$ = 0.15 (2) = 0.3 (in.^{3}/sec)
D = 0.05 in.
P_s = 600 psi

Determination of the poppet valve displacement arm (N) is based on the value of M, X_ℓ, and X_V. A value of L + M = 1.75 in. was chosen to provide for minimum configuration height. Thus, for

$$\frac{L}{M} = 7.06$$

$$L = 7.06M$$

$$L + M = 7.06\,M + M = 1.75$$

$$8.06M = 1.75$$

$$M = \frac{1.75}{8.06} = 0.217 \text{ in.}$$

and

$$N = \frac{MX_y}{X_\ell} = \frac{0.217(0.000757)}{0.000228} = 0.720 \text{ in.}$$

$$L = 7.06\ (0.217) = 1.532 \text{ in.}$$

The feedback spring dimensions can be calculated from the spring rate (K_f) and the deflection equation for circular beam. The spring rate from Equation (1) is

$$K_f = \frac{P_m A_b M}{X_f L}$$

$$K_f = \frac{2(0.5)\ (0.217)}{0.01875\ (1.532)} = 7.55 \text{ lb/in.}$$

With the feedback spring force (w) at 0.01875-in. deflection equal to 0.142 lb and a spring length (ℓ) of 1.250 in., the diameter (D) of the feedback spring becomes

$$D = \sqrt[4]{\frac{64\, W \ell^3}{X_f\, \pi\, 3\, E}} \qquad (11)$$

$$D = \sqrt[4]{\frac{64\ (0.142)\ (1.250)^3}{0.01875(3.14)\ (3)\ (30\text{x}10^6)}} = 0.0427 \text{ in.}$$

With the feedback system establishing practical dimensions for all of the valve elements, the way was clear for further work on the poppet itself.

POPPET VALVE STABILITY

The flow/pressure relationships defined by the contract required that the poppet valve elements be quite small compared to conventional spool-type devices. The poppet/sleeve combination selected for the prototype evolved as shown in Figure 12.

Figure 13 defines the size involved.

The next task was to determine functional characteristics of the poppet unit by actual test. Accordingly, several poppet/sleeve sets were fabricated and a test fixture was made as shown in Figure 14.

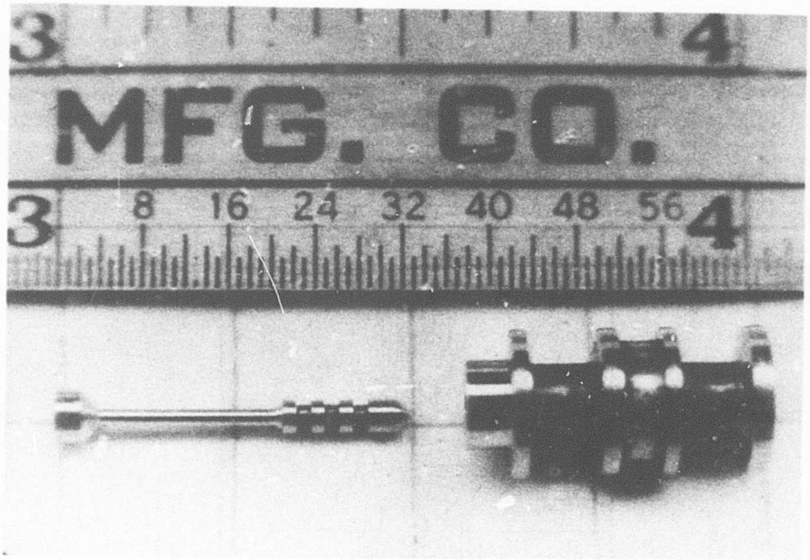

Figure 12. Photograph of Poppet and Sleeve

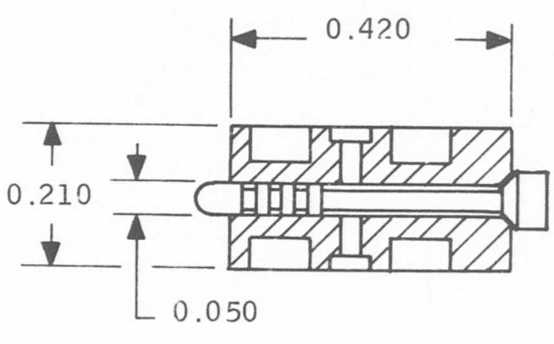

Figure 13. Poppet and Sleeve

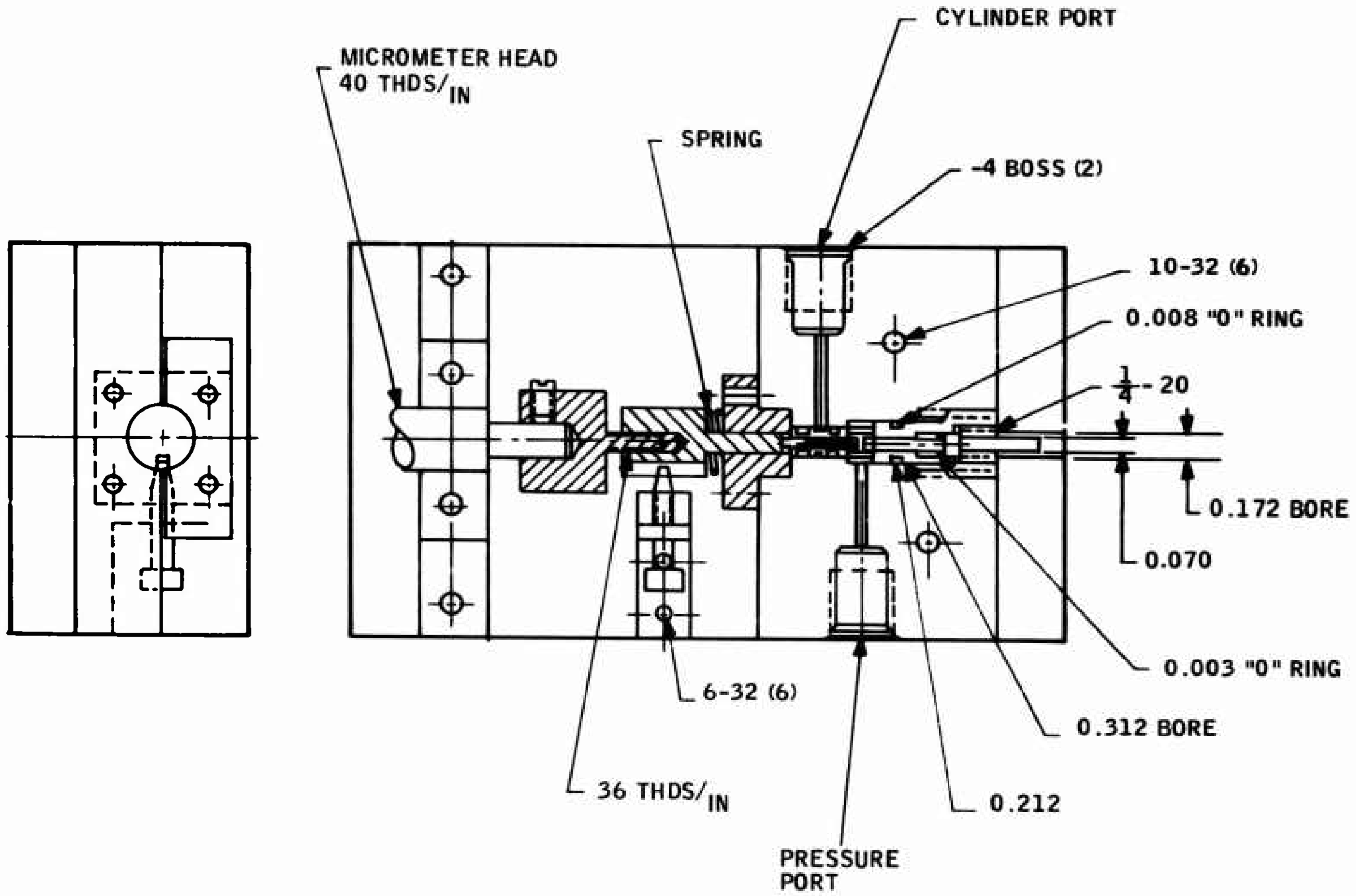

Figure 14. Poppet Valve Test Fixture

This test fixture includes means for manually opening and closing the poppet under test (the micrometer head on the left) and means for sensing valve instability of motion (sample rod on right). Note that the micrometer rod is directly coupled to a 36-pitch screw, thereby giving position control of the poppet via differential action, and providing a net motion of the poppet of 0.002777 in. per turn.

Initial flow and pressure gain measurements are shown in Figures 15 and 16.

While calculations indicated that flow forces on the poppet would probably be less than 0.1 lb at maximum flow, measurements on the test fixture confirmed that they would indeed be less than that value. Because extremely accurate flow force measurements would have required acquisition of more elaborate instrumentation, and because this would have been beyond the scope of this study, additional effort in this area was not attempted.

The test fixture confirmed that the poppet was very stable when operated within its design flow and stroke range, but also showed that its net stroke should be limited to keep the flow forces positive, thus ensuring that the poppet would always close. No indication of instability or poppet "float" was discerned. On this basis, the concept of a single-stage prototype was approved and design layout efforts began immediately.

