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APPLICATION OF FLUIDIC CONCEPTS TO HYDRAULIC CONTROL SYSTEMS

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R. H. Fashbaugh and E. R. Durlak

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NAVAL CIVIL ENGINEERING LABORATORY
Port Hueneme, California 93043

APPLICATION OF FLUIDIC CONCEPTS TO HYDRAULIC CONTROL SYSTEMS

Technical Note N-1255

ZF61-512-001-046

by

R. H. Fashbaugh and E. R. Durlak

ABSTRACT

A survey is presented of the experimental work that has been done in the area of hydraulic fluidics. The importance of Reynolds number is shown when comparing the performance of a fluidic element with hydraulic oil or air as the working fluid. The results of testing of all-hydraulic fluidic actuator control systems conducted at General Electric Company and at the Naval Ship Research and Development Center support the premise that hydraulic fluidic elements are useful in improving all-hydraulic servo-actuator controls. A small scale test conducted at NCEL with pneumatic fluidic elements showed that a pneumatic actuator can be powered directly by a fluidic amplifier cascade at the lower power levels.

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INTRODUCTION

This report presents the interim results of a study concerned with the application of fluidic concepts in actuator control systems in place of the conventional electro-hydraulic or electro-pneumatic systems. Fluidics has much to offer in control systems applications. The elimination of moving parts in the low-power section of a system is a natural way to improve reliability, especially in systems in which a better capability for withstanding severe vibration is needed. The fluidic components offer unique capabilities in explosive environments where electrical controls would be hazardous due to sparking, electromagnetic radiation, or shock. Finally, in either a hydraulic or a pneumatic system, since the hydraulic or pneumatic supply is already available, why not use it to power the control system through use of fluidics as well as to power the actuator?

This report presents a survey of the approaches for applying fluidic controls to hydraulic actuators that have been successfully demonstrated both on production machinery and on an experimental basis. In these investigations both fluidic proportional amplifiers and fluidic vortex valves were utilized. Also presented are the results of a small scale test conducted at NCEL to demonstrate control of a pneumatic actuator by fluidic proportional amplifiers. In addition, a proposed test configuration for testing fluidic control of hydraulic actuators is described.

BACKGROUND IN ACTUATOR CONTROL USING FLUIDICS

Both pneumatic and hydraulic fluidic components have been utilized in the control systems associated with pneumatic or hydraulic actuators. Pneumatic components have been used to a far greater extent than hydraulic components since fluidic technology development has almost entirely been restricted to pneumatic devices. Recently, however, several very successful experimental projects have been undertaken which were concerned principally with the use of hydraulic fluidic components in control systems. The following will be mainly concerned with these developments in hydraulic fluidic controls but a brief description of some fluidic pneumatic controls that are used commercially will be given also.

Pneumatic Fluidic Control

Many control systems involving pneumatic fluidic components have been developed. The majority of these systems are in practical use in the automatic control of production machinery. Typical applications are fluidic controls for a capstan lathe and for a pneumatic press. An example of a capstan lathe that is equipped with a pneumatic fluidic control system is illustrated in Reference 1. In this system a low-pressure pneumatic fluidic circuit receives programmed instructions from a punched card to provide automatic operation of functions previously performed manually. All approach and return movements of the turret and cross-slide are affected at a fixed speed and feed movements are made at a variable speed by hydraulic actuators. The collet of the machine is opened and closed by a pneumatic actuator controlled by fluidic circuits and actuators also control spindle speed changes. On an average cycle the fluidic controls and associated actuators use about 5 cfm of air at an operative air pressure of 60 psi. An example of fluidic controls for a pneumatic press are explained in References 2 and 3. In this application the fluid control system replaces a cam or drum-type timer and associated electrical limit switches and electric solenoid valves. In addition to having a long life and high reliability, the fluidic control in this application has proven to be flexible in control of the press pneumatic actuator and also compatible with computer or centralized control. In a total cycle of operation the fluidic control performs six basic functions which are: cutting and receiving material, conditioning material, forming material, conditioning formed parts, transferring formed part, and conditioning the mold. This illustrates that the actuator control system must be capable of time-based sequential control which is readily attained by use of a fluidic oscillator of fixed frequency and fluidic binary counters. To complete the system the time-based sequential control output affects a bistable fluidic amplifier which controls the actuator pilot valve. The complete system is operated by normal line air pressure of 75 psi. These applications illustrate some very important characteristics of fluidic controls aside from reliability and low cost; they are not only capable of powering a pilot valve and in some cases powering the actuator directly but are also especially appropriate for control systems where a memory capability is required for sequencing of operations or functions.

Hydraulic Fluidic Control

Many of the opportunities for using hydraulic fluidics occur in control systems utilizing hydraulic actuators for the final control element. It is possible to use pneumatic fluidics to control a hydraulic final actuator by means of the several hydraulic/pneumatic interface valves available commercially and this has been done. However, in most applications considerable simplification and cost saving can be achieved if the same fluid is used in the control system and the actuator. Many of the basic concepts of fluidics are equally applicable to air and hydraulic oil (such as MIL-H-5606) operation, but there are important differences between the two modes of operation due to differences in fluid properties. These differences will be summarized prior to discussing experimental investigations concerned with hydraulic fluidic control systems.

Fluidic element performance comparison-air versus hydraulic oil.

In comparing the performance of a given fluidic amplifier with different working fluids, as one might expect, the most important parameter to consider is the Reynolds number based on the nozzle flow.[4] To obtain an equivalent Reynolds number when operating with oil as with air a considerably higher supply pressure is required. For example, for a given nozzle size an oil supply pressure of 375 psig at 120°F will yield the same flow Reynolds number as when operating with air at a supply pressure of 1 psig at 70°F. This follows from the definition of Reynolds number

$$N_R = \frac{\rho D_e V}{\mu}, \tag{1}$$

and the square-law relationship for nozzle flow

$$V = \sqrt{2g \frac{\Delta P}{\rho}} . \tag{2}$$

In these relations, ρ is the mass density, μ the viscosity, g the acceleration due to gravity and ΔP the nozzle pressure differential. The equivalent diameter D_e is defined in the usual manner, with A and P the nozzle area and wetted perimeter respectively, as

$$D_e = \frac{4A}{P} . \tag{3}$$

A comparison of the variation of Reynolds number with supply pressure for air and for MIL-H-5606 oil (from Reference 4) is shown in Figure 1. This comparison clearly illustrates the much higher supply pressures required, for a given fluidic amplifier, for operation with hydraulic oil as compared to operation with air. From relations (1) and (2) the observation can be made that the Reynolds number is related to supply pressure by the fluid property

$$C_R = \frac{\mu}{\sqrt{\rho}} \tag{4}$$

where C_R is termed the Reynolds Coefficient. For a given amplifier the supply pressure required varies as the square of C_R. Some typical values for C_R for air, water, and oil at 40°C are given in Table 1.

