DESIGN OF AN ELECTROHYDRAULIC SERVOVALVE

By

Robert Harley Maskrey

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Signature redacted

Department of Mechanical Engineering, December, 1964

Signature redacted

Thesis Supervisor

Signature redacted

Chairman, Departmental Committee on Graduate Students
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ABSTRACT

A design for the first stage of an electrohydraulic servovalve is presented. The proposal consists of a voice-coil type electro-mechanical actuator, a sliding plate valve, and a pure fluid jet amplifier. The design problems of a no-hysterisis actuator were considered, and an acceptable device was constructed. The important parameters involved in designing a pressure controlled pure fluid amplifier operating on hydraulic oil were investigated analytically and experimentally. A satisfactory model was built on the basis of the findings. The two elements were connected with a sliding plate valve and a graphical method was employed to predict the output. The static behavior of the system agreed well with predicted resulted, but the dynamic response was considerably poorer than that of the actuator alone.

Thesis Supervisor: Herbert H. Richardson

Title: Associate Professor of Mechanical Engineering
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\[ t \] receiver width
\[ v \] jet velocity
\[ w \] receiver separation
\[ x \] coil and valve displacement
\[ x_0 \] axial setback
\[ x_e \] length of flow establishment region (Albertson)
\[ y_0 \] lateral setback
\[ \alpha \] wave number \( = \sqrt{\frac{\mu}{\rho \nu}} \) (Pai)
\[ \delta \] jet deflection
\[ \Phi \] Gaussian integral
\[ \phi \] flux density
\[ \mu_0 \] permeability of air \( 3.2 \text{ maxwells/ampturn in.} \)
\[ \mu \] hydraulic oil viscosity
\[ \rho_w \] wire density - mass/unit length
\[ \rho \] material density
\[ \rho \] hydraulic oil density
\[ \omega_n \] coil natural frequency

Subscript 0 refers to initial condition or static responses
Subscript 1 and 2 refers to right and left sides of amplifier
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I INTRODUCTION AND PROPOSAL
1.1 Introduction

The recent major advances in space technology and industrial controls have brought about new requirements in mechanical hardware. Because of high force, high power, low weight and volume and high reliability requirements of particular control systems, especially in space flight, fluid power control is often chosen to provide necessary mechanical actuation. The requirements of a typical actuating system would be for a high stiffness, high frequency, high force device that could accurately be controlled from low power electrical signals emanating from various transducers and computer operated controllers. The former requirements would call for a hydraulic type of actuator, cylinder-ram combination, while the latter specifies an electrical device. Hence, some form of electrohydraulic actuator is called for that would enable the coupling of these two systems. The resulting device is usually a 4 way hydraulic valve operated with an electrical signal and having an output valve pressure (or flow) proportional to the strength of the input signal.

Various such devices have been designed and produced for specific applications by several companies. The typical configuration of this type controller consists of a moving iron electro magnetic torque motor that translates the electrical input signal into a push-pull output displacement, a pair of flapper nozzle valves that serve to convert the displacement into a sufficient pressure drop to energize a second stage of amplification. The second stage is a spool type
four-way valve acting as the output stage and meeting the hydraulic control requirements.

The above type device combines the advantages of the electromechanical actuator and the hydraulic valve. The electromechanical actuator alone could not provide the required force, nor could it offer adequate stiffness and yet maintain a small size. At the same time the spool valve requires a relatively high force to operate and needs a fairly stiff driving mechanism to counteract the transient and steady state valve forces. So, the combination of these two items requires an interim stage of amplification that is provided by the flapper nozzle valves. This type of actuator is a two-stage electrohydraulic servovalve. A typical configuration is given in figure 1-1.

The advantages of this type actuation as indicated above are: 1) high stiffness - large resistance to force feedback from load. 2) high forces - limited only by the hydraulic source and pressure limitations of the hardware. 3) high response speeds - relatively high frequency possible for a mechanical device. 4) high power gain - large hydraulic power changes possible for small input electrical signals. 5) compatibility - most signal sources are of electrical nature since current or voltage is easier to handle, thus the device is readily adapted to many uses.

There are disadvantages, too, to this type of device. 1) High manufacturing tolerances, hence high cost of finished product. This is the principal limitation to wider industrial acceptance. 2) Steady state leakage. Two stage valves as described generally
Fig. 11  TYPICAL SERVOVALVE
have some quiescent flow through the first stage amplifier even when
the spool is in its neutral position. Additionally, in the torque motor,
there is also a quiescent current present when at rest. 3) Hysteresis -
the primary cause of hysteresis in the electrohydraulic servovalve is
the use of moving iron type transducers.

The purpose of this investigation is to propose and examine
a device that will eliminate the hysteresis problem, reduce the manu-
facturing costs and yet maintain the same advantages presently enjoyed
by existing electrohydraulic valves.

Much of the above emphasis is on the electromechanical
transducer and the first amplifier, or the first stage of the valve, so
the primary concern will be to revise this part of the total valve
ignoring the second stage or spool.

The necessity of reducing manufacturing total cost needs
not be explained. It would greatly expand the possible uses for hydraulic
actuation. Hysteresis is perhaps not so obvious an ill. In any control
system with a closed loop where hysteresis is present, there exists
the possibility for limit cycle oscillation. Since there can be output
with no input, hysteresis leads to error even in open loop. In addition,
the phase shift produced by hysteresis may be undesirable.

1.2 Proposed Design

The device proposed to act as a first stage of a two stage
servovalve uses a voice-coil type electromechanical actuator and a
fluid-jet amplifier. The actuator provides a translation from the
current input to a mechanical motion. This motion in turn controls
the opening of a pair of variable orifices. The orifices regulate the control pressures and flows supplied to the control ports of a pressure controlled proportional pure fluid amplifier. The output of the amplifier is a pressure that serves to operate a conventional spool valve. In essence this first stage is now a two stage device by itself. Figure 1-2 gives a schematic of the operation.

The optimal design of an electromechanical actuator is considered, within the limits of geometry, input current, output displacement, and natural frequency. For a given magnet material and mass load, the design is over specified, so the problem was considered to maximize the natural frequency, retaining the other requirements.

The problems of operating a fluid jet amplifier at low-Reynolds number conditions are considered. The effects of control port geometry and supply pressure on the input characteristics are investigated. A study of receiver port location and geometry was carried out in order to maximize the pressure gain of the device without sacrificing flow gain. A comparison of viscous-incompressible fluid jet amplifiers with pneumatic high-Reynolds number amplifiers is given and some limitations of the device are spelled out.

A sliding plate valve is designed to adequately couple the actuator and the amplifier. A method for prescribing input devices to the amplifier is demonstrated. Impedance requirements and stability problems that determine the operating characteristics of the valve are considered.

A transistor amplifier for operating the valve was designed constructed and tested, and is presented for use with the voice coil
VOLTAGE SIGNAL

Electronic Amplifier 

Voice Coil Actuator 

Variable Orifices 

Fluid Jet Amplifier 

Spool Valve

CURRENT 

DISPLACEMENT 

ΔP 

ΔQ 

Pc 

Ps 

Pr1 

Pr2 

BLOCK DIAGRAM OF PROPOSED FIRST STAGE

Figure 1-2
actuator.

Static and dynamic characteristics of the complete first stage package are presented, and its operation in a closed loop system is evaluated.
II ELECTROMECHANICAL TRANSDUCER
II ELECTROMECHANICAL TRANSDUCER

The most useful electromechanical device for this application is a transducer that translates a current input into a position output. Since a linear spring can be used for reducing a force output to a displacement, the problem then becomes one of causing a bidirectional current to produce a bidirectional proportional force. The three most common ways of meeting this end are through variable reluctance magnetic means, electrostatic means and through the force exerted on a conductor by a magnetic field.

The selection of the proper device depends on the use or type of performance it will be required to attain. The following factors should be considered and their relative importance evaluated to aid in the selection of a device; maximum force, maximum stroke, maximum current, weight of device, natural frequency or speed of response, maximum temperature rise allowable, type of motion desired (i.e., torque or force), geometrical limitations, plus other inherent limitations peculiar to specific devices such as inductance or capacitance effects, hysteresis and quiescent currents.

The particular requirements in this case are for a no-hysteresis, high frequency actuator to meet specific stroke requirements. The first condition would automatically eliminate the moving iron device because of the hysteresis present in a fluctuating magnetic field. The last two conditions would call for a high force to obtain a high frequency (stiff spring) and yet keep an adequate stroke. The remaining methods can be compared on this basis. The force between
two conductors carrying a current is given by:

\[ F_\omega = \frac{\mu_0 l i^2}{2\pi d} \]  

(2.1)

where \( l \) is the length of each wire of distance apart \( d \), and \( \mu_0 \) is the permeability of air. The force on a moving coil device is:

\[ F_c = B l i \]

(2.2)

where \( l \) is again the length of wire carrying a current \( i \) and \( B \) is the magnetic flux density. The ratio of these two forces for the same current and wire length is:

\[ \frac{F_c}{F_\omega} = \frac{B}{\mu_0 i} \]

(2.3)

By limiting \( i \) to 100 ma, and specifying \( d \) to be one half the maximum stroke or .005 inches, letting \( \mu_0 = 3.2 \) maxwells/ampturn - in, and taking \( B \) to be one-quarter the maximum flux in a magnet or about 30 kilomaxwells/in \(^2\), the force ratio becomes

\[ \frac{F_c}{F_\omega} = 10^3 \]

This clearly limits the choice to the moving coil device, where if the limitations were size and not current, the electrostatic device would be comparable since it requires no constant magnetic field.

There are two popular configurations of the moving coil device. The first is the translating coil as in a loud speaker voice coil and the second is the rotating coil as in a d'Arsonval meter movement. Of these two, the voice coil seems the most efficient since the rotating device requires a larger diameter to produce the same force, and only a portion of the coil is in the magnetic field at any time.
2.1 Specifications

There must be some specific requirements for a typical actuator so the design parameters for the coil can be selected and so the feasibility of the particular design can be adjudged. The important specifications are size, natural frequency, displacement, and input signal requirements since these determine the properties of input and output appliances and applications. On the basis of existing torque motors, the following specifications were selected.

- Maximum displacement $\pm 0.005$ in
- Maximum current input $\pm 100$ ma
- Natural frequency 500 cps
- Size $0.750 \times 0.750 \times 0.750$ in

These requirements when coupled with other practical stipulations as temperature rise, available wires sizes, materials, and power limitations define the actuator.

The voice coil actuator can be divided into two parts, the moving coil, its connections and spring, and the permanent magnet. The problem of optimally designing the permanent magnet will first be considered.

2.2 D. C. Magnet

The function of the magnet is simply to provide a constant magnetic field in a gap of sufficient width and length to permit operation of the coil. For convenience a D. C. electromagnet was substituted for the permanent magnet as the effect would be the same.

The configuration that seems to offer the most advantages
as far as compactness and accessibility is a cylindrical magnetic field as in figure 2-1. The geometry and properties of this design must be selected to maximize the flux density in the gap. For compatibility with a suitable coil, the gap dimension of .080 inches width and .250 inches length were chosen, with diameter to be a variable.

The magnetic circuit shown in figure 2-2 can be thought of as analogous to the electrical circuit of figure 2-3. The problem is one of minimizing the leakage permeance or maximizing the permeance of the air gap, similar to minimizing the load resistance and maximizing the parallel resistance in the electrical circuit. At the same time, the permeance of the magnet itself must be a maximum while providing the largest room possible for the D.C. coil. These conditions therefore specify the magnet geometry. To minimize the magnetic "resistance" of the magnet, the flux paths must be as short as possible and the cross sectional area as large as possible. At the same time the area must be small to allow sufficient coil room. The path length is virtually fixed, so the cross sectional areas must be equal and kept a minimum.