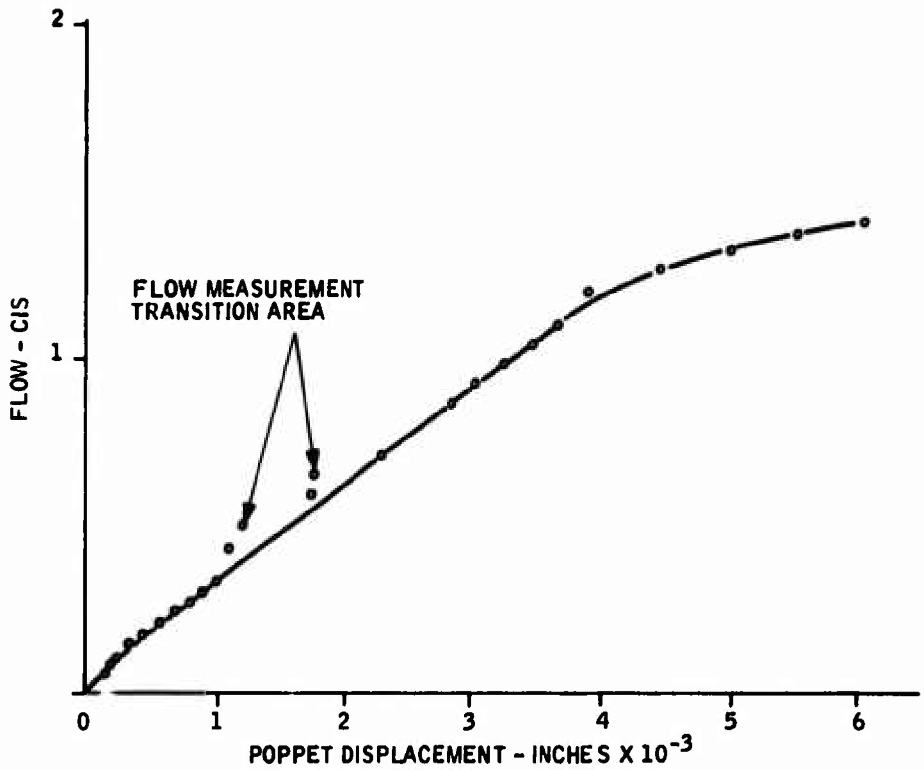

Figure 15. Flow Displacement, 300 psi Supply Pressure

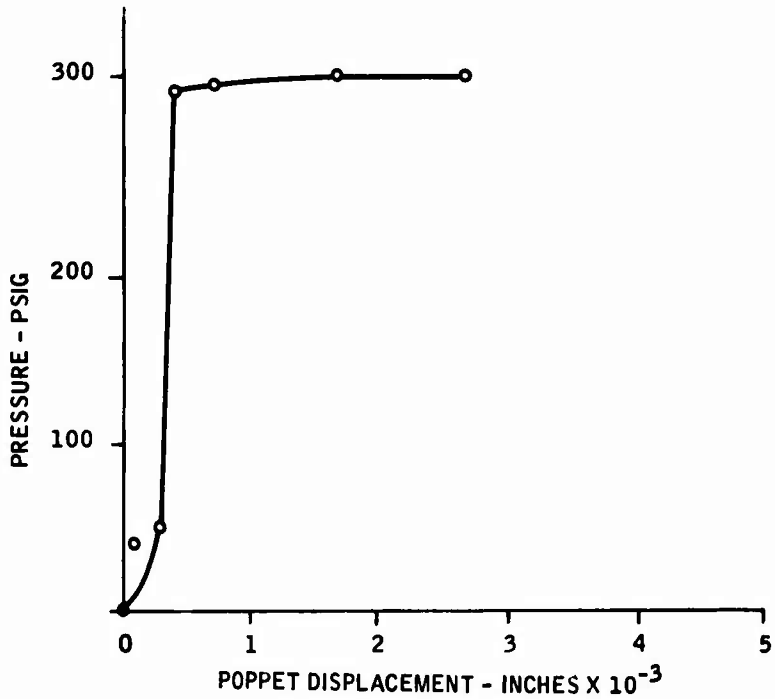

Figure 16. Pressure Displacement, 300 psi Supply Pressure

PROTOTYPE DESIGN

Design of the prototype valve followed the configuration shown schematically in Figure 10 because of the contractually specified installation. In layout form, that arrangement became as shown in Figure 17.

The three major areas of concern became:

- Bellows system
- Feedback summing lever
- Poppet valves and sleeves

All of these were packaged in an extremely compact housing (Figure 18) which could be mounted on the specified actuator.

Note that the valve housing is in two parts; the upper portion (#6, Figure 17) is made of aluminum and contains the bellows assembly. The lower portion (#7, Figure 17) with the integral mounting lugs, is made of steel (for temperature stability) and houses the poppet valves and their sleeves.

BELLOWS

Two electrodeposited nickel bellows (#5, Figure 17) are mounted in opposing manner with conical output ends centered in the force summing lever (#15, Figure 17). Each bellows consists of the basic bellows element soldered to (a) a conical output end, and (b) the internal "plug" (which also serves as a displacement limit). A 0.010-in.-diameter "vent" orifice is machined in the output cone as shown in Figure 19.

Each bellows is mounted to an end cap (#8, Figure 17) with a capscrew and with an intermediate spacer (#11, Figure 17) to provide any initial, coarse positioning of the bellows assembly (see Figure 20).

All porting to the bellows, P_{c1}, P_{c2} and P_{ref}, is completed internally via face-type seals to and through the lower housing block.