Table 1. Reynolds Coefficient at 40°C

	C_R
Air	.0001457
Water	.000194
MIL-H-5606 oil	.00378
SAE-10 oil	.00920

Table 1 shows that the values for C_R for air and water are comparable and that therefore only slightly higher pressures are required to produce a given Reynolds number using water than would be required using air for a given fluidic element. This suggests that the available pneumatic fluidic elements would be useful for control systems (as well as other applications such as proximity sensing) in underwater applications where it might be desirable to use water as the working fluid. The values of C_R reflect the much higher pressures needed for oil as was pointed out previously.

To impart practical meaning to the discussion concerning Reynolds number it is appropriate to look at the dependence of gain of a typical fluidic proportional amplifier on Reynolds number. The gain as a function of Reynolds number is shown in Figure 2 for a General Electric amplifier.[4] Since Reynolds number is a measure of viscous losses it is expected that the performance of the element would degrade at the low Reynolds number. Figure 2 shows this to be true. A typical operating point is shown at a Reynolds number of 4000 which corresponds to a supply pressure of 500 psig. The nozzle dimension for the amplifier used to obtain the curve in Figure 2 was 0.02 inches by 0.02 inches and the curve was determined using MIL-H-5606 oil.

A comparison that is necessary in considering oil in place of air as the working fluid is the relative power consumption. This comparison is best made on the basis of Reynolds number. A curve which shows typical power consumptions for a fluidic amplifier[4] is given in Figure 3. The comparison shows considerably more power is needed for a given Reynolds number when using oil. Even though this is a disadvantage, it is not considered a serious one since the hydraulic fluidic system will usually be used in conjunction with a relatively high-powered hydraulic system where the power drain shown (80 watts at N_R of 4000) will not be significant.

The gain characteristics of a General Electric pneumatic amplifier comparing hydraulic and air performance were determined in the study reported in Reference 5. A part of the results of this study are presented in Figure 4 where the normalized output pressure change is plotted versus the normalized control pressure change. The supply pressure is used as the normalizing parameter. The value of the supply pressure when using oil was 500 psig which corresponds to a Reynolds number of 4000. The gain curves for air and oil are similar with a slightly higher gain when oil is used. The amplifier exhibits a good gain characteristic with oil as the working fluid when the supply pressure is determined by using the Reynolds Number as the criteria as previously discussed.

Another fluidic amplifier characteristic of importance is the dynamic response. Proportional amplifiers operating on hydraulic oil have the potential for quite high frequency response since hydraulic oil with low entrained air content does not suffer from the capacitive effects of air. The inertance effects, however, due to long thin passages can be quite large. The four major dynamic effects to account for when calculating frequency response are: control passage resistance and inertance, time delay between nozzle and receiver, vent region dynamics, and receiver resistance and inertance. With three stages in an amplifier gain block, Reference 4 shows that an open-loop gain of 500 is easily obtainable with a bandwidth of about 1000 rad/sec. The bandwidth (-45 deg. phase shift) of a single proportional amplifier is approximately 5000 rad/sec.

Experimental fluidic-hydraulic control systems

A survey has shown that all-hydraulic fluidic control systems are in the development stage and that the principal effort in this area apparently is being done at the Research and Development Center of the General Electric Company[6,7,8] and at the Naval Ship Research and Development Center.[9] Although fluidic elements which were designed specifically for hydraulic use are not available commercially, work has been progressing on development of both digital and proportional hydraulic fluidic components.[6] However, as shown previously some pneumatic fluidic elements can be used as hydraulic elements by use of proper supply pressure.

The results of an all-hydraulic fluidically controlled servoactuator test is reported in Reference 6. A block diagram of the test configuration is shown in Figure 6. The fluidic amplifier which was used for this investigation was the element reported in Reference 4. The servoactuator utilized 3000 psi oil (SAE 5W-20) in the output stage and developed a force of 50,000 pounds at stalled conditions. Closed loop control was obtained by using position sensors at the operator's input and on the power actuator's output, as shown in Figure 5. The outputs of the two sensors are compared in the hydraulic fluidic operational amplifier which produces a hydraulic pressure signal to the servo valve which controls the actuator. The servo valve was a spool-type valve which was designed for a maximum flow of 25 gpm with a 1000 psi valve pressure drop. The valve maximum input control pressure required was $\pm$ 100 psid. In this application the fluidic amplifier was a three stage amplifier with a supply pressure of 200 psig and an open loop amplifier gain of approximately 500. The system was designed to maintain the output position of the 16-inch stroke power actuator to within 0.060 inches for load changes of 30,000 pounds. Actual values obtained from tests showed a maximum deviation of 0.030 inches. Experience with this experimental system gave valuable insight into trade-offs involved in applying hydraulic fluids. Since internally generated noise in the system increases with an increase in Reynolds number a trade-off must be made between gain and noise in selecting system supply pressure and temperature. In this particular application a uniform oil temperature must be maintained throughout the system to assure accurate tracking of the command and feedback position sensors since the flow in the sensors is laminar and hence sensitive to viscosity changes. High flow rates through the sensors are therefore desirable to provide adequate circulation to assure uniform temperature. The result is a high noise level which limits the allowable amplifier gain but a high gain is desirable to overcome servo valve hysteresis which was appreciable in this case. Through consideration of the trade-offs mentioned, a closed loop system gain of 15 was chosen for the operational amplifier at an operating oil temperature of 140°F. Due to Reynolds number effects the gain dropped to 8 at 110°F and reached unity at about 80°F. A promising method to avoid this temperature sensitive problem is to use water or a water-base fluid as the working fluid. Reference 4 showed water to come close to air in terms of the relatively low pressures and power levels required to attain a given Reynolds number.

An all-hydraulic fluidic control system designed for submarine steering and diving controls is described in Reference 9. This control system in concept is similar to the all-hydraulic fluidic system reported in Reference 6 (Figure 5). The position sensors in this system were specifically designed for the system but the spool valve and actuator were standard items. General Electric fluidic amplifiers were used in a two stage block with a supply pressure of 500 psi. A prototype of the submarine control system was tested on a bench-test basis to determine if the system met the design objectives. Hydrodynamic loads as well as control surface inertia loads were simulated. The actuator supply pressure was 2800 psi with a system inlet oil temperature of 110°F. The test results showed that the system performance was very close to that predicted. In general, the position accuracy of the control surface (measured at the zero position) was within $\pm$ 1/4 deg. The fluidic control system was considered to be a major improvement over the original all-hydraulic (non-fluidic or conventional) control system.