The equal area condition allows the following relations in dimensions from figure 2-4.

\[ \frac{\pi d^2}{4} = \frac{3}{4} \pi t \]  \hspace{1cm} (2.4)

\[ \pi d b = \pi \frac{d^3}{4} \]  \hspace{1cm} (2.5)
CYLINDRICAL MAGNETIC FIELD

Figure 2-1

MAGNETIC CIRCUIT

Figure 2-2
EQUIVALENT ELECTRIC CIRCUIT

Figure 2-3

MAGNET DIMENSIONS

Figure 2-4
And geometric compatibility allows these relations:

\[ 2t + d + 2a = .750" \]  \hspace{1cm} (2.6)
\[ b + c + .250" = .750" \] \hspace{1cm} (2.7)

The coil space is a cylinder with cross-sectional area \( a \times c \). For any given wire material the number of turns in this space varies inversely with the \((\text{diameter})^2\) and the current carrying capacity varies directly with \((\text{diameter})^2\). So, the product should be about constant independent of wire size. For a typical wire size there are 320 turns/in\(^2\) at 1 amp maximum current, so the magnetomotive force is

\[ NI = 320 \cdot a \cdot c \text{ amp-turns} \] \hspace{1cm} (2.8)

The material selected for the magnet is SAE 1010 mild cold-rolled steel and the magnetization curve is given in figure 2-5. From this a typical maximum flux density in the steel is picked at 120 kmax/in\(^2\). So the maximum flux at any place in the magnet is

\[ \phi_{\text{max}} = B \cdot \frac{\pi d^2}{4} = 120 \text{ kmax/} \text{in}^2 \cdot \frac{\pi d^2}{4} \] \hspace{1cm} (2.9)

because of the equal areas.

If \( P_T \) is the total leakage and air gap permeance, then for the complete circuit, assuming that the permeance of steel is infinity,

\[ NI = \frac{\phi}{P_T} \] \hspace{1cm} (2.10)

or

\[ 320a \cdot c = 120 \cdot \frac{\pi d^2}{4} \cdot \frac{1}{P_T} \] \hspace{1cm} (2.11)
MAGNETIZATION CURVE for S.A.E. 10-10 COLD-ROLLED MILD STEEL
FROM ROTERS Ref. (1)
It remains only to solve this equation for the proper geometry.

From Roters\(^{(1)}\) the permeance of the air gaps as indicated in figure 2-6 are given as such:

\[
P_1 = 4\mu_0 \left( \frac{d}{2} + \sqrt{g(a+q)} \right) \ln \frac{a+q}{q} \quad (2.12a)
\]

\[
P_3 = 1.63 \mu_0 \left( \frac{d}{2} + \frac{a}{2} \right) \quad (b)
\]

\[
P_2 = \frac{2\pi \mu_0 h}{\ln (1+2g/d)} \quad (c)
\]

\[
P_4 = P_3 \quad (d)
\]

\[
P_5 = 2\mu_0 \left( \frac{d}{2} + \frac{g}{2} \right) \ln \left( 1 + \frac{d}{q} \right) \quad (e)
\]

Adding parallel permeances,

\[
P_r = P_1 + P_2 + P_3 + P_4 + P_5 \quad (2.13)
\]

This cannot be easily solved, but by using the following series approximations:

\[
\ln (1+\varepsilon) = \varepsilon - \frac{\varepsilon^2}{2} + \frac{\varepsilon^3}{3} \quad (2.14a)
\]

\[
\ln \frac{\varepsilon+1}{\varepsilon} = 2 \left[ \frac{1}{2(\varepsilon+1)} + \frac{1}{3(\varepsilon+1)^2} \right] \quad (b)
\]

\[
\sqrt{1+\varepsilon} = 1 + \frac{1}{2} \varepsilon - \frac{1}{8} \varepsilon^3 \quad (c)
\]

\[
\ln \varepsilon = (\varepsilon - 1) - \frac{1}{2} (\varepsilon - 1)^2 + \frac{1}{3} (\varepsilon - 1)^3 \quad (d)
\]

where \(\varepsilon\) is a small number, and applying to equations 2.12a through 2.12e the total permeance can be determined in terms of \(d, a, h, g,\) and assorted constants, \(h\) and \(g\) are known and \(a\) is related to \(d\) from (2.6), so equation (2.13) can be solved for \(d\). Only terms of third order were kept in each unknown, so the final equation was seventh
AIR GAP PERMEANCES Fig. 2-6

FORCES ON MOVING COIL

Figure 2-7
order in d. The real practical root satisfying the equation is

\[ d = .22 \]

Equation 2.4 to 2.7 can be solved for the entire geometry.

Using this geometry and an estimated mean flux path as shown in figure 2-2 combined with the figure 2-5, the total magnetomotive force can be calculated. Airgap and leakage permeances are as given in equations (2.12a) to (2.12e) with the additional leakage permeance

\[ P_l = \frac{2\pi \mu_0 l}{\ln \frac{d + 2a}{d}} \] (2.15)

This occurs through the coil windings and is used to calculate the flux through parts 4, 5, and 6. A brief table of the calculations is given in table 1. The starting point for this estimation was that one part of the magnet had gone into saturation, saturation being approximately 120 kmax/in². These results indicate that the required force is 311 ampturns to produce a flux of 4.0 kmax in the gap. But of this flux only the fraction proportional to \( \frac{P_2}{P_T} \) is actually useful flux.

\[ 4.0 \cdot \frac{P_2}{P_T} = 2.32 \]

Or, this is a flux density of: 13.4 kmax/in² in the air gap.

This not necessarily the ideal design of the D. C. magnet, since the best design would make the length of the gap as long as physically possible, to make \( \frac{P_2}{P_T} \) large, but for the imposed limitations of .250 inches gap length, this the maximum magnetic field.

The wire used for the coil winding is immaterial in theory (see appendix A) but is limited for practical reasons. Power supplies
### TABLE I

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<th>Area in²</th>
<th>Flux kmax</th>
<th>B kmax/in²</th>
<th>H ampturns-in</th>
<th>F ampturns</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>.165</td>
<td>3.92 x 10⁻¹</td>
<td>4.0 ⑥</td>
<td>10.2</td>
<td>2.5</td>
<td>.413</td>
</tr>
<tr>
<td>2</td>
<td></td>
<td></td>
<td>4.0 ③</td>
<td>~</td>
<td>~</td>
<td>237.</td>
</tr>
<tr>
<td>3</td>
<td>.055</td>
<td>1.43 x 10⁻¹</td>
<td>4.0</td>
<td>28.0</td>
<td>3.6</td>
<td>.197</td>
</tr>
<tr>
<td>4</td>
<td>.585</td>
<td>3.8 x 10⁻²</td>
<td>4.426 ⑦</td>
<td>116.0</td>
<td>118.</td>
<td>68.6</td>
</tr>
<tr>
<td>5</td>
<td>.300</td>
<td>6.94 x 10⁻²</td>
<td>4.639 ⑥</td>
<td>67.</td>
<td>7.</td>
<td>2.1</td>
</tr>
<tr>
<td>6</td>
<td>.585</td>
<td>1.25 x 10⁻¹</td>
<td>4.426</td>
<td>35.4</td>
<td>9.</td>
<td>2.3</td>
</tr>
</tbody>
</table>

1) Max $\phi$ is approximately $120 \text{kmax/in}^2 \times A$ where $A$ is smallest area available, so $A = 3.8 \times 10^{-2}$, $\phi = 4.56$ so let $\phi = 4.0$ through gap.

2) $\phi_6 \& \phi_4 = 4 + 2/3 \phi_L$ where $\phi_L = F_P \frac{1}{2} R_L$

3) $\phi_5 = \phi_6 + 1/3 \phi_L$
typically supply current in the 1 amp range so sufficiently small
wire should be chosen to keep the current in this range.

2.3 Moving Coil

Next to be considered is the moving coil, subject to the
same geometry limitations as the magnet air gap. The function of
the coil is to provide a large force or displacement at a high frequency
for a fixed current input in a fixed magnetic field. With a configura-
tion as shown in figure 2-7, balancing the forces on the bobbin gives:

\[ m \ddot{x} + kx = BNI \pi D \dot{i} \]  \hspace{1cm} (2.16)

where \( D \) is a mean coil diameter, and \( k \) is the elastic constant of the
spring. The static current to position transfer function is then:

\[ \frac{x}{i} = \frac{BN\pi D}{k} \]  \hspace{1cm} (2.17)

while the natural frequency for this second order system is given by

\[ \omega_n = \sqrt{\frac{k}{M}} \]  \hspace{1cm} (2.18)

The mass is in three parts, the load which remains constant,
the bobbin which depends on geometry and the coil itself which depends
on geometry and wire properties. The spring mass can be lumped
in with the load mass \( m_l \). The bobbin mass is approximately
given by

\[ m_b = \rho_b \pi D \left( L + \frac{L}{2} \right) s \]  \hspace{1cm} (2.20)

where \( \rho_b \) is the density of the bobbin material. The second term
comes from the end caps. The wire mass is

\[ m_w = \rho_w N \pi D \]  \hspace{1cm} (2.21)

where \( \rho_w \) is the wire mass per unit length.
Natural frequency is then

\[ \omega_n = \sqrt{\frac{BN\pi D (\frac{1}{N})_{\text{max}}}{\mu + \rho_h \pi D (1.16) N + \rho_w N\pi D}} \]  \hspace{1cm} (2.22)

This does not have a practical maximum, but as \( N \) gets very large it is inversely proportional to the wire density. A large number of

turns of low density wire will maximize the speed of response.

A second problem of consideration is heating in the coil, or required power. For a given current, power depends on the total

resistance of the coil. This resistance, for a wire of diameter \( d \) and resistivity \( \rho \) is

\[ R = \frac{N\pi D \rho}{\pi d^{3/4}} \]  \hspace{1cm} (2.23)

Similarly the wire density per unit length \( \rho_w \) is given by

\[ \rho_w = \pi d^{3/4} \cdot \rho_m \]  \hspace{1cm} (2.24)

if \( \rho_m \) is the actual wire density.

Combining the two:

\[ R\rho_w = N\pi D\rho\rho_m \]  \hspace{1cm} (2.25)

Since it is desirable from above to minimize both \( R \) and \( \rho_w \), a figure of merit for the coil wire would then be a low \( \rho\rho_m \). Of all commercially available wire materials Aluminum possesses this minimum, with Copper second.

The maximum expected speed of response, or natural

frequency, would then occur for an aluminum wire coil. If \( N \) is assumed very large, an asymptotic natural frequency can be calculated. For \( N \) large:
For #38 Aluminum and #40 and #44 Copper wire this number is calculated in table 2. In addition the expected natural frequency from equation 2.22 is shown and the coil resistance. From this table, the aluminum coil material would be superior. Unfortunately for the present case, both of these considerations, bobbin and load, are important. Several wire materials had to be tested to determine an adequate coil.

2.4 Prototype Model and Testing

For the prototype model a three leafed brass spring provided both the required translational spring stiffness as well as a rigid radial support for the coil. Figure 2-8 shows the complete magnet and moving coil assembly. The fiber spacers are for testing purposes only and do not appear on the final model.