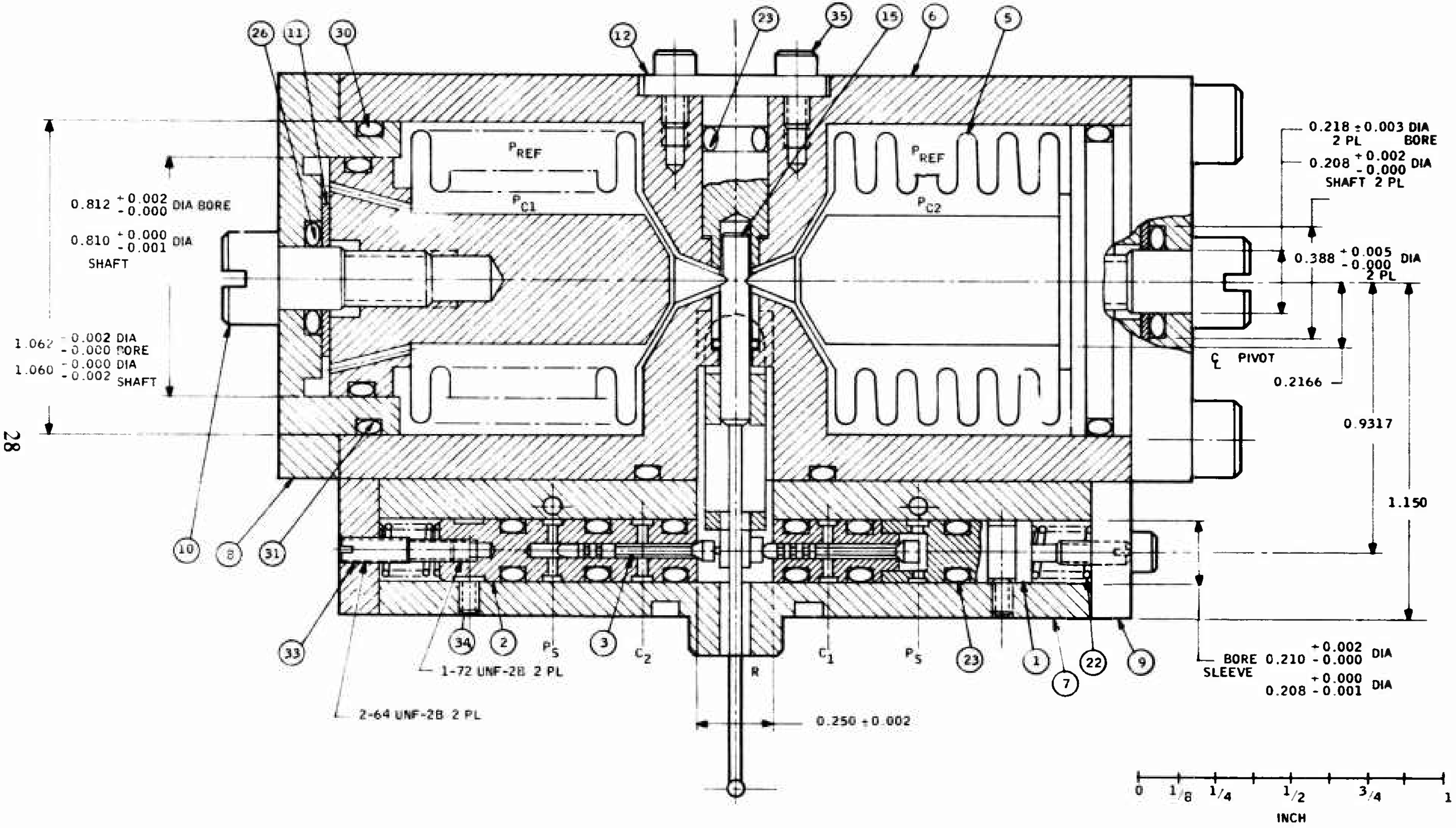

Figure 17. Poppet Valve Section

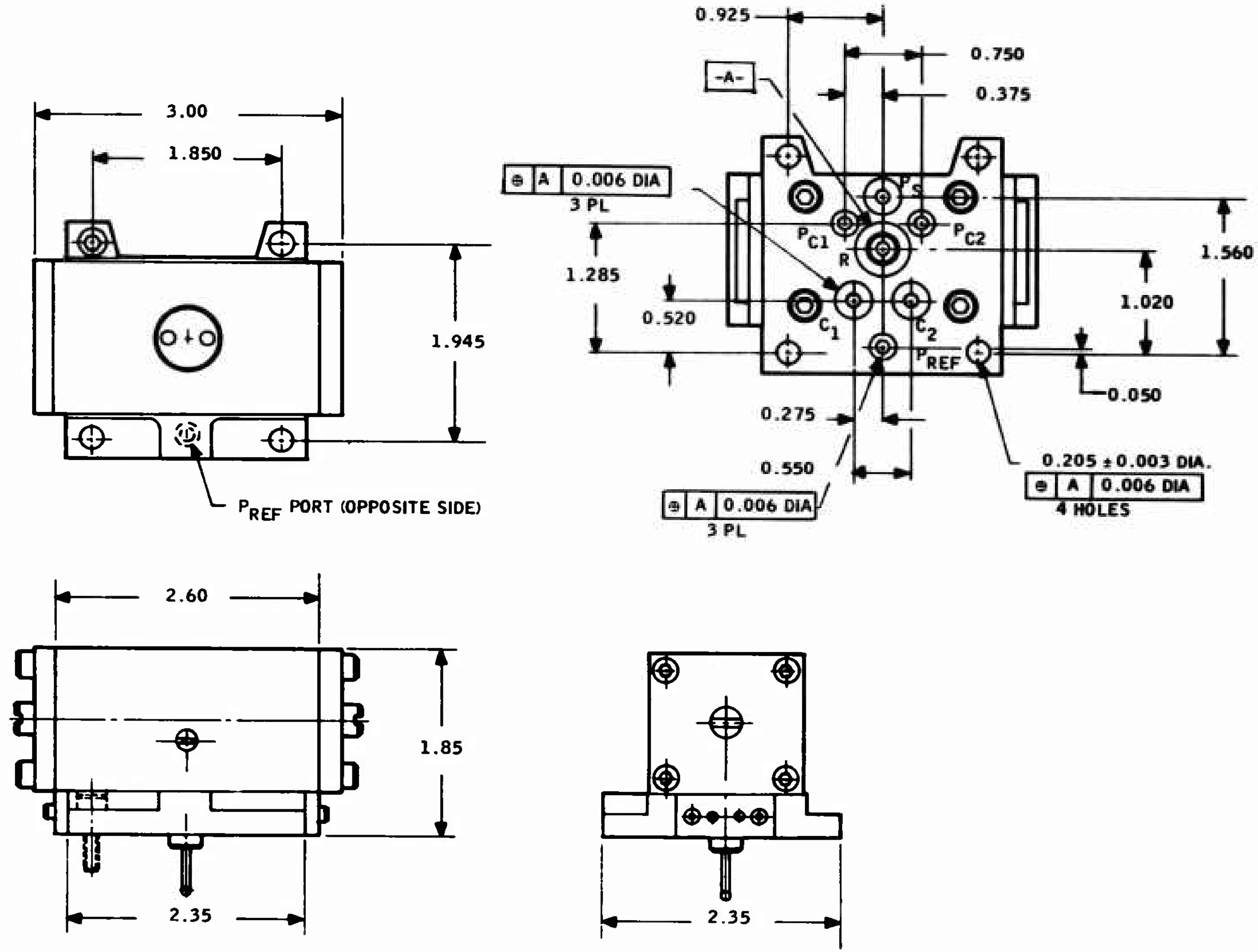

Figure 18. Valve Installation

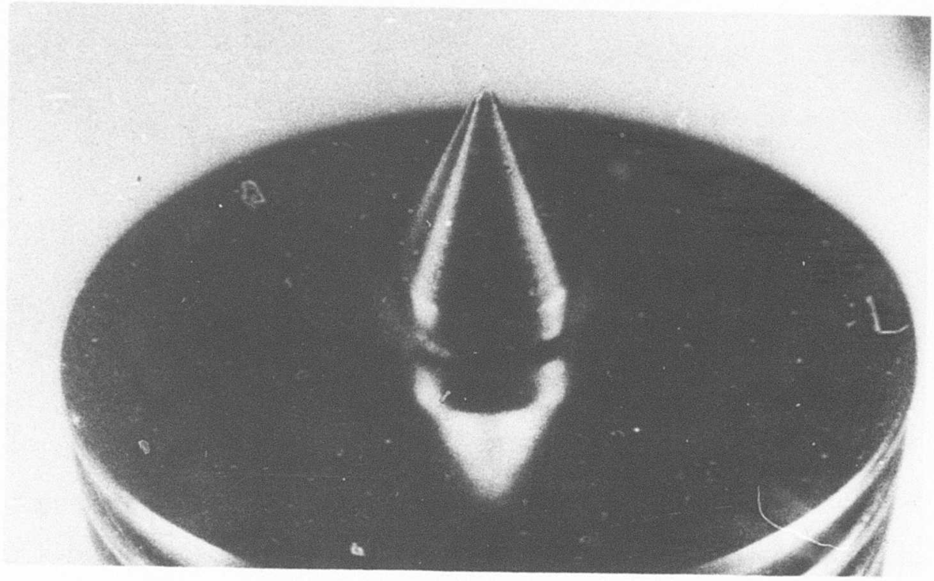

Figure 19. Bellows Cone Vent

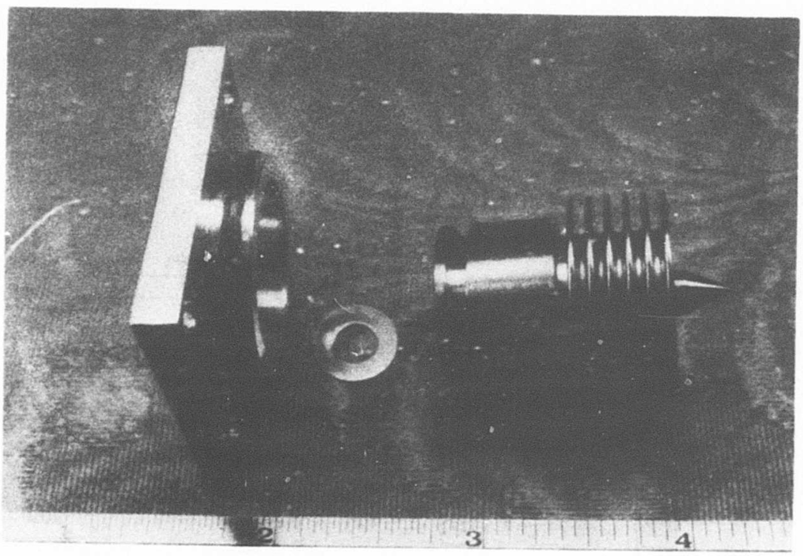

Figure 20. Bellows Assembly

SUMMING LEVER

A two-piece summing lever is used to connect the elements as prescribed by the schematic of Figure 10 and in the vertical section of Figure 21.

The hydraulic system Reference Pressure (P_{ref}) and Return Pressure (P_{ret}) must be kept separate; therefore, a seal is necessary somewhere along the summing lever. In the interest of minimizing friction, the most obvious location would be near or at the point of the pivot. Fortunately, it was possible to make these points coincident as shown in the layout and a standard O-ring is used (#32, Figure 21), mounted on the centerline of the pivot bearings (#14, Figure 21). Note that the pivot bearings are preloaded by a spring (#17, Figure 21) which holds the moveable pivot (#16, Figure 21) loaded through both bearings to the fixed pivot (#13, Figure 21).

The summing lever consists of two pieces: a center section, generally cylindrical, which has the "sockets" for the bellows cones at its upper end and an integral positional feedback spring at its lower end, is pressed into a lightweight "yoke" which contains the poppet valve actuators and the pivot bearings (Figure 22).

The ball on the lower end of the feedback spring fits into the precision groove in the feedback linkage of the matching actuator.

A view of the summing lever, its pivots and output actuators is shown in Figure 23.

An analysis was made of the mass balancing requirements, and it showed that ample space is available for the summing lever to be extended upward into the seal retainer (#12, Figure 17) thus adding adequate mass on the end above the pivot centerline and effecting an adequate counterbalance.

POPPET VALVES AND SLEEVES

Four poppet valves are used in this arrangement as dictated by the equal-area actuator cylinder. Two of these are shown in Figure 17, and they perform the following functions.

The C2-to-return poppet (#3, Figure 17) is maintained in zero lash contact with the pressure-to-C1 poppet (on the opposite side of the summing lever) by P_s directed at the left end of (#3) and at the head of the other poppet. Movement to the right by the summing lever yoke (#4, Figure 21) will cause simultaneous opening of both valves.

A second pair of poppets, connected to provide C1-to-return and pressure-to-C2 functions, is mounted alongside those shown in Figure 17 so as to open when the summing lever moves to the left.