Another concept of all-hydraulic fluidic actuator control is reported in References 7 and 8. This development was done for the U. S. Army Aviation Material Laboratories by General Electric's Specialty Fluidics Operation. In this concept the conventional spool servo-valve is replaced by a fluidic vortex valve as shown schematically in Figure 6. The servoactuator was designed for use in a helicopter flight control system. MIL-H-5606 oil was used for the working fluid at a system supply pressure of 1500 psig. The vortex valve servoactuator was designed for a 100 pound load capability with a maximum quiesent flow of 0.8 gpm. The design dynamic response was 10 Hz. Prototype servoactuator units were manufactured and subjected to acceptance tests. Based on the test results it was concluded that the fluidic servoactuators meet the requirements of a helicopter flight control system and that the performance of the servoactuator is predictable by design analysis.

SMALL SCALE FLUIDIC CONTROL TEST

Description and Purpose of Test

The purpose of the test was to demonstrate the ability of fluidic components to control the speed and position of a pneumatic actuator. This was the first in a series of experiments designed to develop hydraulic actuator fluidic controls for Naval applications.

The test was designed to use two fluidic proportional amplifiers of the type shown in Figure 7. In Figure 7 the amplifier is shown with one control input signal. As the other control input signal is increased, the output flow divides proportionally between the two output ports. These amplifiers are classified as load insensitive. That is, increasing the load on one output does not increase or change the flow through the opposite output. As the load on one output is increased, the supply flow will vent through the "center dump" rather than out the other output. The amplifiers were cascaded as shown in Figure 8 such that the output of the first amplifier (S_1) is the control input to the second amplifier (S_2). The control input of S_1 is the pressure difference provided by the cone jet sensors which are used to detect the position of the control arm. The sensor pressure difference (control input to S_1) is then amplified in a proportional manner by S_1 and S_2 to a level sufficient to drive the pneumatic actuator.

Due to time limitations, all testing had to be conducted with in-house fluidic components. Since only pneumatic components were available the small scale pneumatic test setup was used to gain knowledge in fluidic controls that would be useful in conducting a full scale hydraulic test. The fluidic amplifiers were components of a Fluidonics circuit panel. The control arm was fabricated from stock aluminum material, and the load on the piston was simulated by a small bucket containing lead pellets. The position and speed of the piston is adjusted by movement of the control arm. The test setup is shown in Figure 9.

The circuit in Figure 8 has no feedback loop and is therefore an open-ended system. More precise control could be maintained with a closed loop system and this would be an area to be investigated in a hydraulic control system test.

Test Results

The test results are presented in Table 2. These results represent the best adjustment of system pressures at maximum load. That is, various combinations of supply and sensor pressures were tried to give the highest output pressure difference on S_2. It is this output that is used to drive the pneumatic actuator.

The control of the actuator was good. At the slower piston speeds the movement tended to be jerky due to the relatively large static friction forces (about 20% of the load). Otherwise the movement of the piston was smooth.

The maximum load that could be lifted was 5.6 lbs. However, the maximum useful working load was 4.75 lbs, with a fluidic amplifier output pressure of 16 psig. It follows that the effective actuator piston area was approximately,

$$\text{Area} = \frac{\text{load}}{\text{pressure}} = \frac{4.75 \text{ lb}}{16 \text{ lb/in}^2} = 0.30 \text{ in}^2$$

Table 2. Test Results

	S_1 Amplifier	S_2 Amplifier
Supply Pressure (psig)	21	39
Control Pressures (psig)	0.6 , 0	4 , 0
Output Pressures (psig)	4 , 0	19 , 1

Sensor Input Pressure - 10 psig

From Table 2 it can be seen that the pressure gain is about 30 across the amplifier cascade. The S_2 amplifier output pressure varied slightly with actuator load. The maximum speed of the piston, however, was fairly sensitive to actuator load.

This test represents primarily a demonstration of the principle of fluidic control of an actuator using pneumatic components. The system used, however, can be applied to control systems using hydraulic oil as the working fluid. As pointed out in the Background Section, there has been a substantial amount of experimental work done in the area of hydraulic fluidics. Using information from this experimental work and the results of the test described herein, an experiment, which is presented in the next section, was designed to test a fluidic control system which uses hydraulic fluid to control a hydraulic actuator. This system could have potential application to Navy systems in which hydraulic actuators are used.

DESIGN OF HYDRAULIC FLUIDIC CONTROL SYSTEM TEST

The proposed hydraulic fluidic control system test is similar in concept to the test conducted with pneumatic components. The use of hydraulic oil requires a different type of hardware to be used. High pressure lines must be used in place of plastic tubing. The control sensor must be modified to work with hydraulic oil and the fluidic components must be able to withstand high pressures ($\approx$ 500 psig).

The schematic of a proposed experiment is shown in Figure 10. Due to the lack of available off-the-shelf hydraulic fluidic amplifiers, it is proposed to use General Electric P4 pneumatic proportional amplifiers that are capable of withstanding high pressures. The control sensor is a modified jet-pipe hydraulic controller similar to that used by Singer Industrial Controls (formerly Askania Controls). Figure 10 illustrates the principle of this jet-pipe controller which is used in many control system applications.

Approximate operating pressures are given in Figure 10 which are based on the experimental investigations summarized in the Background Section. The spool valve and actuator are commercially available items. As can be seen by comparing Figures 8 and 10, the hydraulic system test uses the same basic fluidic circuit that was used in the pneumatic test. The circuit in Figure 8 was modified to use hydraulic oil as the working fluid but other fluids such as water could be used in tests concerned with potential underwater applications.

The manner in which the system of Figure 10 works is similar to that of Figure 8. The deflection of the jet-pipe control arm determines the pressure difference applied to the first amplifier. This pressure difference is amplified and applied to the input of the second amplifier where it is again amplified and then used to drive the spool valve. The spool valve moves in a proportional manner determined by the fluidic amplifier output. The spool valve controls the high pressure ($\approx$ 3000 psi) oil flow to the actuator. Therefore, the quantity and direction of oil flow to the actuator is controlled by the deflection of the jet-pipe controller.

The test procedure should consist of measuring the necessary system parameters to determine the servoactuator system dynamic characteristics and actuator positioning accuracy. The pertinent parameters are: jet-pipe displacement, actuator displacement, actuator force, pressure at jet-pipe inlet, amplifier supply control and outlet pressures, and spool valve inlet and outlet pressures and flowrates. Also required are the pressure-flow characteristics of the spool valve, the gain and pressure-flow characteristics of the fluidic amplifiers, the displacement-pressure characteristics of the jet-pipe controller, and the spool valve control pressure-displacement characteristics. These component characteristics are obtained from the manufacturers' data.

SUMMARY

1. Pneumatic fluidic servoactuator control systems are being used successfully to provide automatic control of a capstan lathe.

2. Integrated pneumatic fluidic circuits are being used to replace electrical controls on a pneumatic press which provide better performance and reduced need for maintenance.