For the test series on the actuator, a linear variable differential transformer was connected to the moving coil. When the output signal was demodulated, this provided a position pickoff for the coil and the L.V.D.T. provided a mass load to the coil. Power was supplied to the D.C. magnet from a variable current source, as described in Appendix B. The signal to drive the coil was provided by a low frequency oscillator and amplified by a transistorized D.C. amplifier, also described in the Appendix. A diagram of the apparatus as used to obtain frequency response curves and such is given in figure 2-9.
**TABLE II**

<table>
<thead>
<tr>
<th>Coil Wire</th>
<th># Turns</th>
<th>Resistance</th>
<th>Asymptotic $\omega_n$ cps</th>
<th>Expected $\omega_n$ cps</th>
<th>Measured $\omega_n$ cps</th>
</tr>
</thead>
<tbody>
<tr>
<td>#38 Al</td>
<td>437</td>
<td>41 $\Omega$</td>
<td>440</td>
<td>240</td>
<td>110</td>
</tr>
<tr>
<td>#41 Cu</td>
<td>427</td>
<td>43 $\Omega$</td>
<td>345</td>
<td>220</td>
<td>128</td>
</tr>
<tr>
<td>#37 Cu</td>
<td>302</td>
<td>13.5 $\Omega$</td>
<td>216</td>
<td>175</td>
<td>85</td>
</tr>
</tbody>
</table>

* with instrument mass added
VOICE COIL ACTUATOR — TEST CONFIGURATION

Figure 2-8
Low Freq. Oscillator

Amplifier

Power Supply

Magnet & Coil

L.V.D.T.

Demodulator

'Scope

TEST APPARATUS
SCHEMATIC

Figure 2-9
After a period of testing to eliminate the major apparatus flaws, the first major test to be conducted was that to determine magnet saturation. To operate the magnet most efficiently it should be in saturation, thus the magnetic field in the gap is a maximum, but if it is too far in saturation most of the input energy is going into heat and not into gap energy. Further, if the magnet is saturated, any hysteresis loop due to the small demagnetizing field produced by the moving coil will have negligible effect.

To determine this curve experimentally, a sinusoidal signal with a frequency of 10 cycles per second, far below the coil natural frequency, was fed to the voice coil. The resulting amplitude of oscillation was then measured at various levels of current in the D. C. magnet. Figure 2-10 gives these results plotted against both current and ampturns. From this test the indication is that a good operating point would be just above one amp, and the remaining tests were carried out at this level.

The next series of tests were frequency response tests performed under approximately similar input currents. Sinusoidal input current was varied from about 2 cps. up to the level that motion was no longer resolvable. Several values of spring rate were used and the magnet coil wire, both material and number of turns was varied. The output displacement was plotted versus input current on an oscilloscope, and the resulting lissajous figure gave the amplitude and phase of the output. Most of the results from these tests are given in figures 2-11 through 2-14. The two important pieces of information from these response curves are the "gain", 
Figure 2-10

MAGNET SATURATION CURVE

AMPLITUDE

MAGNET CURRENT

amps

ampere-turns
FREQUENCY RESPONSE

Figure 2-II
Figure 2-12

- 37 Copper Wire
- 302 turns
- .003 in Spring
Figure 2-13

- 41 Copper Wire
- 427 turns
- .002 in Spring

Voice Coil Frequency Response
Figure 2-14

- Copper Wire
- 344 turns
- .002 in Spring

FREQUENCY RESPONSE
or input/output ratio, and the natural frequency of oscillation.

Ideally, the input/output ratio could be adjusted to the desired level of 5 mill/100 ma by simply varying the spring rate, with a subsequent change in natural frequency. For a given coil these two quantities should depend only on the value of the spring constant, according to the relations in equations 2.17 and 2.18. Combining these two gives:

\[
\omega_n^2 = \frac{BN\pi D}{M^2} \cdot \frac{i}{\alpha}
\]  

(2.27)

So, a plot of input/output versus natural frequency for several values of spring rate should lie along the line

\[
\ln \frac{i}{\alpha} = -2 \ln \omega_n + \text{const.}
\]  

(2.28)

where the constant varies only for coil properties. Good correlation is found when these plots are made with springs .002 inches and .003 inches thick. Unfortunately only two points for a given coil could be determined since the stiffer springs had such a small displacement that it could not be recorded accurately, and the softer springs produced limiting of the coil motion, plus a marked friction effect due to inadequate radial stiffness. Figure 2-15 gives these plots for several coils with an indication of what effects changing the various coil parameters would have.

From this information, the natural frequency for the coil operating at the desired displacement can be ascertained. A comparison of this result with the previously predicted results is given in table 2.

The initial tests indicated that lighter weight coil bobbins
EFFECT OF $M$, $k$, $B$, and $l$ ON "BREAK POINT"

Figure 2-15
could be constructed and still retain the desired strength. These were made and were tested, not with brass springs as previously, but with a model of the final plate valve in place. These response curves are in figures 2-16, 2-17. While the performance of these coils was better, it is not reflected in the response curve as the dynamic loading was greater, due to the higher mass of the plate valve.

Although the Aluminum coil did show better response gain characteristics than the #37 Copper and #44 Copper coils, many difficulties occurred that would not recommend its final usage. First, the wire appeared much more brittle than the relatively ductile copper, probably because of the manufacturing method. Mechanical failure due to flexing in the higher frequency tests was common. A second manufacturing headache occurs with the joining of the aluminum wires. While copper can easily be soldered, aluminum cannot, and the high temperature soldering methods usually applied to the aluminum would melt the wire.

The coil chosen for the final model was the #42 Copper, Teflon coated 689 turns, and weighing unloaded approximately 2 grams. It has a resistance of 82 ohms and an inductance of 2 millihenries.
VOICE COIL FREQUENCY RESPONSE

**Figure 2-16**

- 38 Aluminum Wire
- 437 turns
- Plate Valve Spring

<table>
<thead>
<tr>
<th>FREQUENCY (cps)</th>
<th>AMPLITUDE (millinches / 100 ma)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.1</td>
</tr>
<tr>
<td>2</td>
<td>0.2</td>
</tr>
<tr>
<td>5</td>
<td>0.3</td>
</tr>
<tr>
<td>10</td>
<td>0.5</td>
</tr>
<tr>
<td>20</td>
<td>1.0</td>
</tr>
<tr>
<td>50</td>
<td>2.0</td>
</tr>
<tr>
<td>100</td>
<td>5.0</td>
</tr>
<tr>
<td>200</td>
<td>10.0</td>
</tr>
<tr>
<td>500</td>
<td>20.0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>FREQUENCY (cps)</th>
<th>PHASE SHIFT (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.0</td>
</tr>
<tr>
<td>2</td>
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</tr>
<tr>
<td>5</td>
<td>3.0</td>
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<td>10</td>
<td>6.0</td>
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<td>12.0</td>
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</tr>
<tr>
<td>100</td>
<td>180.0</td>
</tr>
<tr>
<td>200</td>
<td>180.0</td>
</tr>
<tr>
<td>500</td>
<td>180.0</td>
</tr>
</tbody>
</table>
Figure 2-17

COIL AND VALVE PLATE FREQUENCY RESPONSE

*42 Copper Wire
689 turns
Plate Valve Spring
III FLUID JET AMPLIFIER
III FLUID JET AMPLIFIER

Pure fluid jet devices have been the subject of much investigation in recent years, as a logic and control element. The first reference to their use was by Coanda\(^{(2)}\) in 1933, and very little came of his ideas until 1960 when both Diamond Ordnance Fuze Laboratories and Massachusetts Institute of Technology initiated work on pure fluid logic. Work is presently being done by a large number of private concerns and institutions related to applications.

The no moving parts advantage offered by pure fluid amplification promises a longer life, higher reliability, easier operation under adverse conditions, and faster speed of response than present mechanical control elements. Unfortunately the apparent simplicity of these devices does not pertain to a mathematical representation. It is virtually impossible to adequately predict the operation or even a simple transfer functions for a full range of operating conditions. While this is detrimental, it by no means eliminates their use as a control element, since they can be classified according to various geometry and fluid properties. The characteristics can then be empirically determined for each classification, much like transistors, their electronic equivalent.

Fluid amplifiers fall into several basic classifications, according to the type of control and method of operation. Analog and digital are the two methods of amplification, and they can be either pressure or momentum controlled. Digital devices are either bistable or monostable two state elements and are principally
used for logic circuits. Generally they operate on the Coanda principle of a jet attaching to a downstream parallel wall. Proportional amplifiers, used as control elements, depend on a jet being deflected toward or away from one of a pair of receivers, without attachment. Momentum control uses one or two control jets to deflect a third power jet by impinging perpendicular to it. Pressure control depends on a pressure gradient across a power jet to deflect it. The differentiation of pressure and momentum control is a relative one as all devices are a combination of the two to a greater or lesser degree. Either method of control can be used in either proportional or bistable elements. The particular element to be investigated is a pressure controlled proportional amplifier.

Much work has been done on this type of amplifier, its operation, and characteristics. This work has all been done using air as the operating fluid, and at a Reynolds number of 10,000 or better. Additionally, the operating pressures considered were from a few inches of water to about 20 psi. An amplifier operating on viscous hydraulic oil must have a Reynolds number in the hundreds in order to keep the power level acceptable, and to be of any practical value as a servovalve should have an operating pressure in the hundreds of psi. This set of conditions falls in a region where existing results cannot be directly applied, but methods of analysis can.

3.1 Amplifier Description

In order to select the appropriate geometry and operating
conditions for the hydraulic amplifier the effects of the various parameters must be known. Figure 3-1 gives the basic defining terminology used, and a brief qualitative description of operation and the geometry effects follows.

A power jet of approximately constant flow issues from a high pressure source through the nozzle. As the jet passes through the environment as a free jet it entrains some flow from the surroundings. As the jet passes the knife edges, however, some flow known as return flow is peeled off from the jet. Continuing downstream the jet impinges on the receiver ports where, in the blocked load case, the jet energy is transformed into a static pressure. In the no load case, the jet is free to pass through the receivers. The pressure drop across the jet in the control region $P_1 - P_2$ determines the jet deflection and hence the portion of the jet to strike each receiver.

Looking into the control ports, the appearance is one of an active source rather than a passive impedance since it can accept flow from a lower pressure source. This input flow is determined from the entrained and return flows as above, plus a third or atmospheric flow. The atmospheric flow occurs when there is a pressure difference between $P_1$ or $P_2$ and $P_e$ and is in the direction of the pressure drop. The relationship between this net flow or control flow, the pressure in each control port, and the pressure drop across the jet determines the amount of control one can impress on the jet, and the stability of the jet.

With this background the geometry influence can be evaluated:
FLUID JET AMPLIFIER TERMINOLOGY

Figure 3-1
1. Nozzle width - b. Determines the absolute size of the amplifier, the amount of flow and power for a given pressure drop.

2. Nozzle thickness - This is the same as the amplifier thickness and also determines the power level. The aspect ratio, or ratio of thickness to width, determines the degree to which the device can be assumed two dimensional.

3. Axial setback - x - Determines the amount of entrained flow, and relates to the overall pressure gain since the pressure-area term increases with x.

4. Lateral setback - y - Determines the amount of return and atmospheric flow and largely influences the stability.

5. Receiver distance - L - Effects pressure recovery, gain and degree of venting. For small , the pressure recovery is higher, but venting is poor, and the opposite for large .

6. Receiver thickness - t - Largely effects the pressure gain. For smaller t a smaller portion of the velocity profile is intercepted and gain is higher. For smaller t however, power gain goes down.

7. Receiver separation - w - Affects amount of pressure gain and unlike other dimensions actually possesses an optimum value. For a high flow gain separation is zero and must be determined for pressure gain.

On this basis, in combination with certain practical limitations, amplifier geometry can be selected.
3.2 Power Jet

Something of the nature and shape of the actual power jet must be known before the device to control and receive it can be considered. Of interest would be both the flow entrained into the jet, for control purposes, and the actual velocity profile, for designing the receiver ports.

The operating Reynolds number, consistent with power and flow capabilities, would lie somewhere between 100 and 500 based on nozzle width. This is an awkward Reynolds number since it neither falls in the turbulent or laminar regions. There is no definite transition point, so even if the jet were laminar through the nozzle any slight disturbance would probably incite turbulence. Both turbulent and laminar representations will be examined, though the turbulent will be used on the basis of observed behavior.