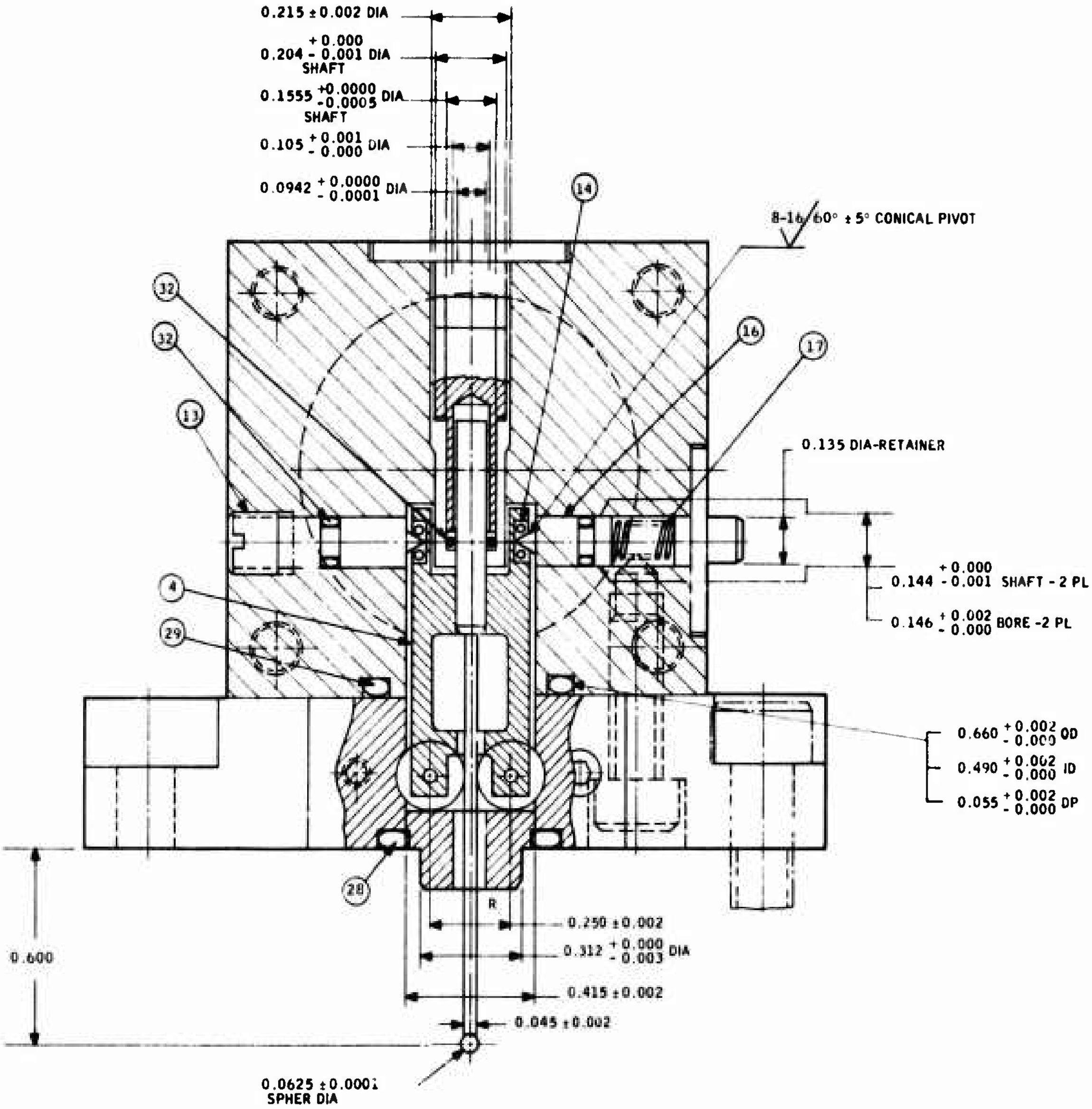

Figure 21. Summing Lever Schematic

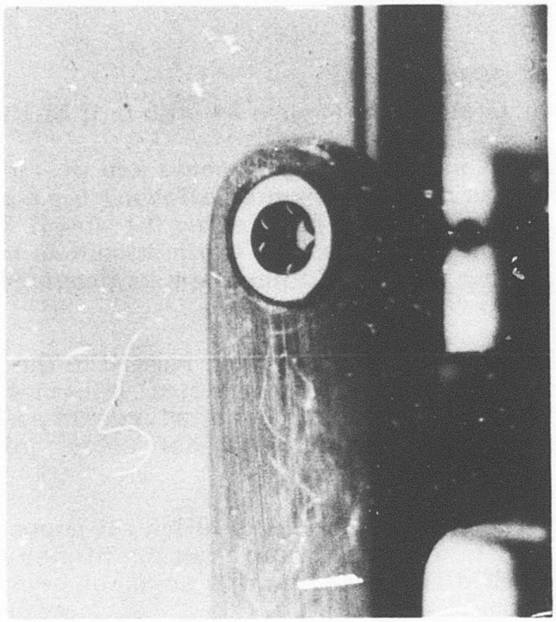

Figure 22. Summing Lever

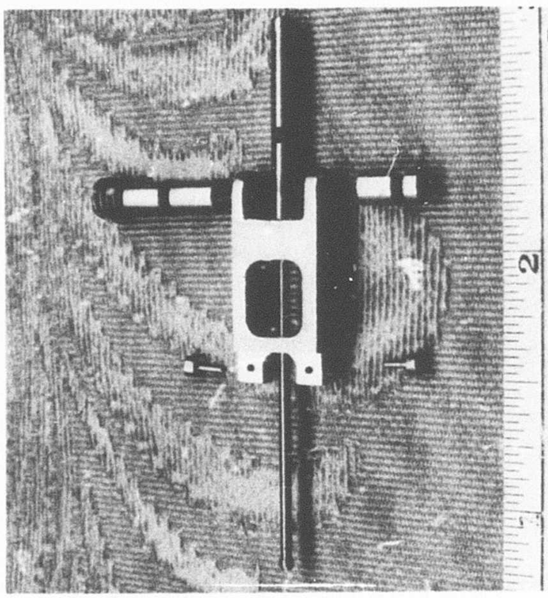

Figure 23. Summing Lever, Pivots and Output Actuators

A "pressure" poppet is shown in Figure 24 with half of its sleeve assembly.

Note (in Figure 17) that individual adjustments are provided for each poppet/sleeve element. This is accomplished by allowing for a small amount of axial sleeve adjustment range and positioning the sleeve with a five-pitch differential screw (#33). Backlash in the adjustment is loaded out by the spring (#22), and locking following adjustment is provided by a set screw for each sleeve (#34).

A word of explanation may be in order with regard to the two-piece sleeve of the pressure poppet (#1); because mechanical integrity is needed at this point, even though hydraulic security is not necessary, a press fit between these sleeves was used -- in production, a soft solder joint is contemplated at this point.

All porting is mutual to the steel block in which all poppet valves are mounted and directly connected to the mating surfaces for proper phasing with the specified actuator. All sleeves and mating surfaces are sealed with O-rings of standard, off-the-shelf sizes.

The entire unit results in a compact assembly, as shown in the photographs of Figures 25 and 26.

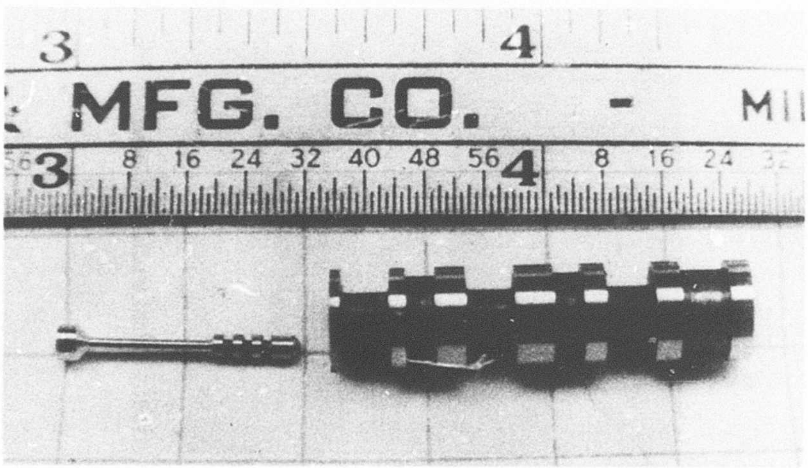

Figure 24. Pressure Poppet and Sleeve

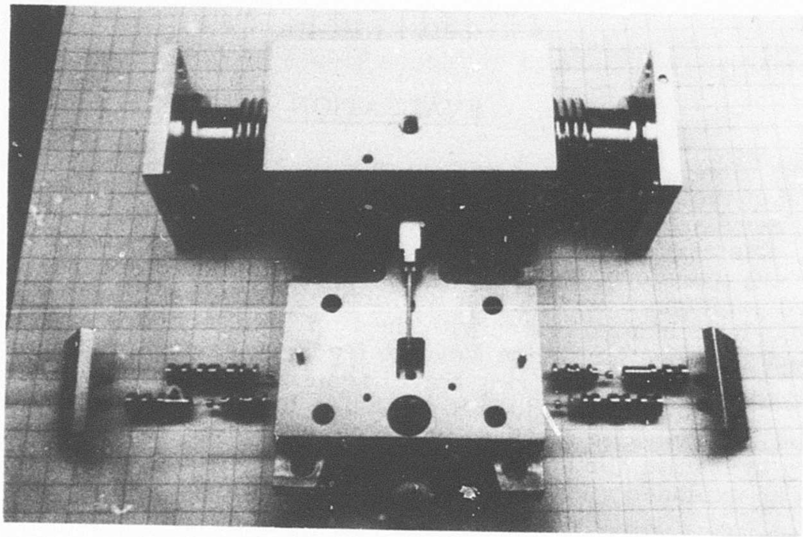

Figure 25. Exploded View of Servovalve

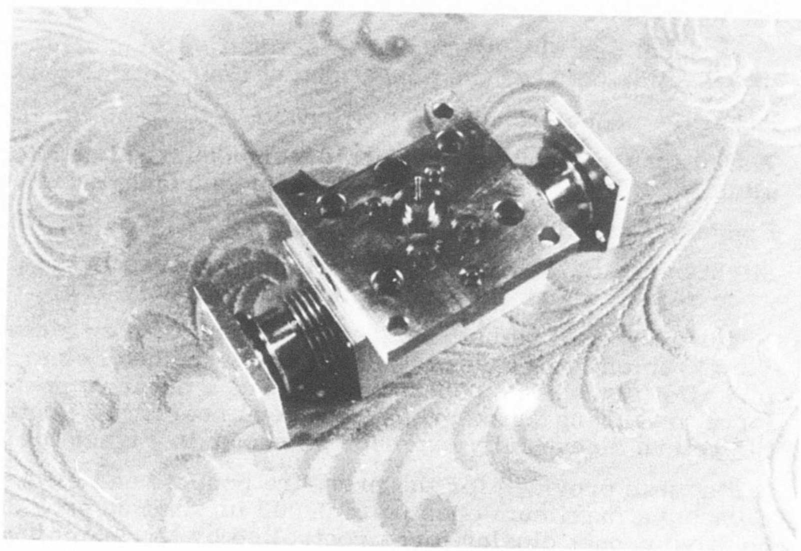

Figure 26. Bottom (Porting Plate) View of Servovalve

EVALUATION

Evaluation of the servovalve was to be accomplished using a Government-furnished servoactuator from a previous contract (DAAJ02-72-C-0051). However, this servoactuator was not available within the contract schedule requirement; therefore, servovalve tests and simulated servoactuator tests were performed using a test bed actuator and test block as shown in Figure 27. It includes a piston with the equivalent actuator area stroke volume and a groove for the servovalve feedback spring attachment. Stroke is limited to the feedback spring travel requirement (±0.020 in.). A means of locking the piston is provided to allow for locking the feedback spring during static servovalve testing. Also included in the test housing are the necessary hydraulic ports for pressure connections and test points. Figure 28 shows an exploded view of the test bed actuator and test block.

Initial poppet and sleeve adjustments encountered a number of problems which had to be solved before a push-pull valve operation was attained to provide for actuator extend and retract.

Problem: Hydraulic porting connection between the poppet pressure and actuator cylinder ports was not open, which prevented flow to the cylinder.

Solution: The poppet and sleeve housing was modified to provide porting intersection between the poppet output and cylinder port.

Problem: Pressure poppets could not be shut off.

Solution: The pressure sleeve poppet seats were deburred and honed slightly, which resulted in good seating surface and shut off.

Problem: Poppets were intermittently sticky.

Solution: Slight burrs in the pressure-equalizing grooves in the poppets were removed and the poppets functioned freely.

Problem: Pressure poppet adjustment resulted in off-center sleeve position. This problem was caused by large pressure head clearance (0.038 in.) allowing a large poppet opening and a negative poppet closing force, resulting in the pressure sleeve nearly fully retracted and the return sleeve fully extended as shown in Figure 29.

Solution: A stop was provided for the pressure poppet head clearance, allowing a maximum opening of 0.008 in., which provides a positive poppet closing force controlled by the summing lever (Figure 30).