3. Tests conducted at NCEL showed that for low power requirements, a pneumatic actuator can be powered directly by a pneumatic fluidic amplifier cascade. The fluidic amplifier cascade replaces the conventional servovalve.

4. Water is potentially a good working fluid for fluidic controls because of the relatively low pressures and quiescent power levels required as compared to oil.

5. Hydraulic oil operated fluidic amplifiers are comparable to pneumatic fluidic amplifiers with regard to gain characteristics, staging, and operation in circuits.

6. Tests conducted at other facilities show that all-hydraulic fluidic servoactuator control systems are practical, analytically predictable, and are an improvement over conventional all-hydraulic servoactuator control systems.

CONCLUSIONS

1. This investigation confirms the known fact that pneumatic and hydraulic fluidic control of actuators is practical and that these controls work well in severe environments such as high vibration, high temperature, or high explosive potential.

2. Replacement of conventional servovalves by fluidic components results in a simpler and potentially less expensive system with lower maintenance requirements.

3. A promising area of investigation is the use of water in fluidic servoactuator control systems for undersea applications.

RECOMMENDATIONS

1. Testing of the fluidic control system shown in Figure 10 should be conducted to provide improved all-hydraulic servoactuator control system techniques for Navy applications.

2. Development of a fluidic servoactuator control system which uses water for the working fluid is recommended for undersea applications.

3. Testing of fluidic actuator controls should be conducted because fluidics possesses a good potential for advancing the state-of-the-art of servoactuator control systems.

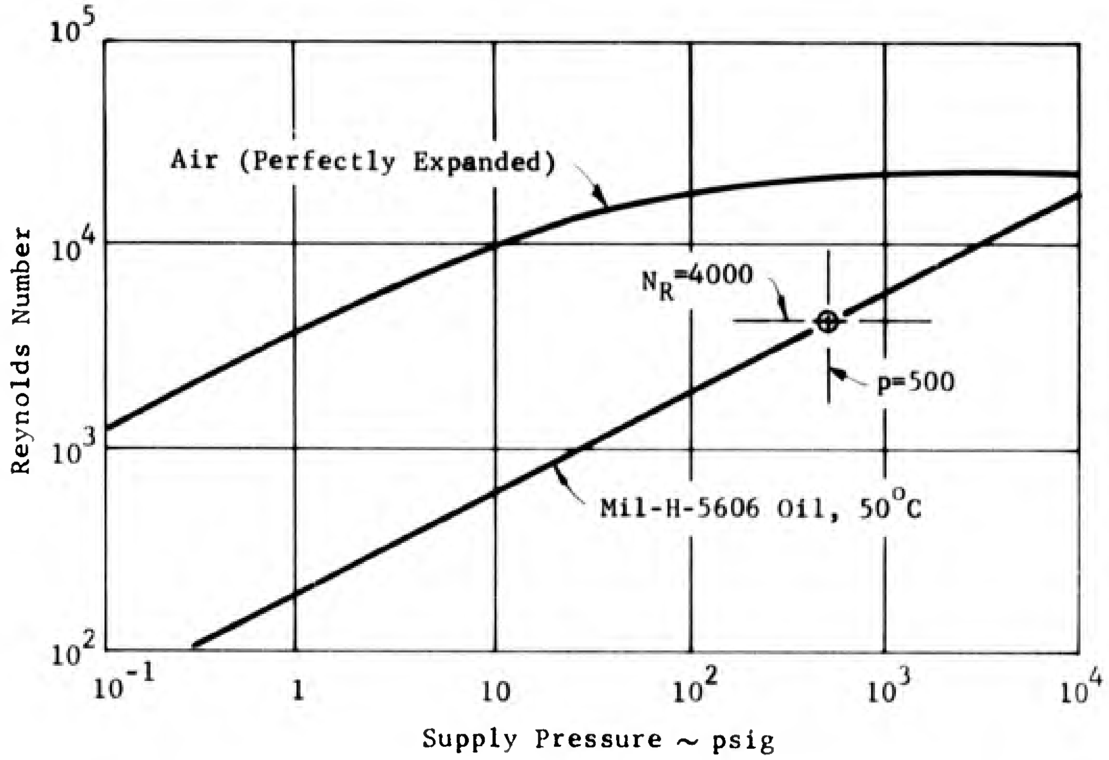

Figure 1. Reynolds Number Change With Supply Pressure - Air and Oil (Reference 4)

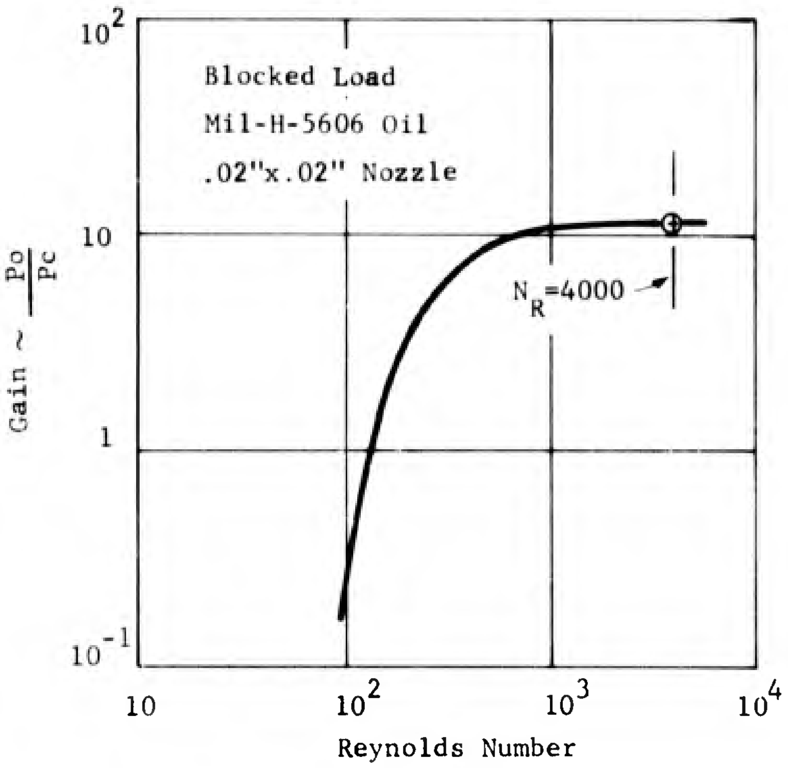

Figure 2. Proportional Amplifier Gain versus Reynolds Number (Reference 4)