A rather extensive analysis of the turbulent jet has been carried out by Albertson(3). He divides the jet into two regions, a zone of flow establishment near the nozzle and an established flow zone where the centerline velocity decays from its previously constant value to zero. The assumptions made are that (1) Jet momentum is constant, or, there are no axial pressure gradients beyond the nozzle. (2) The diffusion process is dynamically similar and (3) The axial velocity component of velocity varies in a gaussian manner at each cross-section.

For the zone of flow establishment (see figure 3.2) the axial velocity is
TURBULENT JET VELOCITY PROFILE
from
ALBERTSON

Figure 3-2
The constant $C_1$ results from assumption (2) and can be evaluated from the jet momentum as

$$C_1 = \frac{1}{\sqrt{\pi}} \frac{b}{x_e} \tag{3.2}$$

Where $x_e$ is the length of the zone of flow establishment. This must be determined experimentally and seems to depend somewhat on Reynolds number.

For the zone of established flow,

$$\frac{V_x}{V_o} = \sqrt{\frac{1}{\pi} C_1 \frac{b}{x}} \exp \left[ -\frac{1}{2C_1^2} \frac{Y^2}{x^2} \right] \tag{3.3}$$

These velocity profiles can then be integrated to give the entrained flow, energy flux and their spatial variation. The constant $C_1$ relates to the amount of spread of the jet and is larger for a faster spreading jet. It is interesting to note that both zones of flow can be described with the determination of only one experimental constant.

A two dimensional analysis of a viscous incompressible jet is offered by Pai (4). Starting with a rectangular velocity profile at the nozzle exit, and a initial velocity of $V_o$ he gives for the velocity as a function of axial distance:

$$\frac{V_x}{V_o} = \frac{1}{2} \left[ \phi \left( \frac{1-Y}{2\alpha \sqrt{x}} \right) + \phi \left( \frac{1+Y}{2\alpha \sqrt{x}} \right) \right] \tag{3.4}$$

where $\phi$ is a modified Gaussian integral.
The divergence of the jet in this case is proportional to the wave number

$$\phi(\varphi) = \frac{2}{\sqrt{\pi}} \int_{0}^{\varphi} e^{-\frac{3^2}{z}} dz$$  \hspace{1cm} (3.5)

The laminar velocity profile is plotted in figure 3-3.

A direct comparison of the laminar and turbulent jets is not immediately obvious. However, for the turbulent jet the spreading rate depends weakly on the Reynolds number, and in fact the spreading rate increases with $R$, while for the laminar case the spreading rate ($q^2$) varies inversely with $R$. For any further comparison, a particular case would have to be examined. This is done for one example in section 3.5.

3.3 Control Region

The function of the control region is to deflect the power jet by an amount dependent on the magnitude and sign of the input signal to the amplifier. The control region extends from the nozzle to the knife edges, and in this zone the input characteristics of the device are determined. Some work on the nature of the input control flow characteristics and their dependence on control port geometry was initiated by Van Koeveering (5). This work was done on a high Reynolds number, low pressure, pneumatic amplifier and is not
TWO-DIMENSIONAL LAMINAR JET VELOCITY PROFILE
FROM PAI

Figure 3-3
directly applicable, but some trends can be applied here.

3.3.1 Characteristics

The conventional and most useful plot of input characteristics indicates the flow into the control port, or demand flow, as a function of pressure in the control port for a given value of pressure drop across the jet. As mentioned before, demand flow is really the sum of three separate flows, the entrained flow, return flow and atmospheric flow (figure 3-4). The entrained flow is virtually constant, independent of pressure. The return flow increases as the jet is deflected toward the knife edge peeling back a larger portion of the jet, so it depends on the pressure difference. The atmospheric flow into the control port increases as the control port pressure is lowered below $P_e$, and decreases as the jet is deflected toward the knife edge since the available "orifice" area is smaller. Since the jet deflection depends on $P_1 - P_2$, the net demand flow depends on both $P_1$ and $P_1 - P_2$. A general shape is given in figure 3-5.

With this general shape in mind, the requirements for a particularly ideal family of curves must be established. The two principal criteria that can be applied to the curves are largest degree of stability, and maximum jet deflection for a given input.

3.3.2 Stability

From continuity, Brown\(^{(6)}\) gives a necessary condition for stability of the jet in the center position using demand flow. This analysis will be modified for the incompressible flow condition
Typical input source characteristics depend on $P_1$, $P_2$, $x_a$, $y$.

Figure 3-4

Figure 3-5
where the only impedance in the control line is a passive resistance.

Continuity of flow requires that

\[ Q_{in} = Q_{demand} \]  \hspace{1cm} (3.7)

Differentiating both sides

\[ dQ_{in} = \frac{\partial Q_d}{\partial P_1} dP_1 + \frac{\partial Q_d}{\partial (P_1 - P_2)} d(P_1 - P_2) \]  \hspace{1cm} (3.8)

Assuming the \( P_2 \) is constant,

\[ \frac{dQ_{in}}{dP_1} = \frac{\partial Q_d}{\partial P_1} + \frac{\partial Q_d}{\partial (P_1 - P_2)} \]  \hspace{1cm} (3.9)

And if the upstream pressure, \( P_c \), is constant \( \frac{dQ_{in}}{dP_1} \) is just the reciprocal of the small perturbation resistance of the orifice or \( \frac{1}{R} \).

This is a minimal condition for stability of the jet in the center position. If the jet deflects a small amount, the flow changes due to the term \( \frac{\partial Q}{\partial (P_1 - P_2)} \), as does the pressure. But the flow changes in the opposite direction due to the other two terms. If the change in flow in is not sufficient to provide the demand change the jet will continue to deflect. As the curves are plotted, this requires

\[ \frac{dQ_{in}}{dP_1} \leq \frac{\partial Q_d}{\partial P_1} + \frac{\partial Q_d}{\partial (P_1 - P_2)} \]  \hspace{1cm} (3.10a)

or, as Brown gives it
\[
\frac{dQ_m}{d(-P_i)} > \frac{dQ}{d(P_2-P_1)} + \frac{dQ}{d(P_2-P_1)}
\] (3.10b)

This criteria indicates that a desirable family would have steep (small \( \frac{dQ}{d(P_2-P_1)} \)) curves, and closely spaced (small \( \frac{dQ}{d(P_2-P_1)} \)) to give stability over the broadest range. To get the largest pressure drop from a change in control orifice, however, the desirable curves would be ones that have a low slope and are closely spaced, so that a small change in flow would produce a large change in pressure.

The most appealing set of characteristics then would be ones that have a medium slope, but are closely spaced.

From VanKoevering's data the trend seems to indicate that a lateral setback on the order of one nozzle width for an axial setback of 4 nozzle widths gives the narrowest spacing, but no conclusions could be drawn because of the seeming disorder to the curves for two-sided amplifiers. It is evident that more information is necessary for the selection of the control geometry.

3.3.3 Geometry

The axial setback of the knife edges is somewhat arbitrary between 2 and 6 nozzle widths. Any less than 2 and it would no longer be a pressure controlled amplifier, and more than 6 the control flow would be large and the control region would be extending into the zone of established flow and the realm of the receiver ports. On the basis of VanKoeveering's work the axial setback was selected as 4 nozzle widths.
With the previous conditions satisfied, the lateral setback should be chosen to give the largest level of control flow without allowing appreciable atmospheric flow. This, in effect, raises the level of the characteristic curves, so larger pressure changes are possible, and yet by keeping the atmospheric flow small, maintains control of the jet. To determine the relation between \( \gamma_c \) and control flow a plot of \( Q_c \) versus jet pressure for various \( \gamma_c \) was obtained. The following section describes the apparatus used for these tests.

3.3.4 Experimental Equipment

To aid in selecting the control port geometry and for measuring the demand flow characteristics an experimental model was constructed. In order to avoid any Reynold's number problems in applying data to the final model, the experimental model was constructed to be the same size, and was operated under the same conditions as anticipated for ultimate prototype operation. The amplifier was designed with variable nozzle width, axial, and lateral setback, to be as flexible as possible and allow a wide range of geometry variation. Input ports were provided for the power jet and for both control ports. A diagram of the apparatus is shown in figure 3-6.

The device consists of three layers. The base plate contains the input ports and provides a base, the amplifier itself is composed of various shaped plates 1/32 inches thick held to the base with machine screws, and the top is a 1/8 inch plexiglass cover. By
Supply Input

Movable Knife Edges

Control Input

FLUID JET AMPLIFIER - TEST MODEL

Figure 3-6

Flowmeter Amplifier Model

Figure 3-7 APPARATUS
moving a side plate, the nozzle width can be adjusted over a limited range, and by moving the knife edges both the lateral and axial setback can be varied. A distinct disadvantage to the model as designed was the long region downstream of the control ports. With a cover in place over the whole length, this area acted as a downstream control region and caused the jet to attach to the side wall. This problem was remedied by shortening the length of the cover to only include the control region.

It was impossible to operate the amplifier in an atmospheric environment with oil as the fluid. Presumably because of the lower density of air, it was drawn into the control region much easier than the hydraulic oil and provided for highly erroneous results. So, the model was operated submerged in a bath of oil.

The necessary measurements for this series of test are, the pressure in each control port, the flow into one control port and of course the jet supply pressure. A schematic of the apparatus used for these measurements is given in figure 3-7. Because of the stability problem it is desirable to keep the control line resistances small, so the smallest possible value of control supply pressure, and the largest valves were used. For points where the control port pressures are negative, the control supply was simply an open tank.

The nozzle width was determined from power consumption considerations. The flow through the amplifier should be no greater than maximum servovalve leakage flow, (about .2 gpm) at 500 psi. This places the nozzle width at .016 inches for a depth of .032 inches.
The axial setback was maintained at 4 nozzle widths or .064 inches for all tests. These dimensions were set under a toolmakers microscope before each series of tests.

The control flow into the amplifier was measured as a function of jet pressure for various values of lateral setback. The results are plotted in figure 3-8. These tests were made with no pressure difference across the jet, and with the pressure in each control port at ambient, so there would be no atmospheric flow component. For small values of \( \gamma_0 \), the knife edges protrude well into the jet and a substantial amount of flow is returned, so the control flow is small. At very large values of \( \gamma_0 \), no flow is returned, and the control flow is just the entrained flow. For small values of supply pressure (low jet velocity) the jet spreads very fast, more of the jet is returned and the device ceases to behave as an amplifier.

Several demand curves for various \( \gamma_0 \) were made to verify that the shape of the curves is similar, and only levels change appreciably with small variations in lateral setback.

In effect, the greater the control flow, the more control one has over the jet for a given orifice change. For a given change in area of the orifice in the supply line, the pressure change is higher for a curve with a larger mean flow. This is shown in figure 3-9.

This leads to the selection of a value of \( \gamma_0 \) for which the control flow is largest. But, as \( \gamma_0 \) increases, the atmospheric flow increases, so beyond a certain point, increasing \( \gamma_0 \) is undesire-
CONTROL FLOW AS A FUNCTION OF $P_s$ AND LATERAL SETBACK

Figure 3-8
EFFECT OF HIGHER MEAN FLOW

Figure 3-9
able, as an increase in atmospheric flow can lead to instability. From the above, investigation, \( \frac{\gamma_0}{b} = 1.25 \) seems to be near the edge of the jet, since the change in return flow for a change in is small, yet is not too large to allow atmospheric flow since there is still some return flow.

For this value of \( \gamma_0 \), and for supply pressure of 300 psi a complete set of demand flow characteristics was measured. This curve was taken by varying the upstream orifices, and by varying the control supply pressure to keep the pressure difference constant, and changing the control pressure level. The flow into one control port was measured for each value of control pressure. Because of the dependence of the control pressures and flows on both the jet deflection and the control supply pressure, \( P_c \), it was impossible to vary them independently. This plus trying to keep \( P_c \) low and the orifice resistance small for stability reasons made the data taking quite tedious. It should be noted that the jet position was monitored with a small tube downstream and no instabilities were observed.