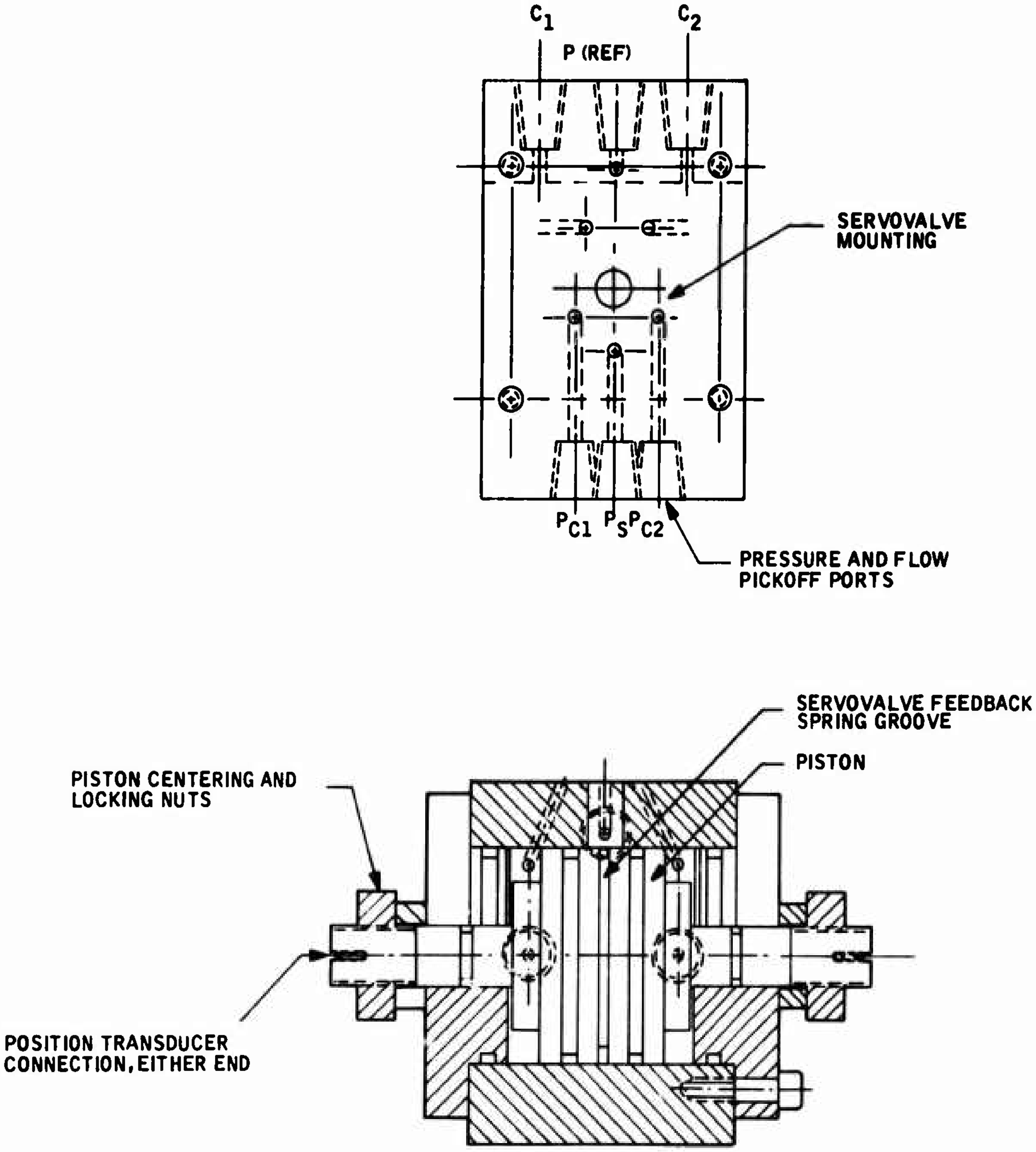

Figure 27. Servovalve Test Bed

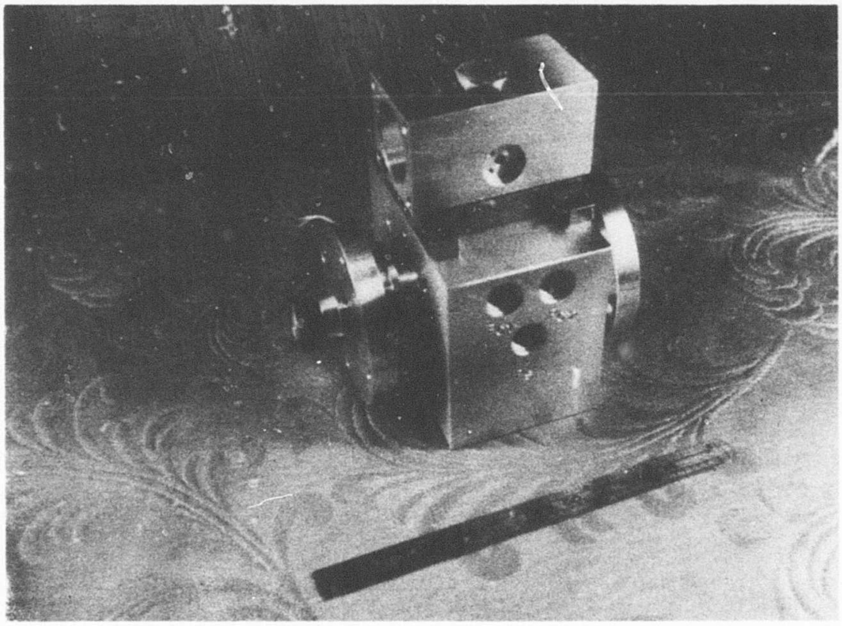

Figure 28. Exploded View of Servoactuator Test Bed

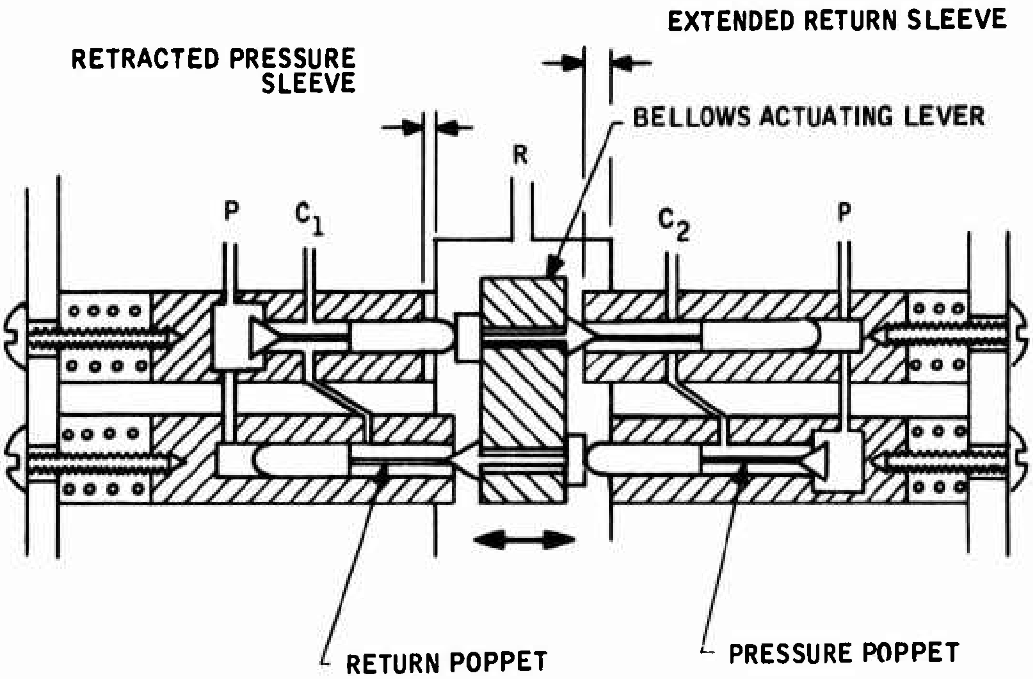

Figure 29. Off-Center Sleeve Position

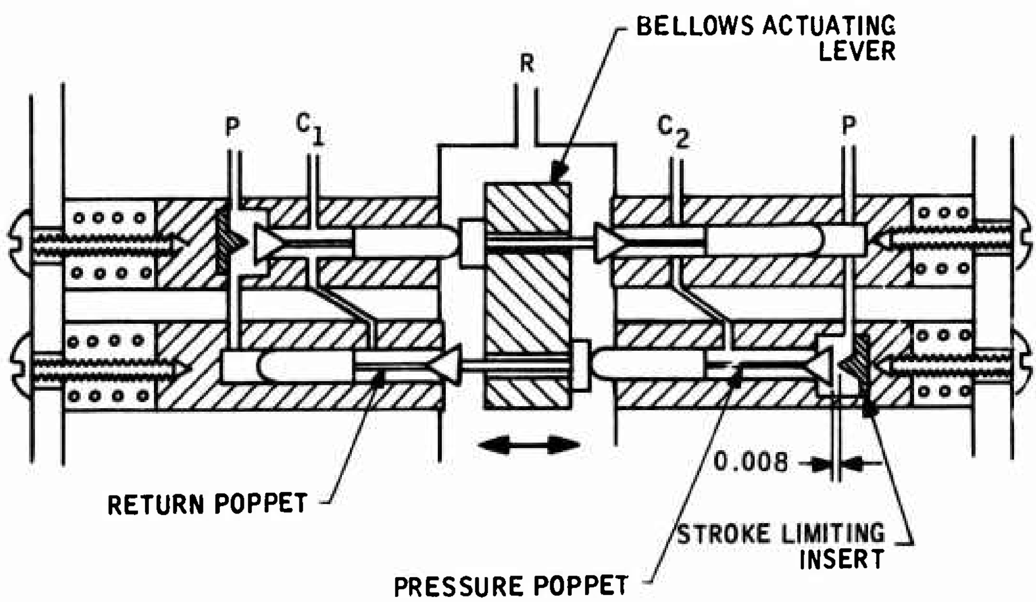

Figure 30. Pressure Poppet Stroke Limiting

With the above corrections made, adjustment still resulted in off-center sleeve positioning as shown in Figure 29. This was believed to be caused by unequal pressure forces on the return and pressure poppets, resulting in the return poppet displacing the pressure poppet until the return poppet is bottomed against the actuating lever. To determine if the pressure poppets were operating with positive closing pressure and if push-pull operation was possible, new poppet pegs were made which were pressed into the actuating lever, eliminating the return poppet from displacing the pressure poppet. This resulted in centered sleeve adjustment (proving pressure poppet positive closing) and push-pull operation with differential pressure into the input bellows.