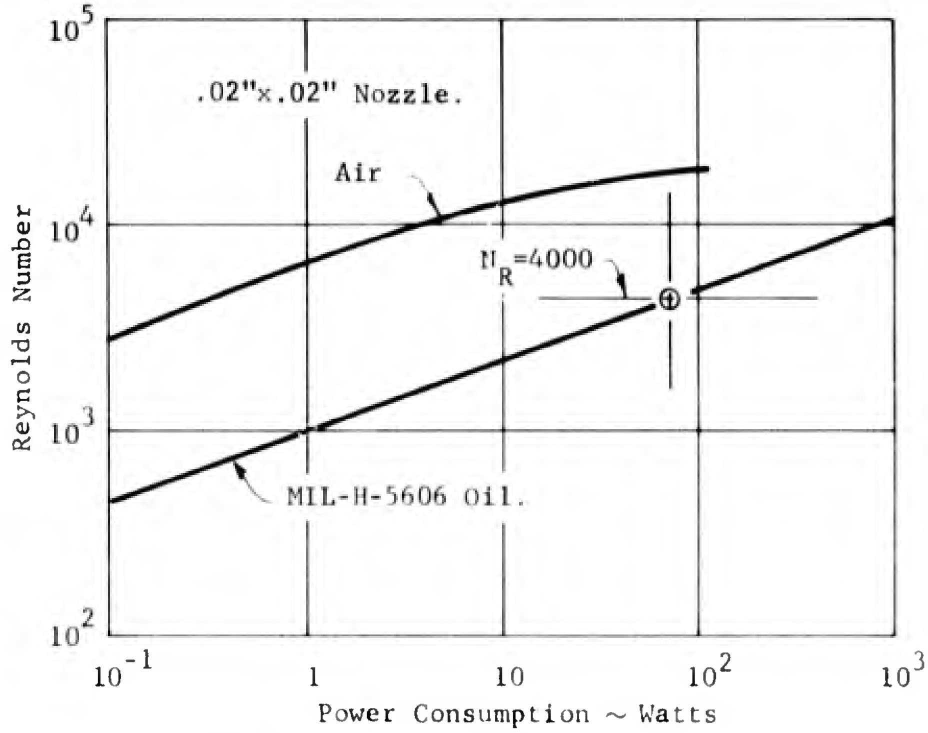

Figure 3. Power Consumption Change With Reynolds Number - Air and Oil (Reference 4)

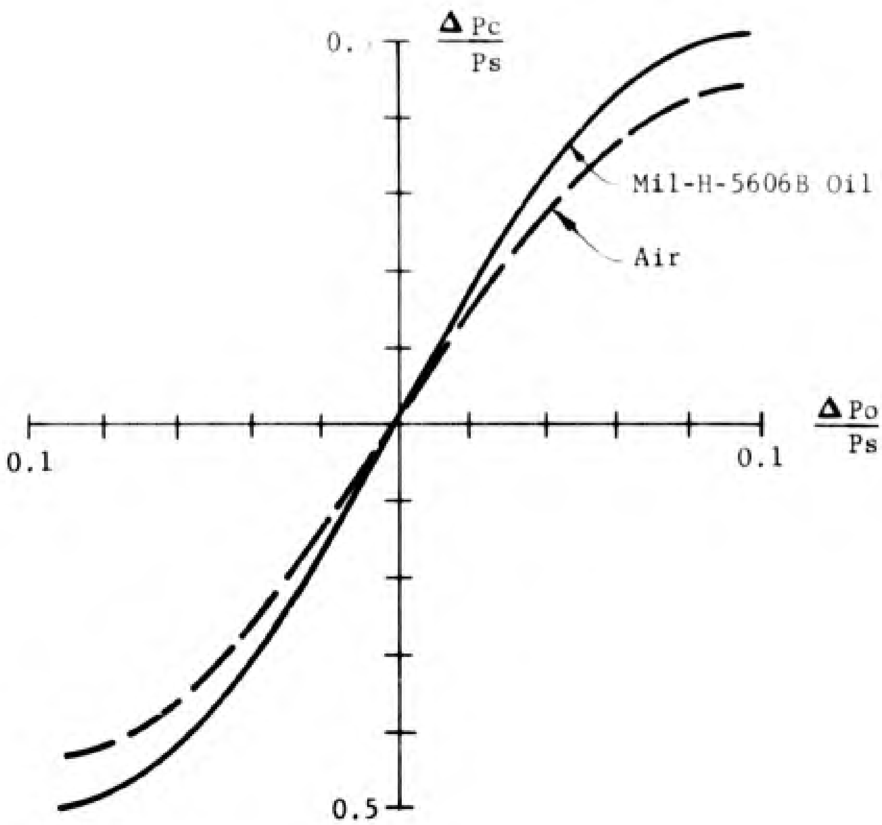

Figure 4. Pressure Gain Comparison - Air and Oil Operation (Reference 5)

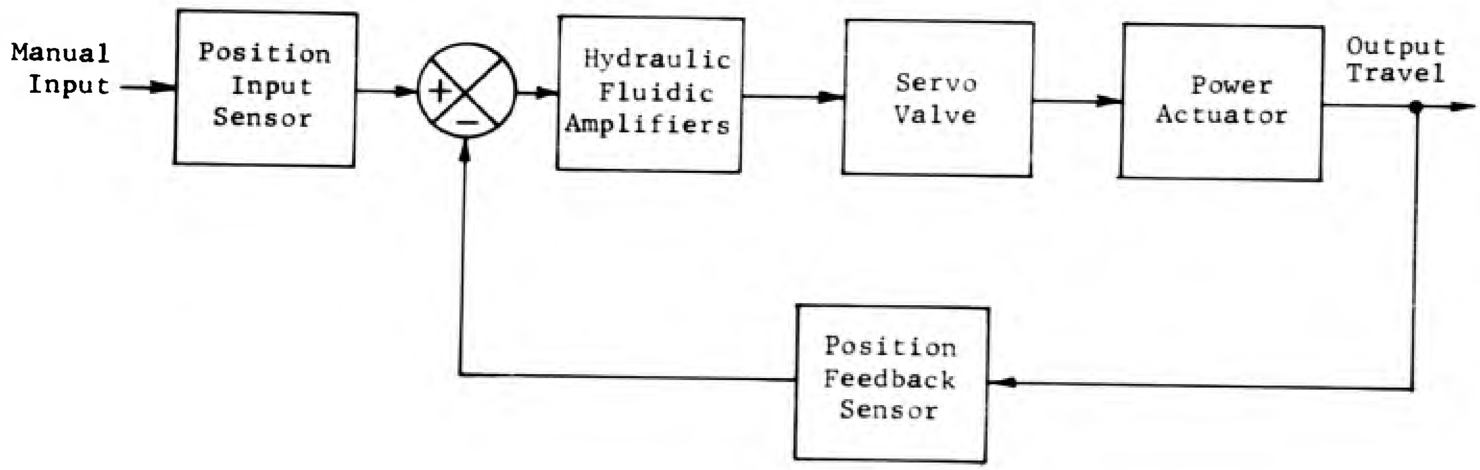

Figure 5. Block Diagram, All-Hydraulic Fluidically Controlled Servo Actuator (Reference 6)