Figure 3-10 gives these curves for one side of the amplifier. The flow was nondimensionalized with respect to the control flow at \( P_1 = P_2 = 0 \) for one side, where \( Q_{\omega} = 94 \text{ ml/min} \).

An adequate control region geometry for proportional amplifier operation then, would find \( \frac{X}{b} = 4 \) and \( \frac{\gamma}{b} = 1.25 \) for a jet supply of about 300 psi. Since this information was obtained without receiver ports, the condition of negligible receivers must hold for their design.
\[
\frac{y}{b} = 1.25 \\
\frac{x}{b} = 4.0 \\
P_s = 300 \text{ psig} \\
\alpha P_2 = 8 \text{ psi} \\
\beta P_2 = -4 \text{ psi}
\]

Figure 3-10

CONTROL PORT PRESSURE \( P_2 \) \( \text{psig} \)

CONTROL PORT DEMAND CURVE

RIGHT HAND CONTROL PORT DEMAND CURVE
3.4 Jet Deflection

The deflection of the jet due to the imposed pressure field in the control region determines the relation between the receiver section and the input section of the amplifier. It is desired to know the deflection of the jet center line as a function of axial distance from the nozzle due to a given pressure drop across the jet.

If the jet is considered steady flow, and viscosity is ignored, a force balance on a particle of fluid exiting from the nozzle gives in the s direction:

\[ \sum F_s = \rho V \frac{\partial V}{\partial s} \]  

(3.11)

And perpendicular to the jet

\[ \sum F_\perp = \rho \frac{V^2}{R} \]  

(3.12)

But from the previous assumption of constant jet momentum, there are no pressure gradients along the jet, providing only small deflections are considered. Hence, consider only equation (3.12). This states that the motion of the jet will be circular for a constant force field.

Assuming then a constant pressure field, the jet deflection can be calculated simply as follows. Consider the control volume shown in figure 3-11. The forces acting across the control volume are the pressure and momentum forces. The sum of the pressure forces is \((P_1 - P_2) A_c\). The momentum terms are not easy to compute unless some assumptions are made. The entrained flow has no contribution since it is equal for both sides. The return
CONTROL VOLUME FOR
JET DEFLECTION

Figure 3-11
and atmospheric terms do have a contribution equal to their respective momenta if the knife edges are perpendicular to the jet. The distribution of these flows is not known, so it will be assumed to be uniform across the control region. If this assumption is made then the control flow can be used to evaluate the momentum since

\[ Q_c = Q_e + Q_r + Q_a. \]

The complete force balance is then

\[ (P_1 - P_2) A_c + \frac{\rho Q_1^2}{A_c} - \frac{\rho Q_2^2}{A_c} = \frac{\rho Q_i^2}{A_i} \tan \Theta \]

(3.13)

The flow terms are often small, and since the characteristic curves are flat, \( Q_1 \) is not very different from \( Q_2 \) so this term is usually neglected.

If the angle is small, \( \tan \Theta \approx \Theta \) so,

\[ (P_1 - P_2) A_c = \rho \frac{Q_i^2}{A_i} \Theta \]

(3.14)

Downstream of the control region, the jet follows a straight line at an angle \( \Theta \) from the center line with its apparent origin at \( \frac{1}{2} X_0 \) from the nozzle. The deflection \( \delta \) at any point \( l \) can then be given approximately by

\[ \frac{\delta}{l - \frac{X_0}{2}} \approx \Theta \]

(3.15)

If the jet flow \( Q_j \) is given by

\[ Q_j = A_n C_d \sqrt{\frac{2P_i}{\rho}} \]

(3.16)
where \( A_1 = A_n C_d \), then the deflection is

\[
\frac{S}{l - x/2} = \frac{P_1 - P_2}{P_s} \frac{A_c}{C_d A_n}
\]

(3.17)

Where \( C_d \) is the orifice discharge coefficient.

For the purpose of evaluating this term, the discharge coefficient for a rectangular nozzle as a function of jet pressure \( P_s \) was measured. This curve is given in figure 3-12 for nozzles with an aspect ratio of 2:1 and 4.5:1.

3.5 Receiver Ports

The function of the receivers is to collect an appropriate amount of the deflected jet some distance downstream of the control region and to either pass this portion as a flow to the load, or to transform the jet kinetic energy into a pressure for blocked load conditions. There are three important geometric properties associated with receiver design. The distance downstream to the receivers, \( l \), determines how "flat" the velocity profile is, as seen in Section 3.2, thus determining how much the jet must be deflected to cause a change in flow or pressure in one receiver. This distance also is related to the amount of "venting" present for the power jet, so it determines whether or not the receivers influence the control region. The receiver width, \( t \), largely influences the ratio of pressure recovered to jet deflection. It also determines the maximum pressure recovery, and whether the important output is pressure (small \( t \)) or flow (large \( t \)). The amount of receiver
AMPLIFIER NOZZLE DISCHARGE COEFFICIENT AS A FUNCTION OF SUPPLY PRESSURE AND ASPECT RATIO

Figure 3-12
spacing \( w \), slightly influences the maximum pressure gain, and has an optimum value for maximum pressure gain, while for maximum flow gain \( w = t \).

It is not real clear whether a maximum pressure gain or a maximum flow gain device is desired, for driving a ram type load. Since the area of the load is small, however, and much of the operation of the ram is assumed static, the design will be oriented toward a pressure gain device without sacrificing flow.

3.5.1 Analysis

Assuming that the flow is purely two-dimensional, that the receivers do not affect the jet, and that the operations in the control port do not significantly affect the jet velocity profile, the influence of geometry on the pressure and flow recovery is investigated. The validity of the above assumptions, and the important practical considerations will later be considered.

Figure 3-13 shows a deflected jet impinging on a receiver port. For a blocked load condition, there is no flow through the receiver, and all of the velocity entering the jet is assumed to be recovered. If \( \delta \) is the jet deflection at the receiver, and the velocity profile is given by \( v(y) \), then the pressure in one receiver is given by

\[
P = \frac{1}{2} \int_{-\delta}^{\frac{1}{2} - \delta} \rho v^2(y) dy
\]  

(3.18)

The integrals are taken from \( -\delta \) to \( \frac{1}{2} - \delta \). For the case that the
DEFLECTED JET IMPINGING ON RECEIVER

Figure 3-13
receivers are separated by a splitter only, as shown, (i.e. \( w = t \)), the pressure difference between the ports is given by

\[
\Delta P = \frac{\rho}{2} \int_{-\delta}^{t-\delta} \gamma(y) \, dy - \frac{\rho}{2} \int_{-\delta}^{t-\delta} \gamma(y) \, dy - \frac{\rho}{2} \int_{t-\delta}^{-\delta} \gamma(y) \, dy - \frac{\rho}{2} \int_{t-\delta}^{-\delta} \gamma(y) \, dy
\]  

(3.19)

Substituting the velocity profile from equation 3.3, where \( x = \ell \) and the integral in the denominator is just \( \ell \), gives

\[
\frac{\Delta P}{\frac{\rho}{2} v_0^2} = \frac{b}{\sqrt{\pi} \zeta \ell t} \int_{-\delta}^{t-\delta} e^{-\frac{\gamma^2}{\zeta^2 \ell t}} \, dy - \frac{b}{\sqrt{\pi} \zeta \ell t} \int_{t-\delta}^{-\delta} e^{-\frac{\gamma^2}{\zeta^2 \ell t}} \, dy
\]  

(3.20)

For the constant \( C_1 \) the substitution

\[
C_1 = \frac{1}{\sqrt{\pi}} \frac{b}{\chi e}
\]  

(3.2)

can be made. At the suggestion of Simson\(^{(7)}\), the value of \( \chi e/b \) is taken as 5.2, where actually it is probably somewhat smaller. So, the resulting profile will be an overestimate. The pressure drop is then given by

\[
\frac{\Delta P}{\frac{\rho}{2} v_0^2} \frac{\ell}{b} = \frac{\ell}{t} \left[ \Phi\left(\frac{t-\delta}{\ell \zeta / \sqrt{2}}\right) - \Phi\left(\frac{-\delta}{\ell \zeta / \sqrt{2}}\right) \right] - \Phi\left(\frac{-\delta}{\ell \zeta / \sqrt{2}}\right) + \Phi\left(\frac{t-\delta}{\ell \zeta / \sqrt{2}}\right)
\]  

(3.21)
where $\Phi$ is the value of the probability integral

$$
\Phi(z) = \frac{1}{\sqrt{2\pi}} \int_{-\infty}^{\infty} e^{-\frac{z^2}{2}} \, dz
$$

The resulting pressure profile is plotted in figure 3-14, for the case that $x_e = 5.2b$, for various values of $t/l$. From this plot, the "gain", or change in pressure for a unit change in deflection, increases as the receiver width decreases, everything else remaining constant. But, the change is small for values of $\frac{t}{l} < 0.25$, and the useful pressure range seems to be largest for $\frac{t}{l} \sim 0.2$. The gain would continue to increase for smaller values of $\frac{t}{l}$, and the useful range would decrease. Physically this can be explained in terms of the velocity profile. The pressure recovered depends on the average velocity striking the port. For smaller receivers, the average velocity is higher since only the peak of the profile is intercepted, while wider ports recover a lower average velocity.

The flow into the receivers for a no-load condition behaves just the opposite. Since it depends on the average velocity times the area, the flow increases with port width.

No-load flow for a single receiver is given by

$$
Q = \int_{-\delta}^{t-\delta} u(\gamma) \, d\gamma
$$

(3.23)

where $Q$ is the flow per unit height, or two-dimensional flow,
PREDICTED
OUTPUT PRESSURE
DROP
vs.
JET DEFLECTION
and
RECEIVER WIDTH

Fig. 3-14  JET DEFLECTION  d/1

OUTPUT PRESSURE DROP

\[ \frac{\Delta P_1}{2 \rho v_2^2 b} \]

- 71 -
assuming that the receivers are lossless. Again substituting the established flow velocity profile,

\[ Q = \nu_0 \sqrt{\frac{1}{2\pi}} \frac{b}{c_1 x} \int_{-\delta/c_1}^{+\delta/c_1} e^{-\frac{y^2}{c_1^2 x^2}} \, dy \]  

(3.24)

with the same substitutions as previous,

\[ Q = \nu_0 \sqrt{c_1 b l 2\pi} e^{-\frac{y^2}{2c_1^2 x^2}} \, d \left( \frac{y}{c_1 l} \right) \]

or

\[ \frac{Q}{\nu_0 \sqrt{c_1 b l 2\pi}} = \Phi \left( \frac{t-\delta}{c_1 l} \right) - \Phi \left( \frac{-\delta}{c_1 l} \right) \]  

(3.25)

This term is plotted as a function of jet deflection and receiver width in figure 3-15. Only the flow into one receiver is considered, since the receiver flow difference with a ram type load is rather meaningless. Figure 3-16 gives both single receiver flow, and pressure difference as a function of deflection and port width over the range of deflection considered in an amplifier. This plot is for \( \chi e = 5.2b \) again. While the changes in pressure gain with port widths are considerable, the flow is only a weak function, so any reasonable selection of \( t/l \) based on pressure gain would provide adequate flow.

To this point, the receiver spacing has not been considered. The effect of varying \( w \) can be evaluated for a given \( t/l \) only. As \( w \) increases from \( w = t \), the pressure in the second receiver de-
NO-LOAD FLOW AT ONE RECEIVER vs. JET DEFLECTION and RECEIVER WIDTH

Figure 3-15
Figure 3-16

SINGLE RECEIVER FLOW and OUTPUT PRESSURE as a function of JET DEFLECTION and RECEIVER WIDTH
creases. Beyond a certain point $w = w_{opt}$, however, the receivers are so far apart that the jet must be deflected a large distance to reach the other receiver, and the "gain" is lower. For a particular case, $t/l = .1$ the effect of varying $w/l$ is seen in figure 3-17. Close to $w = w_{opt}$ the variations in $w$ produce very little change in $\Delta P/\delta$.