Initial static pressure and flow gain curves were taken with the pressed-in poppet pegs, as shown in Figures 31 and 32, respectively. The test schematic is shown in Figure 33. The detailed specification and test plan are included in Appendixes A and B, respectively. Both curves show hysteresis and low gain values. The low pressure gain is probably due to poor return poppet adjustment which prevents pressure buildup due to the poppet not completely closing. This is also noted from the low pressure recovery (275 psi as compared to approximately 600 psi). The flow gain is shown to be 0.024 cis/psid as compared to the required 0.3 cis/psid and the maximum flow of 0.04 cis, which is considerably lower than the velocity limit flow of 0.3 cis. This low flow is probably due to insufficient pressure poppet opening.

Frequency response data was also taken with the same valve adjustments as above. Operation with sinusoidal inputs was accomplished as shown in the traces of Figure 34. The response curves of Figure 35 show a very low 90-degree lag point at 0.25 Hz at an amplitude change of -8 db. This low response is again attributed to the low fluid flow. Tests were also run on the effect of temperature on null bias; however, no conclusive results were obtained due to a transducer drift problem. Environmental testing was halted to try to improve the valve operation prior to continuing.

Valve adjustments were attempted to increase its flow and pressure output. Adjustment is very difficult due to the very small poppet displacement required for full output (0.0007 in.) Differential screws were designed for this adjustment; however, fabrication errors in the poppet sleeves prevented differential screw operation. During adjustment, one pressure sleeve press fit pulled apart, making further adjustment impossible and requiring poppet and sleeve disassembly. In repairing the poppet sleeve, both pressure poppet seats were increased in diameter (0.002 in.) to provide a pressure differential which will tend to produce a resultant pressure force to close the pressure poppets. The pressed-in poppet pegs were also replaced with free pegs to provide normal valve operation as initially designed.

Valve adjustment was again attempted and indicated centered sleeve position which shows that the pressure seat increased diameter did effect a pressure poppet positive closing condition. Proper adjustment could not be attained because of a sticky pressure poppet and a coarse sleeve adjustment. Further evaluation, including environmental testing, was halted, due to the inability to accomplish proper valve adjustment and the depletion of contract funds.