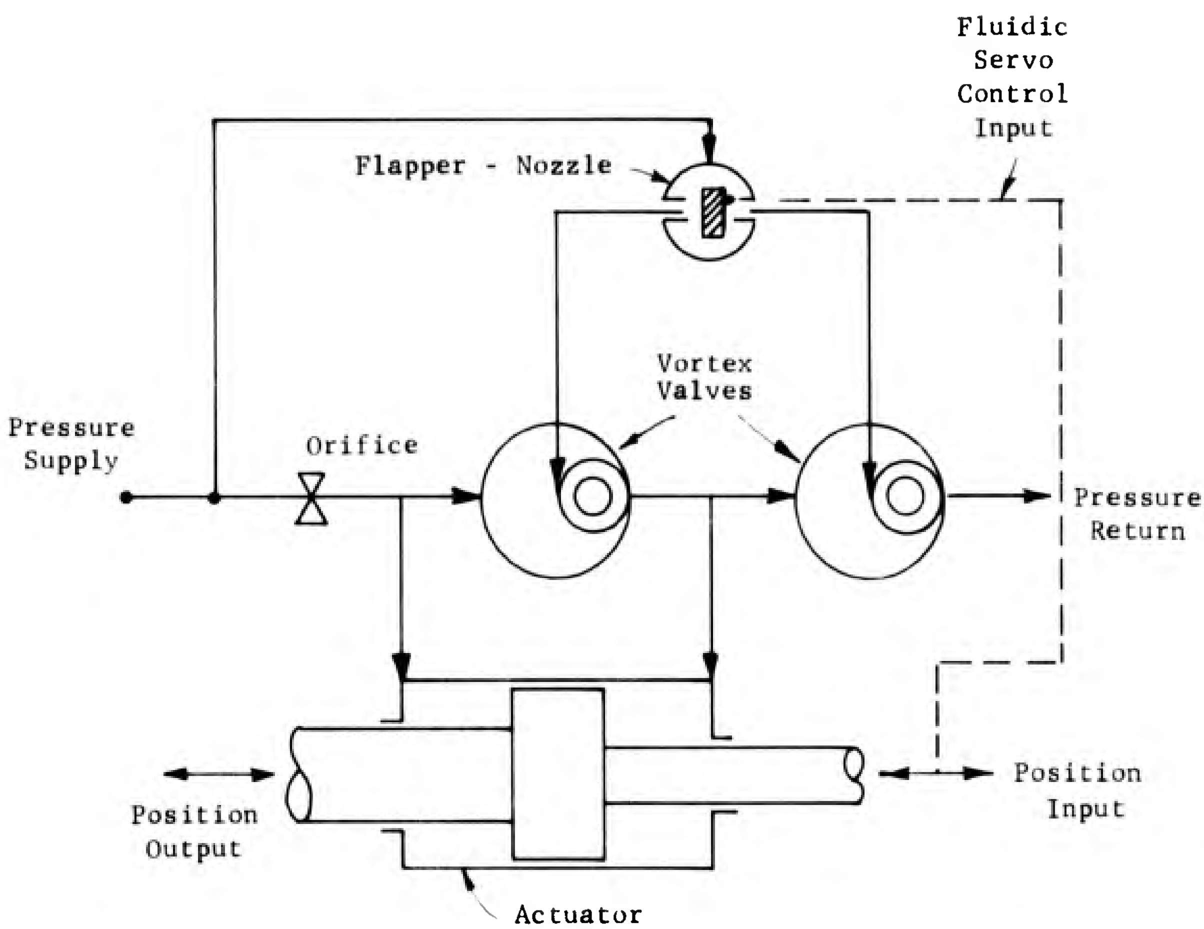

Figure 6. Schematic - Fluidic Vortex Valve Servoactuator (Reference 7)