The selection of $l$, the axial receiver distance is based primarily on practical reasons. From a purely theoretical standpoint, $l$ should be as close to the end of the zone of flow establishment as possible. At this point, the centerline velocity is still a maximum, so the pressure recovery is largest. Beyond this point and in the zone of flow establishment the velocity profile is flatter, so the pressure "gain" is smaller. The real function of axial displacement is to provide adequate venting for the jet after stagnation at the receivers. If $l$ is too small the pressure in the receiver area will increase because of insufficient space for the fluid to exhaust. This point is not easily determined, but satisfactory operation should be expected for $l = 2x_e$ if the downstream knife edge angle is large.

3.5.2 Experimental Investigation

Since the accuracy of modelling a low Reynolds number high viscosity jet with a turbulent two-dimensional profile is somewhat doubtful, an experimental investigation into the effects of parameters discussed in the previous section was undertaken. For these tests, a series of model amplifiers was constructed with identical control regions. The models were machined in Plexiglass
Figure 3-17

INFLUENCE OF RECEIVER SPACING
for \( \frac{d}{l} = 0.1 \)
on a Green Instrument Pantograph milling machine. This proved to be the simplest means for producing a large number of models quickly and with the most flexible geometry. This placed some limitations on geometry, however, since the nozzle width, aspect ratio, and receiver width are limited in size by available milling cutters.

For purposes of evaluation, blocked load conditions were considered, and the receiver pressure as a function of differential control port pressure was measured. Except for the venting measurements, the models had only one receiver located on the center line, so only one receiver pressure is given. Bias of the jet proved to be a problem and in several instances the control pressure required to center the jet was equal to the useful range of deflection. The profile is shown centered for ease of comparison.

The parameters that were varied on the 15 models tested included the downstream receiver distance, aspect ratio, supply pressure, receiver port shape, and venting possibilities. The criterion for evaluating the results were pressure recovery, or ratio of pressure recovered to supply pressure, and "gain" or ratio of pressure change at the receivers to control port pressure difference.

For convenience in measurements, it was desirable to run these tests at a jet supply pressure of 150 psi. Theoretically $P_s$ makes no difference on either pressure recovery or gain. As the pressure increases, the jet momentum increases in proportion, and the control pressure required per unit deflection is also increased.
But, the pressure at the receivers increases in proportion, so the gain is constant. Practically, however, the supply pressure does have an effect due to changes in the discharge coefficient, degree of two-dimensionality, entrance conditions at the receiver etc. The effect of supply pressure on pressure recovery was measured for a nozzle width of .014 inches, $\frac{t}{l} = .10$, $\frac{a}{b} = 10$ and an aspect ratio of 4.5:1, and is given in figure 3-18. The indication is that a slight improvement can be anticipated for an increase in pressure.

For a highly viscous jet, it is expected that the aspect ratio would have considerable importance. Unfortunately it was possible to investigate only two values because of machining problems. A 2:1 aspect ratio was easy to obtain with a conventional 1/64 milling cutter, but for a 4.5:1 ratio a special cutter was purchased .014 inch in diameter and .065 inches long.

For these two aspect ratios, pressure recovery as a function of control pressure drop is plotted in figure 3-19. To match these curves with the ideal two-dimensional curves, the pressure profiles of both the laminar and turbulent jets for the same conditions are plotted on the same figure. These ideal curves were obtained by integrating the velocity profile for one receiver as in Section 4.3.1, then applying equation (3.17) to relate the jet displacement to control pressure. All plots are for $\frac{t}{l} = .1$ and $\frac{a}{b} = 10$.

Although the maximum pressure recovery is lower, an aspect ratio of 4.5:1 provides a close approximation to the theoretical turbulent jet results. The laminar profile predicts a much higher
Figure 3-18

PRESSURE RECOVERY AS A FUNCTION OF SUPPLY PRESSURE

\[ \frac{P_R}{P_S} \text{ versus } P_s \text{ psi} \]
Figure 3-19

PRESSURE RECOVERY AS A FUNCTION OF ASPECT RATIO

Pai Laminar
Albertson Turbulent

Aspect Ratio
A.R. = 2:1
A.R. = 4.5:1

RECOVERED PRESSURE

CONTROL PRESSURE DIFFERENCE \( \frac{P_c}{P_0} \)

\( P_{R/P_0} \)
gain than seems attainable, indicating that the original turbulent assumption is probably valid. The difference between 2:1 and 4.5:1 aspect ratios is small, but sufficient to warrant the change.

Several pieces of information can be gleaned from these figures. Since the slope of the curve is \( \frac{\Delta P_r}{\Delta R} = \frac{\Delta P_e}{\Delta P_c} \), this is just 1/2 the maximum pressure gain for the amplifier,

\[
G_{\text{max}} = \frac{\Delta P_R}{\Delta P_c} - \frac{\Delta P_{R1}}{\Delta P_c} = 2 \left| \frac{\Delta P_{R1}}{\Delta P_c} \right|_{\text{max}} \tag{3.26}
\]

providing the distance between the receivers is appropriately chosen. The value of \( w \) can also be chosen from the figure. For a device with maximum gain at null, \( w \) should be the width of the profile at the point of maximum slope. For a device with maximum operating range, \( w \) should be the profile width at a point half of the profile height. Since the "width" on these curves is a pressure ratio, \( w \) can be calculated by applying equation (3.17).

Since the ordinate of the curve is a pressure and the abscissa is effectively a displacement (jet deflection), for a similar set of conditions, the area under the curves must be constant, if the jet momentum is in fact constant. So, the area deviation is a measure of the two-dimensionality or efficiency of the amplifier.

Similar tests were performed on amplifiers with an aspect ratio of 4.5:1 and a receiver width \( \frac{1}{b} = 1.0 \) for various \( \ell \). Figure 3-20 shows the pressure profile for \( \frac{1}{b} = 7.5, 10 \) and 15. As expected, the gain does decrease with \( \ell \) because of a flatter velocity profile, however, the improvement due to changing from
Figure 3-20

Pressure Recovery as a Function of Control Pressure Difference $\Delta P_r/P_s$ for $U/b = 7.5$, $U/b = 10$, and $U/b = 15$. 

Recoverd Pressure $P_r$
If viscosity is a problem as anticipated, it seems reasonable that the rectangular velocity profile exiting from the nozzle will not only decay from the sides, but also from top and bottom. Hence, the profile reaching the receivers would not be "wedge" shaped as assumed, but more "cone" shaped. The best receiver port then would be a circular shape located at the center of the profile. Several of these were investigated, and proved to have a slightly higher gain, but the maximum pressure recovery was smaller. See figure 3-21. The improvement was not sufficient to merit the increased manufacturing problems.

Several methods of venting were investigated in lieu of increasing $\ell$. Since the usual condition of operation is blocked load, the power jet stagnates at the receiver, and provision must be made for exhausting the flow. Otherwise, the minimum receiver pressure would be limited, and the assumption made in Section 3.3 that the control region is independent of the downstream geometry would no longer be true. These models were investigated with different venting methods, and are shown in figure 3-22. The first model uses vents placed far downstream in the nozzle, and $\ell$ is small so the gain should be large. The second model uses vents cut into the sides of the receiver walls close to the edge, but has the receivers placed twice as far downstream. The third model uses no vents at all but depends on the large value of $\ell$ ($\ell \approx 10b$) to provide venting. A plot of the pressure recovered as a function of
Figure 3-21

Pressure recovery as a function of control pressure difference $P_1 - P_2 / P_3$ for different aspect ratios and receiver diameters:

- Aspect Ratio = 4.5:1
  - Conventional Receiver
  - 0.155" Dia. Receiver

- Aspect Ratio = 2:1
  - Conventional Receiver
  - 0.014" Dia. Receiver
MODELS FOR VENTING STUDY

Figure 3-22

LOAD PRESSURE vs. CONTROL PRESSURE
DIFFERENCE for VENTED MODELS

Figure 3-23
control pressure for the three models is given in figure 3-23. Also indicated is the mean recovery pressure ratio. The first model is poorest from both gain and pressure recovery, while the other models have either good gain, or good pressure recovery. The following reason is offered for the poor showing of model I. As the jet enters the receiver it has a high velocity and low static pressure, while farther down the tapered receiver the pressure becomes higher and velocity lower. Since the velocity has direction while the pressure does not, it is more advisable to vent at the high velocity low pressure region. At the downstream location, both static pressure and flow are bled. The pressure recovery is higher for model III because of losses in the entrance region of the receiver occurring in model II. The difference in gain presumably occurs because of momentum effects present in model II. Because of the higher pressure recovery, the no-vent model was adopted on the assumption that the gain could always be increased by altering other geometry.

3.6 Extraneous Effects

A whole host of variables complicate the design of a fluid jet amplifier in addition to those just discussed. Bias of the jet and lack of two-dimensionality seem to be the most significant.

The degree of bias is not predictable, and the cure is not at all simple. Several causes manifest themselves. First is turbulence incited upstream of the nozzle, causing the jet to leave the nozzle in a non-uniform profile. This is the reason for the long, uniformly tapered region prior to the nozzle, to damp out any
irregularities. Next is the nozzle itself. If one side of the control wall next to the nozzle is set back axially farther than the other, a severe bias in the jet occurs. This can only be cured by accurate machining techniques. Probably the most significant cause is different amounts of lateral setback for each of the knife edges. This is a difficult dimension to control since the knife edge is quite sharp. If it were made blunt and easy to locate, the jet would attach to the downstream edge, and the amplifier would be bi-stable. The easiest solution proved to be machining the control walls with a small flat edge parallel to the center line, which was finished in the last pass of the milling cutter.

Estimating the amount of two-dimensionality is difficult because the velocity of the jet is constantly changing. Two approximations can be made, as order of magnitude guesses.

One approximation is that of flow through a channel. This is not strictly true because the assumption of channel flow is that the flow at all cross sections is the same, and that the flow is two-dimensional only. Since there can be flow in the Y direction in an amplifier, this assumption is not warranted. A second approximation is that of flow past a flat plate, considering only the bottom half of the device. This is a closer approximation, but still flow is two-dimensional.

Schlichting\(^\text{(8)}\) gives the boundary layer growth for both of these situations as a function of distance downstream. Considering the boundary layer depth at the jet center line and assuming constant velocity, following results are obtained at the receiver ports. For
the first model, \( b = 0.016 \), \( P_s = 150 \) psi, aspect ratio = 2:1 and \( \ell/b = 10 \), the estimated boundary layer would have penetrated a depth of about 90\%, for both assumptions. While, for the second model, essentially the same except \( P_s = 300 \) psi and aspect ratio 4.5:1, the estimated penetration is only about 40\%. The two-dimensionality can be improved by increasing both the depth to width ratio and the jet velocity.

3.7 Fluid Jet Amplifier Model

A fluid jet amplifier model was machined in Lucite with dimensions selected on the basis of the preceding investigations. The primary limitation on size was the availability of milling cutters. This dimension specified the minimum nozzle width and receiver width, and the maximum aspect ratio. The remaining dimensions picked on the minimum nozzle width are as follows.

\[
\begin{align*}
    b &= .014 \text{ in.} \\
    X &= 4b = .056 \text{ in.} \\
    Y &= 1.15b = .016 \text{ in.} \\
    \text{Aspect Ratio} &= 4.5:1 \\
    l &= 10b = .14 \text{ in.} \\
    t &= .1l = .014 \text{ in.} \\
    w &= .046 \text{ in.}
\end{align*}
\]

The supply pressure was selected as 300 psig. These conditions allow a much higher flow through the amplifier than would ordinarily be desirable, but it was tolerated in the hope that better manufacturing methods could produce a smaller similar device, that would be practical.