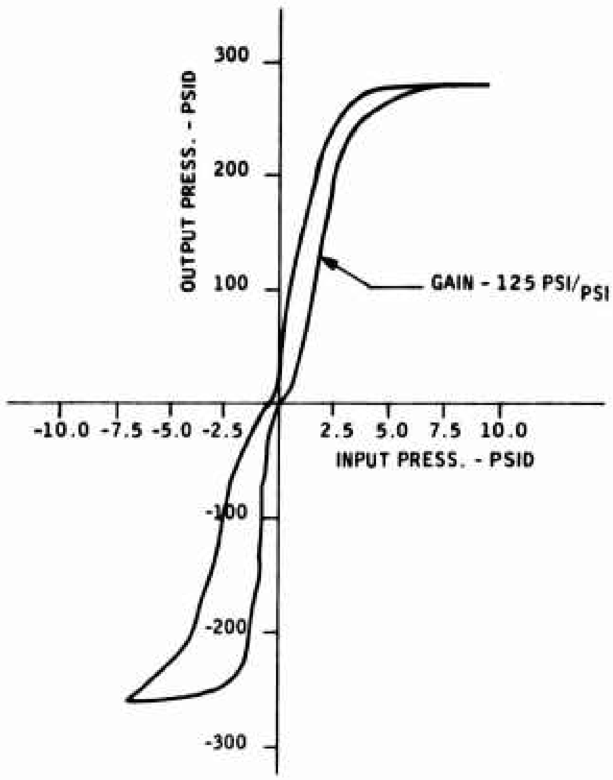

Figure 31. Servovalve Pressure Gain, 110°F

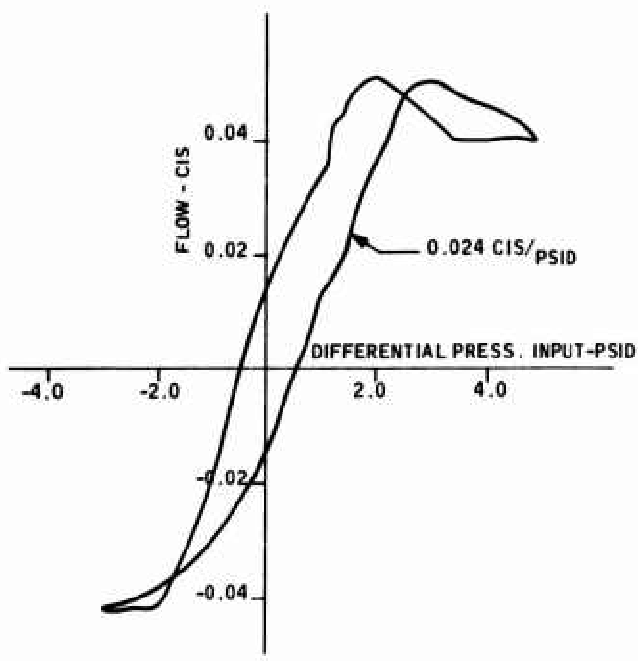

Figure 32. Servovalve Flow Gain, 110°F

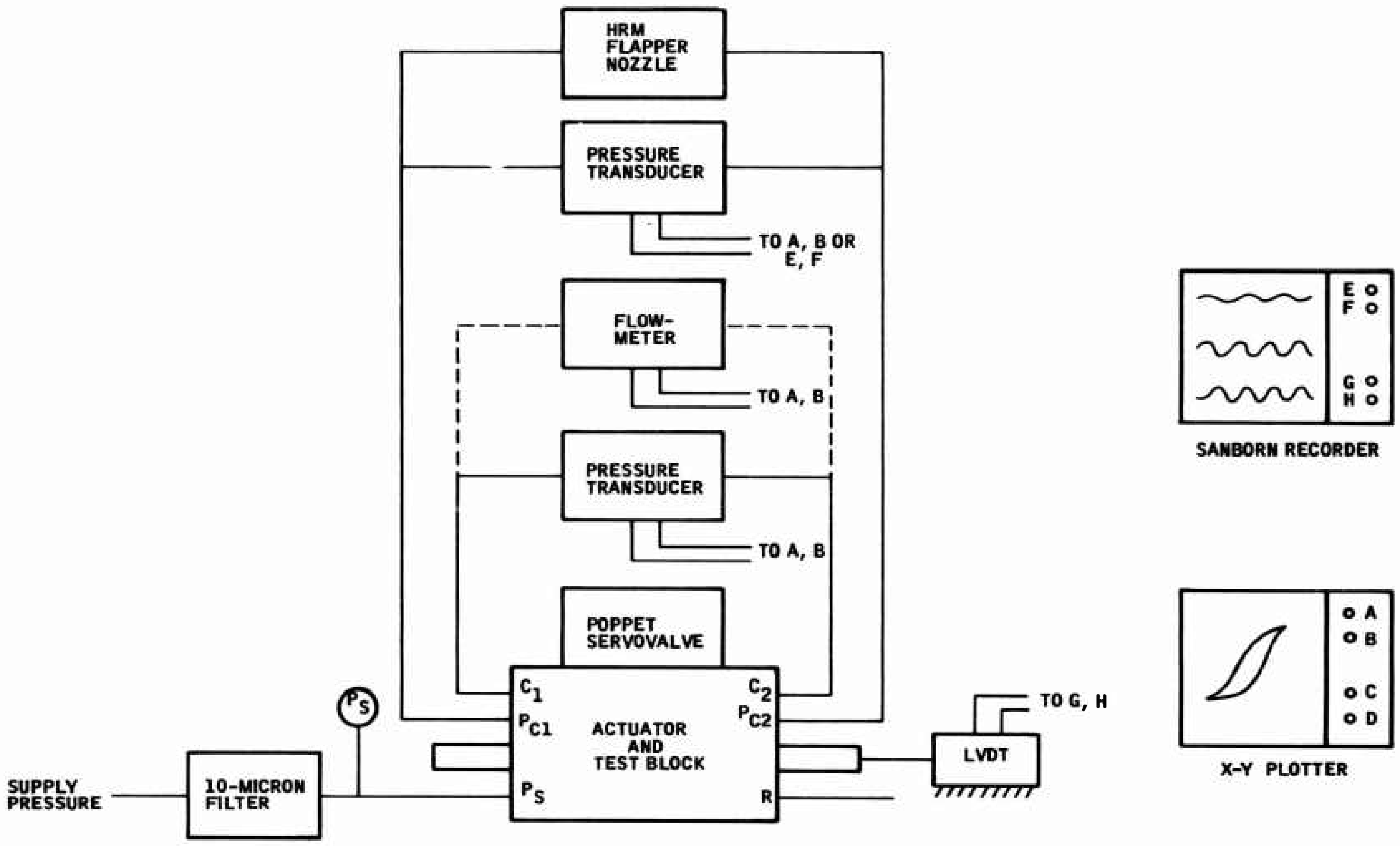

Figure 33. Test Schematic

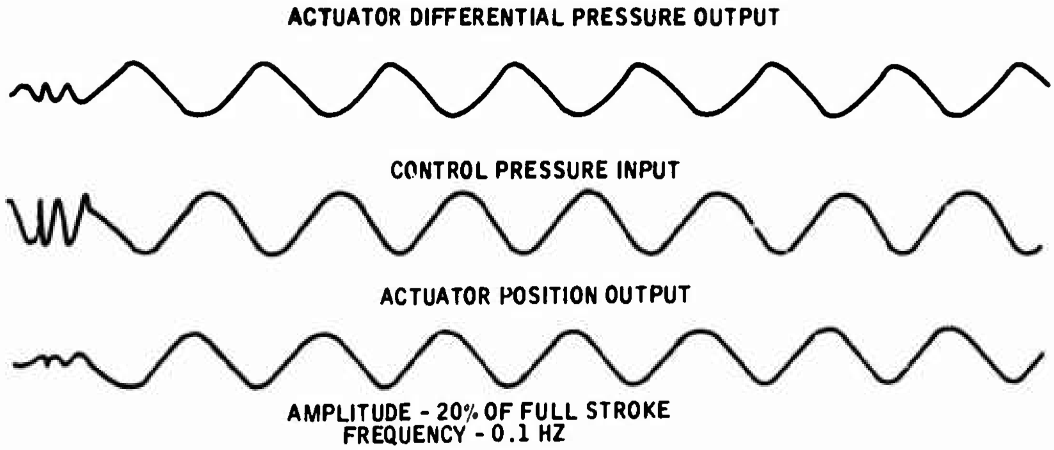

Figure 34. Frequency Response Tracer

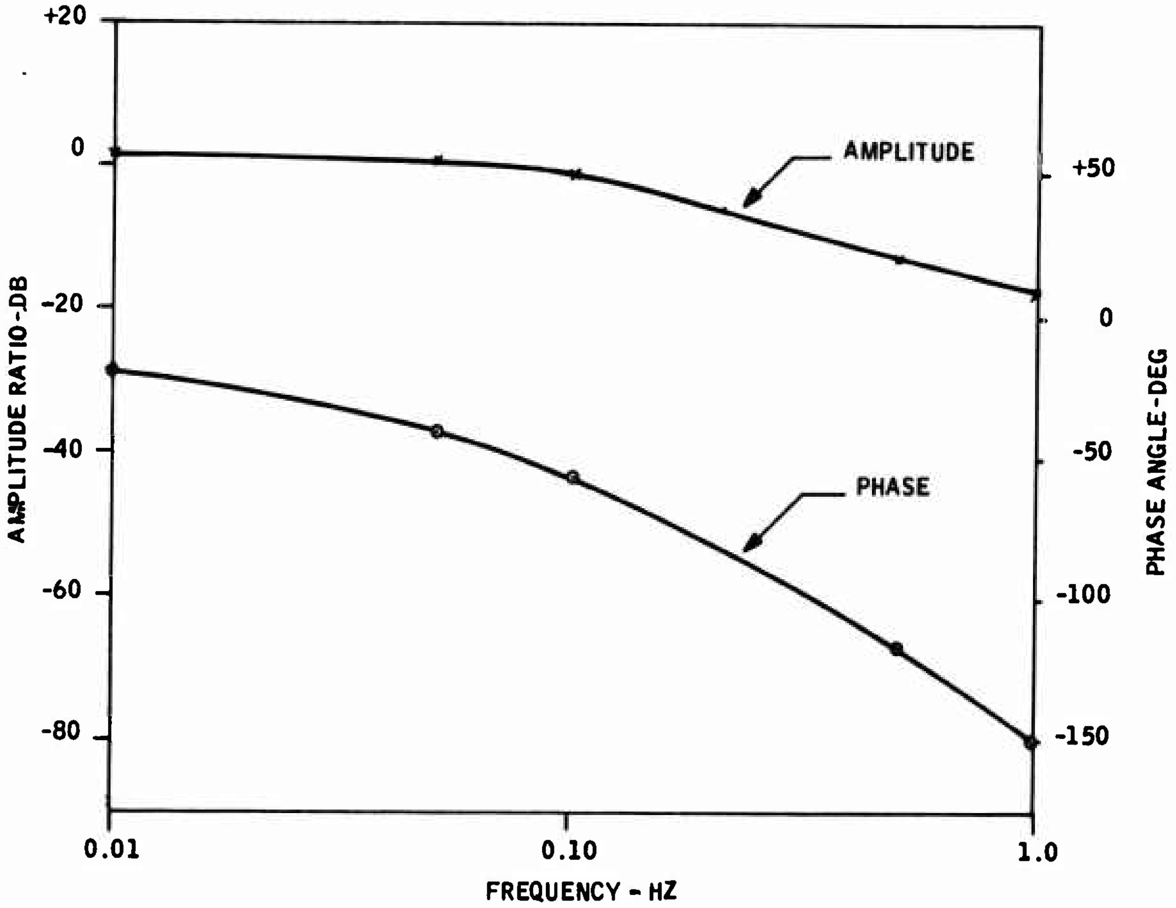

Figure 35. Frequency Response

CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

Although the program did not result in a successful operational poppet-type servovalve, it did demonstrate the basic capability of servovalve operation with the following conclusions:

- Poppet-valving can be mechanized to operate as a servovalve in driving an actuator piston.
- The design approach and application constraints required for this program [actuator interface mounting configuration and force summing (spring) feedback] severely limited the poppet-valve design configuration and valve operating forces.
- Individual poppet-sleeve adjustment is a very critical factor in poppet-type servovalve applications.
- Poppet-type valves can control low flow rates and high pressure gains.
- The program results did not prove or disprove the poppet valve claimed attributes of improved reliability and lower costs, but the unit tested did show a considerable reduction in size with respect to the present hydrofluidic servovalves.

RECOMMENDATIONS

As a result of the program effort, the following recommendations are made:

- Continue effort on poppet-type servovalve development to optimize poppet geometry; to improve mechanization configuration (adjustment, feedback, etc.), to determine more effectively the capabilities of poppet valving to servovalve applications;and to verify the claimed attributes of improved performance, reliability, and lower cost.
- Future poppet-type servovalve studies should include the following:
  1) Additional effort in flow force compensation and poppet angle optimization.
  2) Means of adjusting poppet position to within 5 percent of maximum operating travel.
  3) Fabrication of poppets with optimized geometry in the 0.0001-sq.-in. size range (less than 0.020 in dia) for increased poppet travel and improved adjustment.

4) Future configurations should use fluidic-type positional feedback systems for the following reasons:

   a) Arrangement of valve elements would be less restrictive, so alternate adjustment means would be possible.

   b) Available poppet operating force could be increased significantly with the present driving amplifier.

   c) Overall size could be reduced.

APPENDIX A

HYDROFLUIDIC POPPET - TYPE SERVOVALVE SPECIFICATION

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 poppet-type valving in place of the typical spool valve configuration. Mounting configuration and interfacing shall be compatible with the servoactuator used on Contract DAAJ02-72-C-0051, or similar.

1.1 Classification

The servovalve described herein shall be classified as experimental. Performance and configuration requirements shall be design goals and compatible with the servoactuator requirements of Contract DAAJ02-72-C-0051.

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 superseding 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	Environmental Test Methods

3.0 REQUIREMENTS

3.1 Item Definition

The subject item shall be a single- or two-stage, four-way flow control servovalve that provides control flow (at constant load) which is proportional to the applied input signal pressure. The flow control to the actuator shall be of a poppet-valve configuration. Flow from the servovalve is used to drive a hydraulic actuator. Feedback shall be provided to the first-stage input from the second-stage valve and/or the actuator position.

3.1.1 Servovalve Block Diagram (Figure 36)

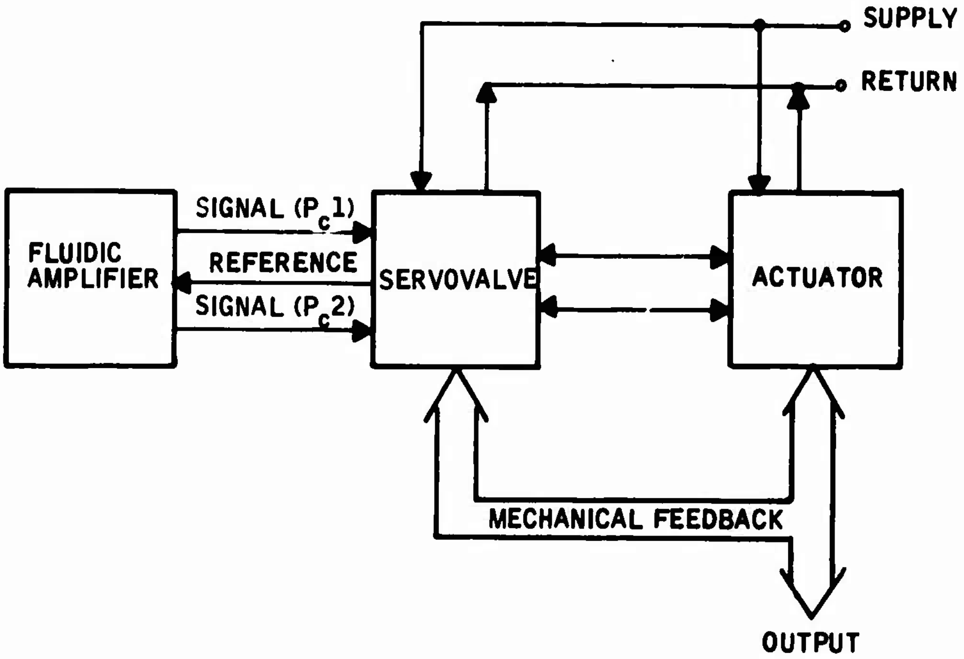

Figure 36. Servovalve Block Diagram

3.1.2 Servovalve Features and Physical Interface

The servovalve shall incorporate the following features and physical interface:

a) Housing.

b) Provisions for mounting to and interfacing with the servo-actuator assembly 3004830 used on Contract DAAJ02-72-C-0051 (Ref - Interface Dwg. SK 13998).

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°F 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	60 ± 3 psig
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 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 0.4 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 0.15 cis ±10 percent with ±2.0 psid signal pressure.

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 ±0.1 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 With Temperature Variation -- The null shift (from the null bias at standard test conditions) shall not exceed ±0.10 psid signal pressure over the temperature range of -65°F to 160°F.

3.2.1.3.3.2.2 With Supply Pressure Variation -- The null shift (from the null bias at standard test conditions) with a variation of supply pressure of ±20 percent shall not exceed ±0.02 psid signal pressure.

3.2.1.3.3.2.3 With Return Pressure Variation -- The null shift (from the null bias at standard test conditions) with a variation of return pressure of 50 psig shall not exceed ±0.020 psid.

3.2.1.3.3.2.4 With Signal Quiescent Pressure Level -- 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 0.02 psid per psid signal pressure between 4 psig to 6 psig.

3.2.1.3.4 Linearity -- All the gain curve test points shall fall within a ±0.10 psid band of the best straight line through these test points.