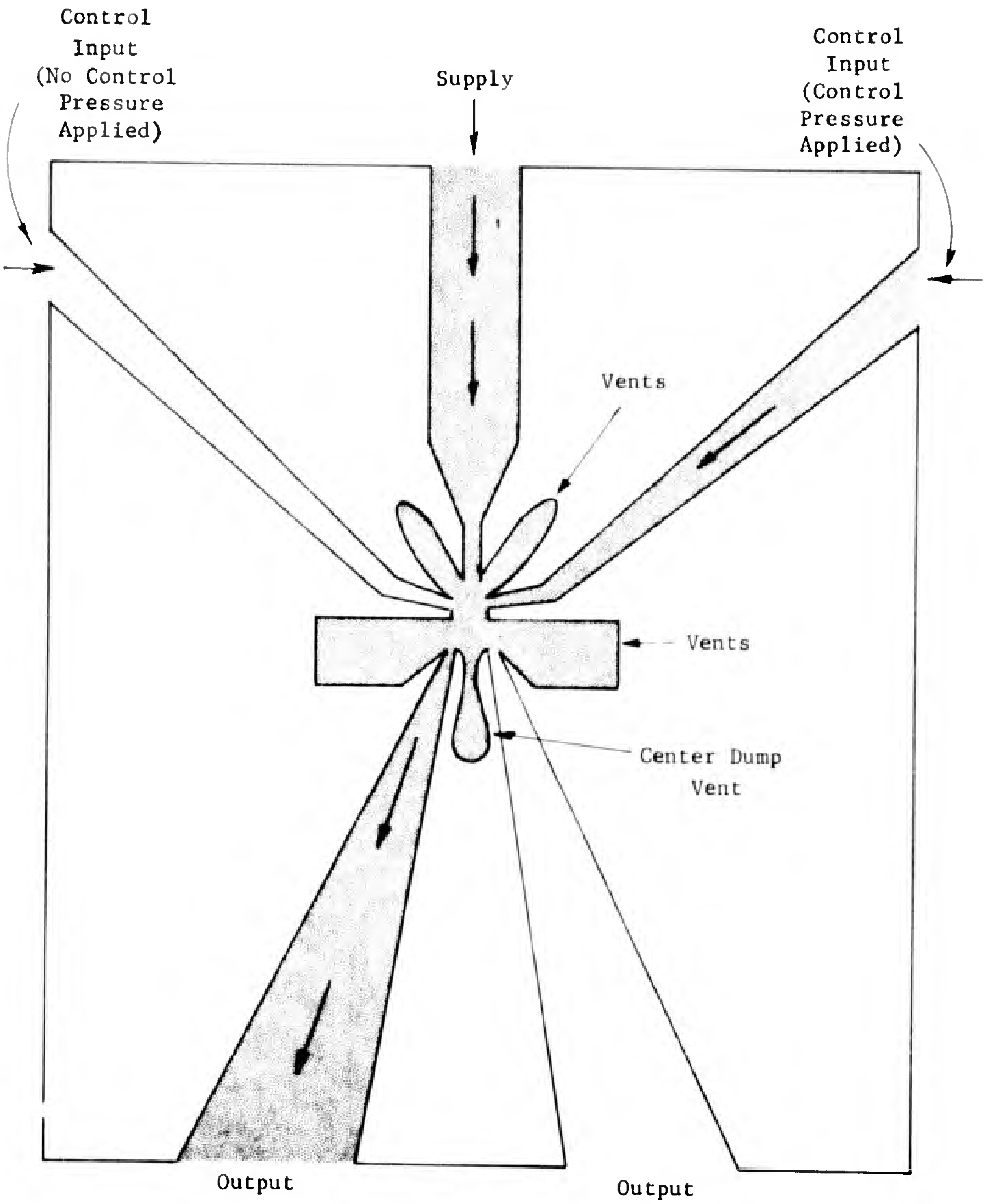

Figure 7. Fluidic Proportional Amplifier Schematic

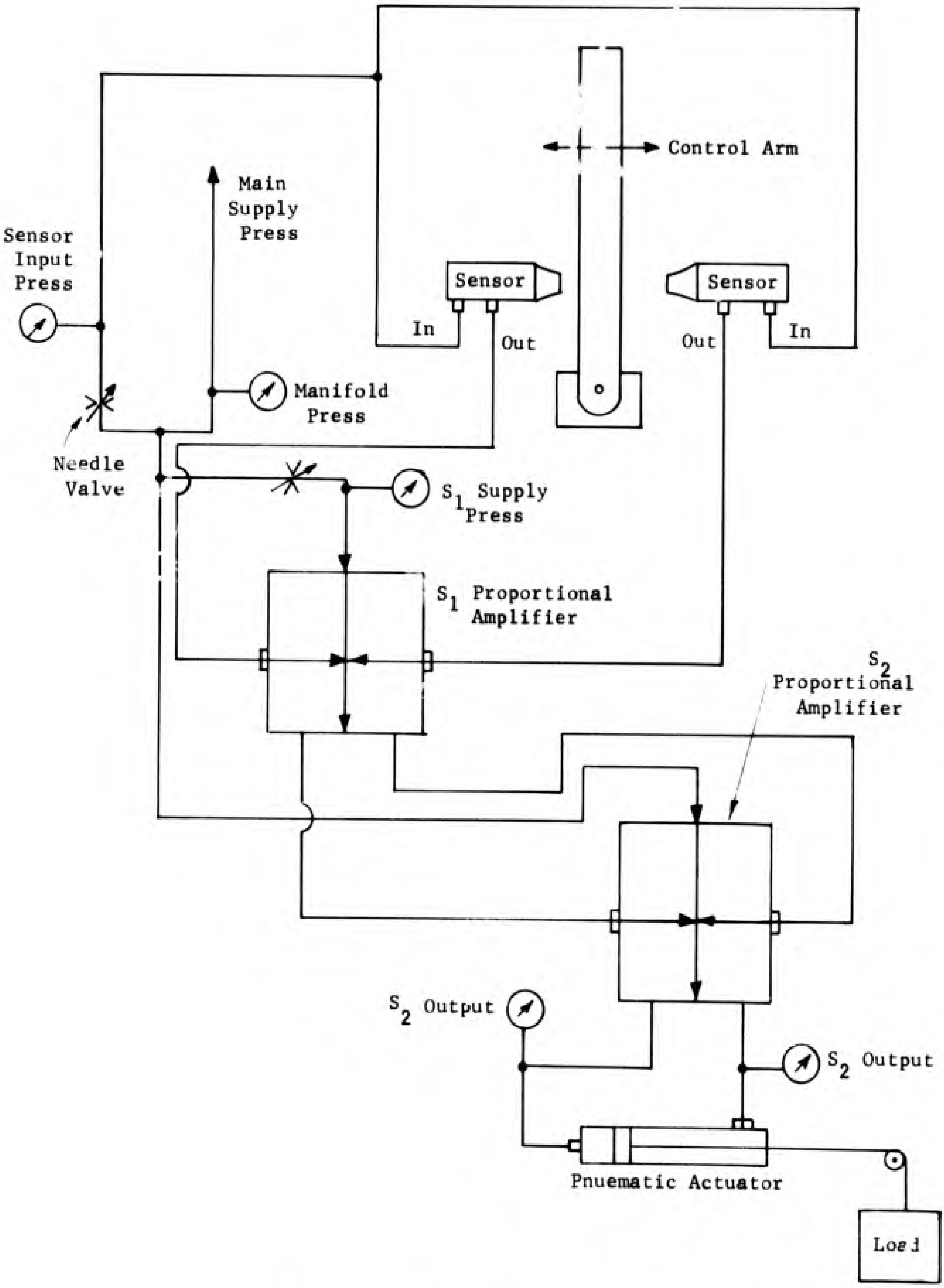

Figure 8. Pnuematic Actuator Control Circuit

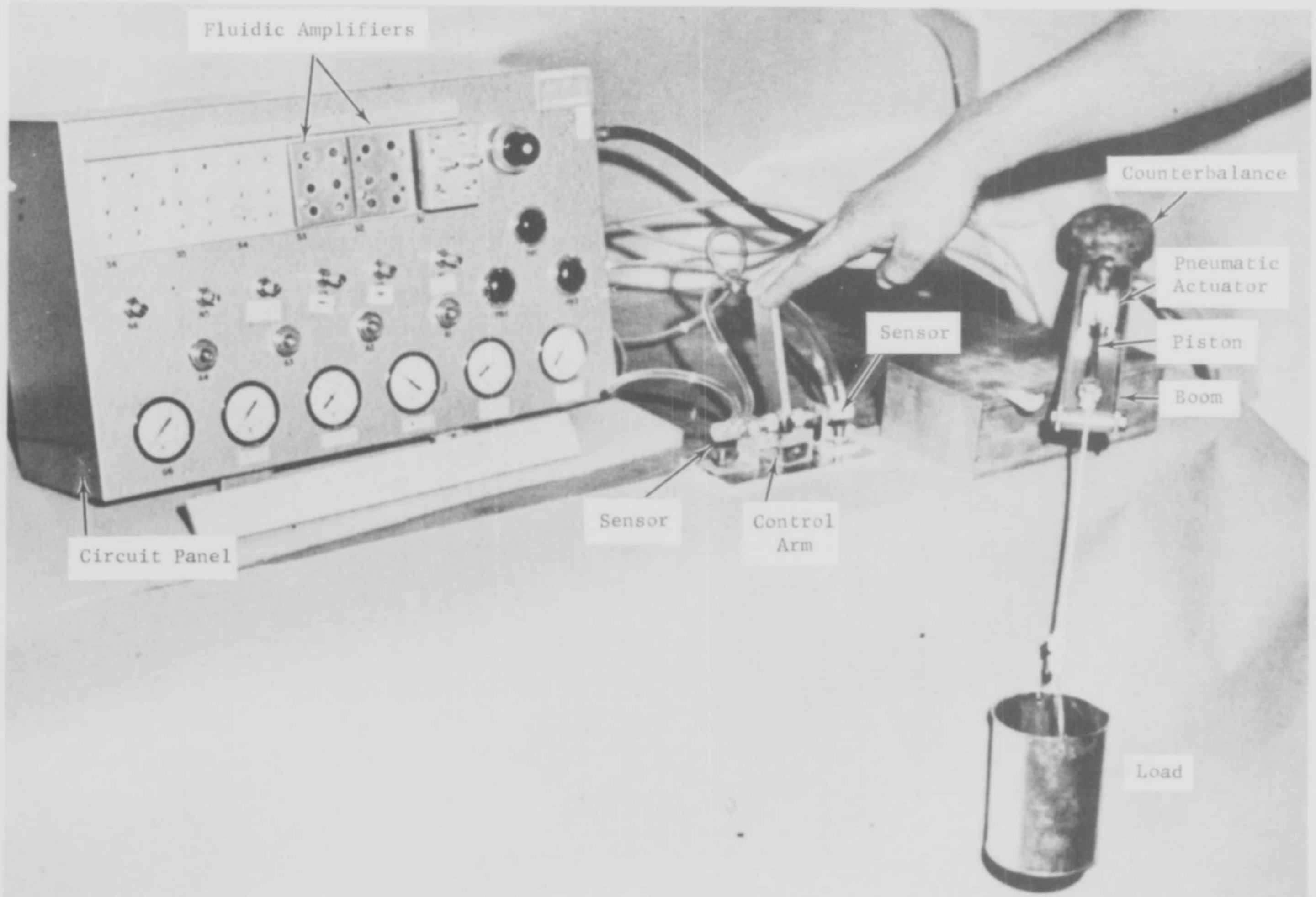

Figure 9. Small Scale Fluidic Control Test

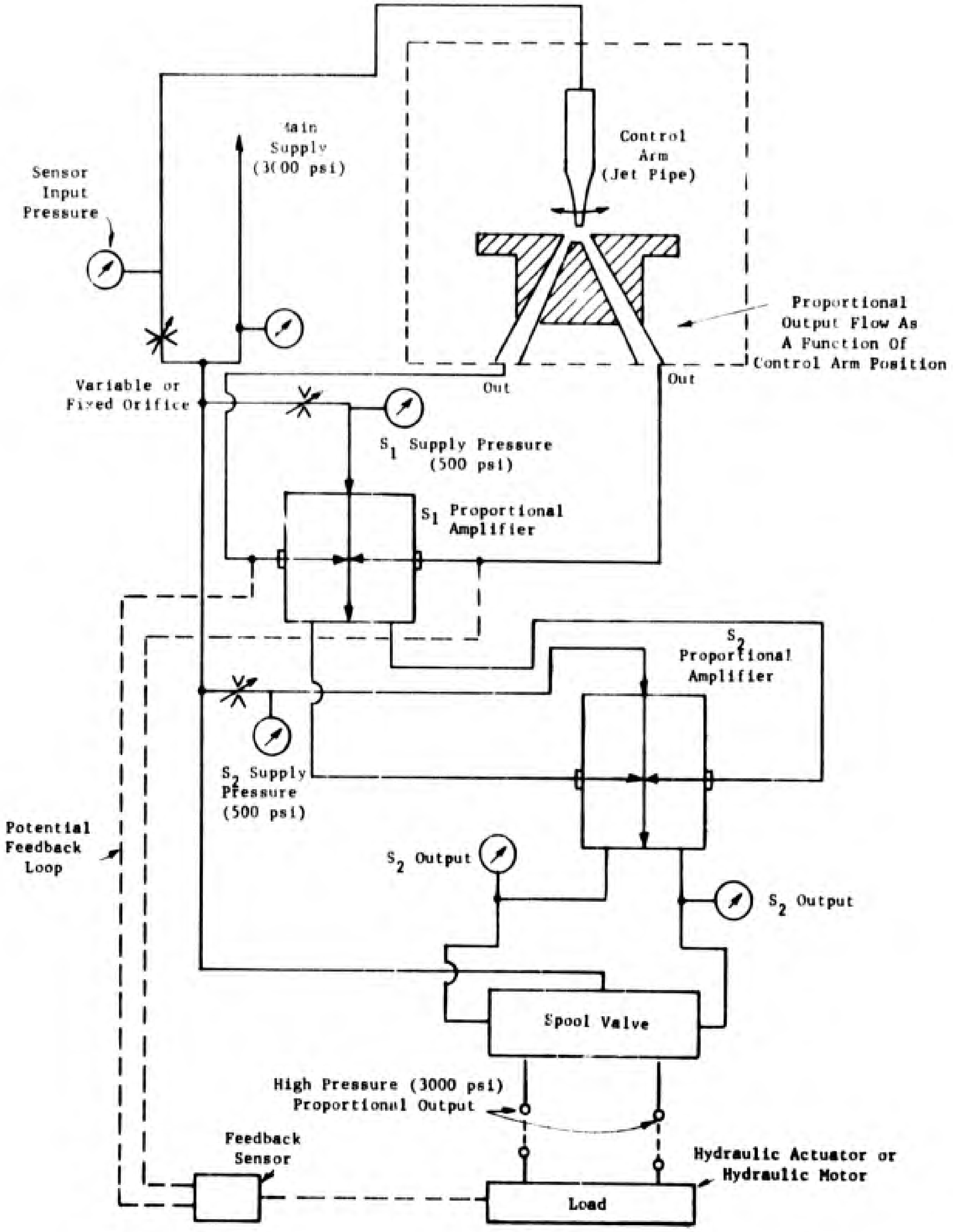

Figure 10. Proposed Hydraulic Fluidic Control System Test

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13. ABSTRACT

A survey is presented of the experimental work that has been done in the area of hydraulic fluidics. The importance of Reynolds number is shown when comparing the performance of a fluidic element with hydraulic oil or air as the working fluid. The results of testing of all-hydraulic fluidic actuator control systems conducted at General Electric Company and at the Naval Ship Research and Development Center support the premise that hydraulic fluidic elements are useful in improving all-hydraulic servo-actuator controls. A small scale test conducted at NCEL with pneumatic fluidic elements showed that a pneumatic actuator can be powered directly by a fluidic amplifier cascade at the lower power levels.

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Water						
Working fluid						
Reynold's number						
Relative power consumption						

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Unclassified
Security Classification