A photograph of the amplifier is in figure 3-24.

Lucite was chosen for the model when Brass proved too
FLUID JET AMPLIFIER PROTOTYPE

FIGURE 3-24
difficult to machine with the small milling cutters, and the plastic showed no perceptable wear after several hours of testing.

A plot of the demand flow as a function of control port pressure and pressure drop across the jet is given in figure 3-25. This curve was measured in the same manner described in section 3.3.

The blocked-load pressure recovery for each receiver was measured as a function of control pressure drop across the jet. This curve is plotted in figure 3-26 along with the pressure difference as a function of control pressure. From this curve, the linear region can be observed and the maximum gain can be determined. The value of \( w \) selected seems to be a little too large. If a smaller \( w \) were used, the receiver pressure curves would intersect at a higher level, thus increasing the useful range.

The amplifier appeared to operate satisfactorily, and provided the expected static characteristics. While the gain was low, this could be attributed to the fact that \( P_s = 150 \text{ psi} \) for the tests.
\[ \frac{\gamma}{b} = 1.15 \]
\[ \frac{x}{b} = 4.0 \]
\[ P_s = 300 \text{ psi} \]
\[ A_P = 0 = P_1 - P_2 \]
\[ \Delta P = +5 \text{ psi} \]
\[ \Delta P = +10 \text{ psi} \]
\[ \Delta P = -5 \text{ psi} \]
\[ \Delta P = -10 \text{ psi} \]
\[ Q_0 = 180 \text{ ml/min} \]

Figure 3-25

CONTROL PORT DEMAND CURVE

CONTROL PORT PRESSURE \( P_1 \) psig

CONTROL FLOW \( Q_1/Q_0 \)
RECEIVER and LOAD Pressures as a function of control pressure

Figure 3-26
IV SYSTEM
IV SYSTEM

The electro mechanical transducer, and the pure fluid amplifier have been designed and tested, and it remains to adequately couple the two elements and examine both static and dynamic operation. A sliding plate valve is designed and evaluated with each element as a proposed controller to transfer coil motion to a pressure difference. The complete assembled package is tested as a pressure control system and its shortcomings are evaluated.

4.1 Plate Valve

The function of this stage is to provide a spring force to restrain the voice coil, provide sufficient radial support for the coil, and to vary the area of a pair of orifices feeding the pure fluid stage in proportion to the actuator force. The spring rate is determined in Section 2 from the necessary displacement force relations of the moving coil. The variation in orifice area is determined from the conditions specified in Section 3.3. Because of the stability requirement (equation 3.10b) the orifice area can never be zero, hence this must be effectively an open center valve. Several types of valves could be proposed to meet these requirements, but the most promising from a fabrication standpoint is a sliding plate valve. The valve design is shown in figure 4-1. The two horizontal arms supporting the orifices have a reduced section at each end providing flexures so the motion of the valve is essentially parallel to the mounting block. Since the arms are spaced far apart, the stiffness
SLIDING PLATE VALVE

Figure 4-1
in the vertical direction, and torsional stiffness is considerably higher than the horizontal stiffness.

The orifices must provide a high gain (large change in area for a unit displacement) and yet must be easy to machine. A plain hole provides too low an area gain when the minimum area is considered, so slots with round ends that could be milled were selected. The slots in the sliding plate fall over another pair of slots in a receiver plate, but the spacing of the two pairs is different. The minimum area condition and area gain determines the size of the slots while the area at null determines the spacing.

Figure 4-2 gives the opening area as a function of displacement hole width and length. From figure 3-25 the minimum area can be computed, and allowing for leakage through the space between the plates the actual area can be determined. The maximum $\Delta x$ is .010 inches, so from figure 4-2 the orifice dimensions are:

$$R = .016$$
$$\frac{L}{R} = 4$$
$$x_u = .025$$

Where $x_u$ is the underlap or orifice centerline spacing at null.

4.2 Valve Plate - Moving Coil Combination

The valve plate as designed was constructed from Delrin to provide high strength to weight ratio. The valve was then coupled to the voice coil, and both static and dynamic tests performed as indicated in Section 2.4. Since the desired spring rate could not be designed into the plate, it was "tuned" to the necessary spring rate.
Figure 4-2

VALVE PLATE OPENING
A sinusoidal input current of 100 ma was provided to the coil, and the resulting displacement of the coil and valve measured. The reduced sections of the suspension arms were trimmed until the displacement was about .005 inches.

The dynamic response of the combination is that shown in figure 2-17. However, this is with the displacement instrumentation in place, and the effect is considerable. To ascertain the real natural frequency of the system two methods were employed. Since the system is very lightly damped, the natural frequency peak is easily observable. A second method also depends on the lack of damping, and the fact that at the natural frequency the velocity is high. Since velocity is out of phase with displacement and displacement is $\frac{T}{2}$ out of phase with input current, the velocity is in phase with the current. There is a "back emf." generated in the coil which is proportional to velocity, so if the current into the coil for a constant applied A. C. voltage is observed there is a marked change in impedance at the natural frequency. These two methods both indicate an actual natural frequency of about 210 cps.

Figure 4-3 gives the static characteristics of the valve coil combination. Except for some friction in the L. V. D. T. any hysteresis present was immeasurable. This friction problem was solved by adding mechanical dither during the measurements.

4.3 Load Matching

To evaluate the transfer function from the valve plate position to the fluid amplifier output, the relation between valve
INPUT OUTPUT STATIC CHARACTERISTICS
VOICE COIL AND VALVE PLATE

Figure 4-3
pressure-flow-displacement relation and the amplifier source characteristic are both extremely nonlinear, the best matching method is a graphical technique.

It is assumed that the flow through the orifice obeys the parabolic law

\[ Q = C_d A \sqrt{\frac{2(P_c - P)}{\rho}} \]  

(4.1)

where \( C_d \) is a constant. The control supply pressure \( P_c \) is assumed constant, and the value of \( A \) varies as in Section 4.1. The minimum value of \( A \) is the orifice opening area for maximum deflection one way plus leakage area. The leakage area is assumed constant and equal to the orifice perimeter times the plate spacing (.001 in.). The area for all subsequent deflections can then be calculated.

The source characteristics for the amplifier are those in figure 3-25 and both sides are assumed identical.

The following equations couple the two elements, where the orifices are considered separately with the subscripts 1, 2 referring to the separate orifices and control ports.

\[ -\Delta X_1 = \Delta X_2 \]  

(4.2)

\[ Q_{in} = Q_{demand} \]  

(4.3)

\[ \Delta P_1 = (P_1 - P_2) = -\Delta P_2 = -(P_2 - P_1) \]  

(4.4)

Equation (4.1) can be plotted on the source characteristics for the possible values of \( \Delta X \) to give the loci of possible stable
operating points where \( Q_{in} = Q_{demand} \). This is done in figure 4-4 for side 1, for \( P_c = 10 \) psi, and an exactly similar plot can be made for side 2. Figure 4-5 then gives the control pressure drop \( \Delta P_1 = P_1 - P_2 \) as a function of \( P_1 \) for each value of \( \Delta X \) from these loci. From this figure, using the relation in equation (4.4) a plot of \( \Delta P_2 \) as a function of \( P_2 \) for the values of \( \Delta X_1 \) can be constructed. This is done in figure 4-6 and superimposed on this is a plot identical to figure 4-5 for side 2, since the two sides are identical. Using equation (4.2) the intersection points then give \( \Delta P_2 \) or \( P_2 \) as a function of \( \Delta X_2 \). Figure 4-7 shows the final relation between \( \Delta P \) and \( \Delta X \) from the points just determined.

To complete the transfer relation, figure 4-7 can be combined with figure 3-26 as a figure 4-8 showing the predicted pressure displacement relation for the particular case that \( P_s = 300 \) psi and \( P_c = 10 \) psi.

### 4.4 Static Valve Test

The valve plate and fluid jet amplifier were combined and tested statically by indexing the valve plate manually. Figure 4-9 shows a photograph of the apparatus. The differential lead screw was used to position the valve plate by means of the music wire extended through the side of the device. The dial indicator gave a measure of the actual valve motion.

The receiver port pressures as a function of displacement are given in figure 4-10. The effect of changing the control supply pressure is indicated. From the previous section the value of \( P_c \)
MATCHING ORIFICE AND DEMAND FLOW CHARACTERISTICS

Figure 4-4

CONTROL PORT PRESSURE $P_1$ psig

CONTROL FLOW $Q_c/Q_o$

$\Delta P_1$ = 5

$\frac{X}{X_m}$ = -1.0

$\Delta P_1$ = -10 psi
LOCi OF POSSIBLE OPERATING POINTS
Figure 4-5

P CONTROL PORT PRESSURE  psi

ACTUAL OPERATING POINTS
Figure 4-6
PLATE VALVE DISPLACEMENT $X_m$

PREDICTED CONTROL PRESSURE AS A FUNCTION OF VALVE DISPLACEMENT

for

$P_s = 300$ psi   $P_c = 10$ psi

$X_m = .005$ in

Figure 4-7
Figure 4-8

Predicted load pressure as a function of valve displacement for

\( P_g = 300 \text{ psi} \quad P_c = 10 \text{ psi} \quad X_m = .005 \text{ in} \)
RAM TYPE LOAD

STATIC TEST APPARATUS

FIGURE 4-9
VALVE PLATE DISPLACEMENT \( X \) inches

RECEIVER AND LOAD PRESSURES AS A FUNCTION OF
VALVE DISPLACEMENT AND CONTROL PRESSURE

Figure 4-10
would seem to have little effect, and beyond a certain value would probably have no effect at all. While no saturation point was determined the small change in gain with $P_c$ was noted. Figure 4-10 also shows the gain, or load pressure, $P_{R1} - P_{R2}$ as a function of displacement.

There are two reasons for the difference between positive and negative displacements. First, one of the orifices in the valve plate was poorly made, and not only was it over size, but the metering edges were not straight or parallel. Second, when the plate is pushed it tends to lift up while when it is pulled it moves down because of a cocked linkage between the lead screw and valve.

The important design curves for a valve are the pressure-flow curves for a four-way configuration. These curves show the flow through the load as a function of pressure drop across the load, for a given displacement. The load is assumed to be purely resistive, and that all flow passing out of one side of the valve goes into the other side. This curve is measured by blocking the load flow with a variable orifice in series with a flow meter. The unusual thing noted when measuring these curves was that as the load was released, and the pressure drop decreased, $P_{R2}$ approached $P_{R1}$, where $P_{R2} < P_{R1}$. This would indicate that the primary flow limitation is the resistance provided by the receiver port entrance to returning flow. The complete pressure flow curves for one side are measured for $P_c = 15$ psi and $P_c = 10$ psi in figures 4-11 and 4-12.

These curves can be compared with previous semi-analytical and analytical ones. Figure 4-10 compares with that
FOUR-WAY VALVE CHARACTERISTICS for FLUID AMPLIFIER AND PLATE VALVE

LOAD PRESSURE $P_m$ psi

LOAD FLOW $Q_m$ gpm

$P_s = 300$ psig
$P_c = 15$ psig
$X_m = .005$ in

$\frac{X}{X_m} = 1.0$
FOUR-WAY VALVE CHARACTERISTICS for FLUID AMPLIFIER AND PLATE VALVE
predicted in Section 4.3. The pressure flow curves give results as expected from Section 3.5. Note that while the load pressure monotonically increases with displacement, the load flow only changes slightly, much as predicted in figure 3-16.
4.5 Complete First Stage

The entire package was assembled and tested for dynamic response. An exploded view of the package is shown in figure 4-13, and a photo of the assembled device is in figure 4-14. For testing purposes, both a receiver block and a ram type load were constructed to fit the valve output. The ram load was built to simulate a spool valve and is shown in figures 4-9 and 4-15. An L.V.D.T. is used to measure the position of the spool. Pressure transducers were used to pick off the output receiver pressures under blocked load conditions for dynamic response.