3.2.1.3.5 Threshold -- 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 percent, ±15 percent of supply pressure.

3.2.1.4 Dynamic Performance -- The nominal servovalve transfer function with signal input of ±0.20 ±0.04 psid shall be as shown below. The servovalve dynamic response shall be within the limits shown in Figure 37.

$$\frac{Q}{P} = \frac{0.15 \quad (125)^2}{S^2 + 2\,(0.9)\,(125)\,S + (125)^2} \quad \frac{\text{cis}}{\text{psid}}$$

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.

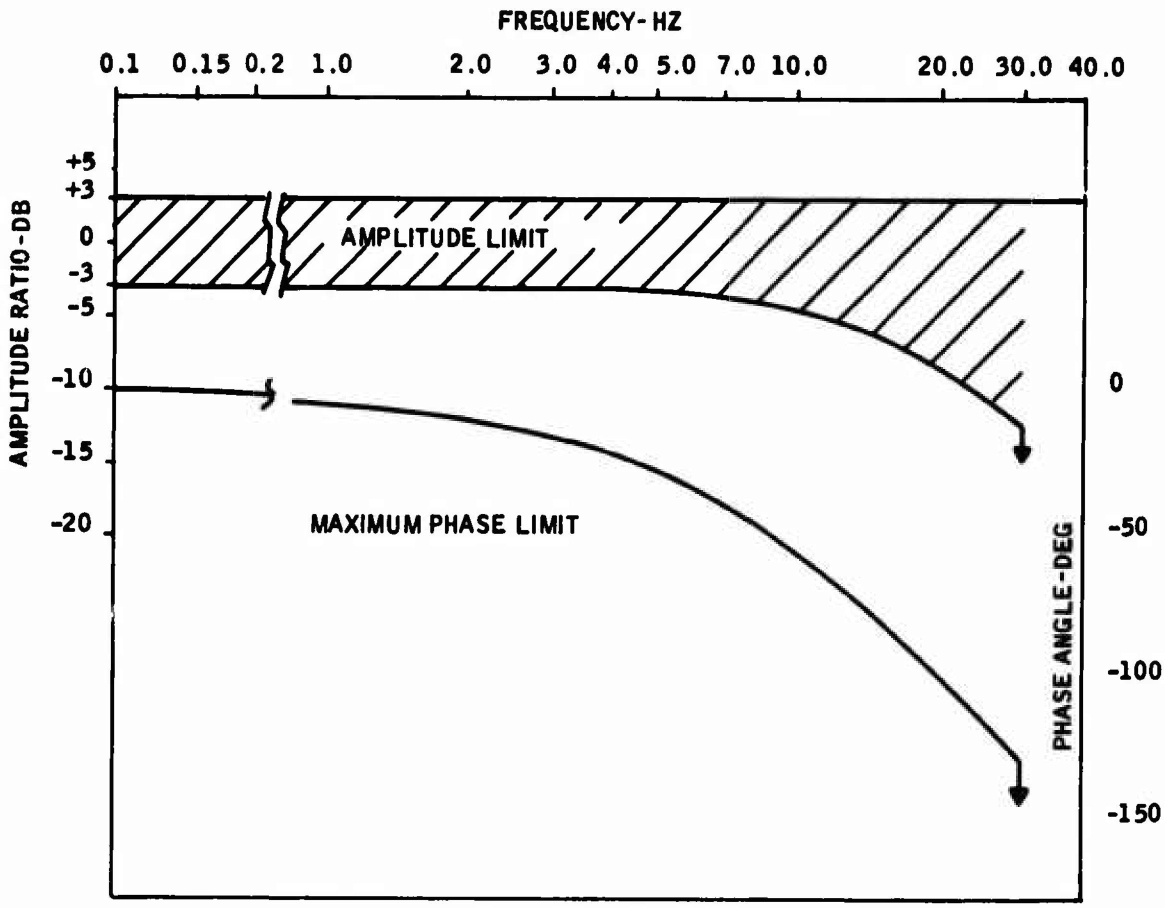

Figure 37. Servovalve Frequency Response

3.2.1.5.2 Linearity -- All test points shall fall within a ±0.015-inch band of the test 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.

3.2.1.5.6 Null Bias -- The null bias shall not exceed ±0.015 inch, with the input signal ports open to ambient pressure and completely full of oil.

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

- Temperature: ±0.015 inch (-20 to +160°F)
- Supply pressure: ±0.003 inch (±120 psig)
- Return pressure: 0.06 inch (50 psig change)
- Quiescent signal pressure: 0.006 inch (4 to 6 psig)

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 38. The actuator-driven load shall consist of mass equal to 0.15 lb/sec^2/ft and friction equal to 2.0 lb. The driving device shall have an output impedance of 80 lb-sec/in.[5]

3.2.1.5.8.2 Null Hunting -- The hunting of the output shaft shall not exceed 0.018 inch.

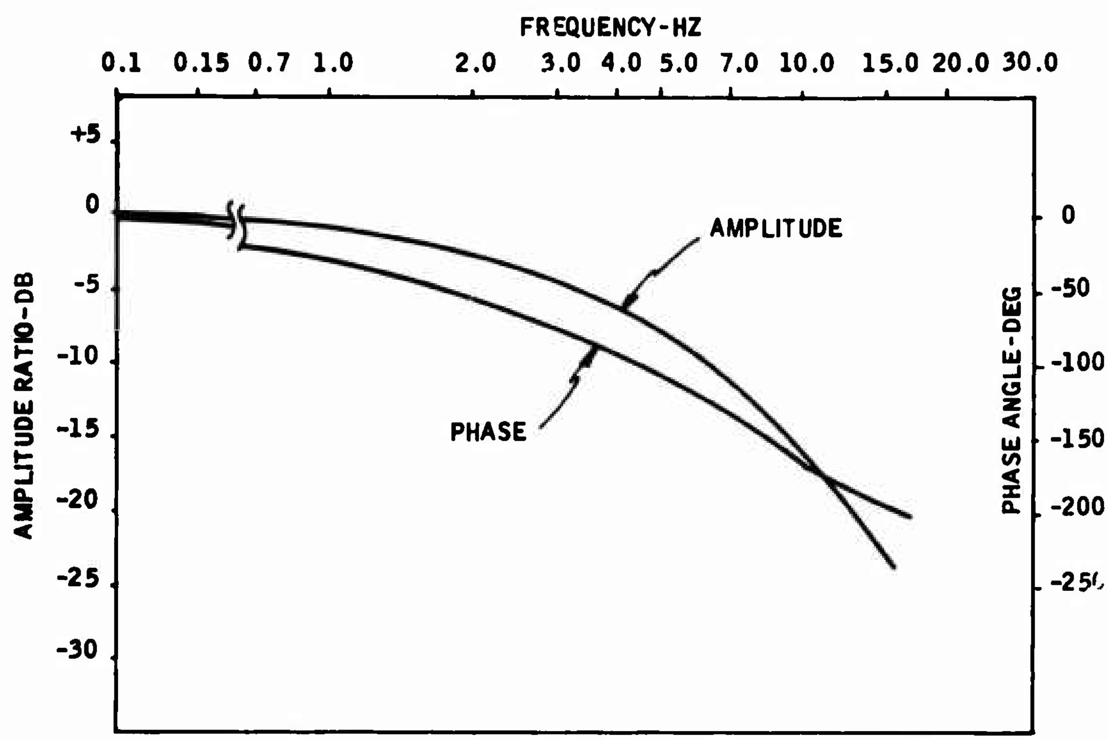

Figure 38. Servoactuator Frequency Response

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 the requirements of 3.2.1.5.7 and 3.2.1.5.8.2. From 0°F to 160°F the servoactuator shall meet the requirements of 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-I 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 DAAJ02-72-C-0051.

3.2.2.2 Proof Pressure -- The servovalve shall withstand the following proof pressures without evidence of external leakage or permanent performance degradation:

a) 2250 psi applied to the servovalve supply port with (1) the return port and load ports open, and (2) with the load ports blocked and 1500 psi maintained on the return port.

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) 3750 psi applied to the servovalve supply port with the cylinder ports blocked and the return port at 2250 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 wire shall be applied in accordance with MS33540.

3.2.2.7 Driving Fluidic Amplifier -- The output impedance of the driving amplifier shall be 80 lb-sec/in^5.

3.2.3 Environmental Requirements

3.2.3.1 Temperature -- The servovalve shall meet the requirements of 3.2.1 when subjected to temperatures from 0°F to 180°F. From -65°F to 0°F the servovalve shall function with degraded performance and shall meet the requirements of 3.2.1.2.1.

3.2.3.2 Vibration -- The servovalve shall perform satisfactorily when subjected to Procedure I, Curve B of Figure 514-I 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

APPENDIX B

HYDROFLUIDIC POPPET-TYPE SERVOVALVE TEST PLAN

1.0 SCOPE

This test plan outlines the procedures to be followed for conducting the performance tests on a hydrofluidic poppet servovalve. 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 DAAJ02-72-C-0051 (or equivalent) modified with the poppet servovalve.

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 3004830 used on Contract DAAJ02-72-C-0051 (or equivalent) 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 input signal.

5.0 PERFORMANCE REQUIREMENTS

The performance requirements shall be used as design goals for the servovalve specification of Appendix A; 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 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

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	60 ± 3 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 A) 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 Bias (3.2.1.3.3.1)

Null Shift (3.2.1.3.3.2)

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 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 Servovalve (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 (specification paragraph 3.2.1.5.7) and null hunting (specification paragraph 3.2.1.5.8.2).

With the system deenergized, stabilize the ambient temperature at 0°F. Energize and record servoactuator performance (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 full of oil and shorted together. 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 specification 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 (paragraph 7.11).

LIST OF SYMBOLS

A_b	bellows area (in.^{2})
F_v	poppet valve force (lb)
K_b	bellows spring rate (lb/in.)
K_f	feedback spring rate (lb/in.)
L	feedback pivot arm (in.)
M	bellows pivot arm (in.)
P_ℓ	bellows ΔP pressure input for maximum velocity (psi)
P_m	maximum ΔP pressure input (psi)
X_f	feedback spring deflection (in.)
X_ℓ	bellows deflection for maximum velocity (in.)
X_v	poppet displacement for maximum velocity (in.)