To determine the dynamic response of the plate valve and amplifier, the valve was stroked with a conventional torque motor and the blocked load pressure was measured as a function of valve position. This frequency response is shown in figure 4-16. The response seems to be that of a second order system with a damping ratio of about 0.7.

The frequency response of the system with the voice coil actuator in place was then measured. The frequency for a 90 phase shift between the current input and pressure output was about 12 cycles per second. The cause for this low frequency seemed to be the excessive amount of damping provided by the voice coil operating in hydraulic oil. The center pole of the magnet acts like a piston inside the coil, so there is a resultant flow proportional to the coil and plate velocity. The only exhaust for this flow is through the magnet gap clearances. Hence, there is a pressure drop across the coil, or a force proportional to velocity, adding to the damping term.
FIGURE 4-14

COMPLETE FIRST STAGE PACKAGE
Figure 4-16

FREQUENCY RESPONSE OF PLATE VALVE-AMPLIFIER AND COMPLETE SYSTEM

FREQUENCY RESPONSE

AMPLITUDE $R_m/R_{m0}$

FREQUENCY $f$ (cps)

PHASE ANGLE (degrees)

FREQUENCY $f$ (cps)
This problem was eliminated by drilling four holes in the front of the coil so the fluid could pass freely through. The frequency response was again measured, and is given in figure 4-16. The position of the ram load was identical to the pressure response, indicating that the natural frequency of the ram mass-spring is high and that the ram velocities are sufficiently small so that the flow influence is negligible.

While hysteresis was not present in either half of the device when tested separately, a marked amount was noted in the operation of the complete package. This is entirely due to friction effects. There is some viscous friction in the sliding plate valve, but probably only important at higher frequencies. A majority of the hysteresis is due to the moving coil sliding on the pole face of the magnet. The plate valve and connection means did not provide the required radial stiffness, to keep the coil centered. Furthermore, the coil seemed to enlarge slightly in oil despite the low absorbency property of Delrin, and the immersed D.C. magnet collected a substantial quantity of dirt on the pole faces adding to the friction problem.
V SUMMARY AND CONCLUSIONS
5.1 Summary

Electromechanical Transducer

The study of an electromechanical actuator using a constant magnetic field indicates that it is indeed possible to eliminate hysteresis in such a device. This gain, however, comes at the expense of force, for a given size and power input. So, for a specific displacement output, the response time is necessarily higher for a moving coil device that for a moving iron device. The manufacturing problems are different for this type actuator. Its simplicity plus the lack of electrical - mechanical balancing difficulties, would probably yield a much lower production cost. A different driving method is needed because of the single coil operation.

Fluid Jet Amplifier

A study of fluid jet amplifiers operating at low Reynolds numbers points out some practical limitations, and shows the degree of validity of validity of an analytical study. For a practical device, the efficiency and pressure recovery are low. This is anticipated from the turbulent model of the power jet, which seems to approximate the actual behavior, predicting a maximum pressure recovery of 50. Linearity is good, but the input impedance is low.

The single most important practical restriction is the minimum size that the nozzle and receiver port can be machined. As these dimensions decrease, the power consumption decreases with no loss in gain, and as the aspect ratio increases, the efficiency and predictability increase. If a method of manufacturing a small
nozzle width, high aspect ratio amplifier is perfected, two or more
could be staged to provide the needed pressure gain at a small power
loss. For example, a device with a nozzle width of .005 inches and
height of .025 inches would have about the same gain as the one in-
vestigated, but would draw only about 1/9 the power.

**System**

As a pressure control system, and with a ram type load, the package operated rather poorly. The power efficiency at full
stroke was extremely low, and the frequency response was consider-
ably lower than expected, although, comparable to response of a
typical servovalve. Due to stability requirements, the valve controlling
the input to the fluid jet amplifier must be open-centered and in fact
partially open for the full stroke, so the power consumed at the input
terminals is not negligible.

While the fluid amplifier cannot be compared with traditional
four way control valves, the complete package can. The maximum
ratio of load pressure to supply pressure at full stroke for this
scheme is about .5 while it is about .8 for a pair of flapper nozzles
and .9 for a jet pipe valve.

To obtain a maximum efficiency, the device should be
operated with a jet supply pressure only one order greater than the
control supply pressure, $P_c$, since the amplifier gain is independent
of $P_s$. This is done, however, at the expense of linearity. To
achieve a large load pressure under this condition requires that $P_c$
be large, and so the value forces become appreciable and the voice
coil motor would no longer be adequate.

Operation of the voice coil as a wet motor proved inadvisable since the additional damping was excessive. In addition, the dirt accumulation on the magnet face caused rubbing of the coil.

The device did prove to have adequately stiffness to drive a ram spring load. The jet deflection was insensitive to load pressure, indicating that no pressure feedback was present and that the device could be used for isolation purposes.
5.2 Conclusions

As a practical device, the system would have several shortcomings in a first stage servovalve application. The quiescent power loss is too high and the pressure recovery and power efficiency are too low. If, as mentioned in the previous section, a method of accurately manufacturing an amplifier with very small apertures is perfected, two stages of fluid amplification could be used to provide pressure gains approaching 100, with less power loss. This would be the factor determining its use in a servovalve.

As it now exists, the primary use of the package would be as an interface between electric and fluid systems. Of interest would be such systems as process control, or pure fluid logic and control systems. Process control systems using electronic data processing or data logging as control initiation methods, and pure fluid power control for diverting or actuation could use the servovalve as a coupling stage, and operate it at a lower pressure with smaller power loss. In pure fluid control systems such as aircraft pitch rate sensor and fluid logic networks, often an electrical signal is the only way to introduce an actuating signal, or a particular feedback loop. The electrohydraulic valve could then be used to provide a fluid signal proportional to the electrical signal.
APPENDIX A

DETERMINATION OF COIL WIRE SIZE
DETERMINATION OF COIL WIRE SIZE

Assume that a coil is composed of \( N \) turns of wire with diameter \( d \), that the shape of the coil is a rectangle \( a \times b \) revolved about an axis parallel to one of its sides, and that the mean diameter of the coil is \( D \). If the coil is potted it can be further assumed that the heat transfer from the coil is independent of the wire size.

For a given maximum heat generation \( W_m \), the maximum current through the coil is then

\[
I_m^2 = \frac{W_m}{R}
\]

\( R \) is the resistance of the coil, or in terms of resistivity \( \rho \)

\[
R = \rho \frac{l}{A} = \rho \frac{N\pi D}{\pi d^{3/4}}
\]

If the cross sectional packing density of the coil \( K \), or amount of wire per unit area, is independent of wire size, then

\[
N = \frac{Kac}{\pi d^{3/4}}
\]

This is a better approximation for \( d \ll a \).

The maximum force expected from the coil is then

\[
NI_m = N\sqrt{\frac{W_m}{R}}
\]

Using equation 3 and 4 gives

\[
NI_m = \sqrt{W_m} \cdot \sqrt{\frac{Kac}{\rho D}}
\]

and this is independent of wire size.
APPENDIX B

AMPLIFIER AND POWER SUPPLY
B.1 Amplifier

Input devices as signal generators and such generally have too low an output impedance to directly drive the input stage of a valve. Some type of amplifier is required to provide sufficient current, in phase with the voltage output of the signal source, to run the torque motor. There are several standard amplifiers for this purpose, all of which are either unsatisfactory or impractical for this project.

Typical servovalve amplifiers are push-pull vacuum tube devices. They are large and expensive, but well suited to the task of driving a conventional torque motor. Since a torque motor is a resistive load of several thousand ohms in series with an inductance in the henry range, the amplifier needs current feedback to keep the output current in phase. In lieu of current feedback a current source (pentode) is sometimes used. A distinct advantage to the torque motor is the split winding feature which makes push-pull operation much simpler from the amplifier standpoint. A typical circuit is shown in figure B-1.

A second type of amplifier, readily available, is the audio amplifier. This much better suited to the voice type load, but to obtain plus and minus output current unsatisfactory methods are usually employed. Transformer coupling is used either at the input stage, or at the coil output, as in figure B-2, thus rendering this type amplifier inadequate for D.C. operation. Or, as with the torque motor amplifier, the coil is split.

A proposed design to eliminate these problems uses a
TORQUE MOTOR AMPLIFIER

Figure B-1

AUDIO AMPLIFIER

Figure B-2
pair of transistors in a complementary-symmetry configuration. Transistors are much better suited to provide constant current, and because of a peculiarity in their manufacture this circuit is possible. There are two basic types of transistors NPN and PNP and they are complementary, so an input signal will have opposite effects on their conduction. In addition they are symmetrical so the collector currents are of opposite sign. If then, the collector currents are added, the result will be a positive or negative current proportional, and of equal sign, to the input signal applied to the base of the transistors.

The particular circuit adopted is shown in figure B-3. This circuit is adequate in the case that the inductance of the load can be neglected. When a voltage is applied to the input terminals, and hence to the bases of the transistors, the transistors conduct in their respective directions to make the emitter voltage relative to ground equal to the input voltage. The circuit has inherent voltage feedback, and if the load is purely resistive, this is the same as current feedback. If the inductance is in fact low, the load current is proportional to input voltage as desired. Because of the feedback feature this circuit is essentially independent of the specific transistor properties. This is beneficial from a cost standpoint and often it is impossible to exactly match a PNP and NPN type semiconductor. One transistor property that is important is the emitter-base voltage, as this produces a steady error between input and output. This small voltage, however, can be adequately biased out. A potentiometer is in the base circuit to provide bias.
COMPLEMENTARY - SYMMETRY AMPLIFIER

Figure B-3

AMPLIFIER WITH FEEDBACK

Figure B-7
This resistance is varied so that no output signal occurs when the input is shorted.

Figure B-4 shows the frequency response of the amplifier, where amplitude ratio is the ratio of output current to output at D.C. This data was taken with a load of 47 ohms and 2 millihenries, a typical voice coil. Figure B-5 gives the input-output characteristics of the amplifier. The measured input impedance is about 880 ohms. This can be increased at the expense of "gain", or an additional similar stage of amplification can be added. An illustration of the amplifier and power supply is in figure B-6.

For a coil in which the inductance is no longer negligible, a circuit with current feedback must be used. One suggested circuit is given in figure B-7.
Figure B-4

AMPLIFIER FREQUENCY RESPONSE

FREQUENCY (cps)

PHASE ANGLE (degrees)

AMPLITUDE RATIO
AMPLIFIER INPUT OUTPUT CHARACTERISTICS

Figure B-5
SERVO TESTER

D.C. MAGNET
POWER SUPPLY

SIGNAL INPUT

AMPLIFIER

TRANSISTOR AMPLIFIER
and
POWER SUPPLY

FIGURE B-6
B. 2 Power Supply

A power supply was needed to provide a constant D. C. current at low voltage for energizing the D. C. magnet. A circuit that performed satisfactorily is given in figure B-8. A diode bridge provides a full wave rectified D. C. voltage, supplying a pair of transistors in an emitter follower configuration. This configuration has a voltage gain of 1 so the emitter voltage across the 2 ohm resistor is determined by the setting of the potentiometer. This voltage in turn is regulated by the Zeener diode. The collector current is approximately equal to the emitter current, so the load current too, is a function of potentiometer setting.

The voltage supply circuit for the amplifier is included for reference purpose in figure B-8.
Figure B-8

AMPLIFIER

D.C. MAGNET POWER SUPPLY

PANEL MARKING
O BANANA JACK
Bibliography


