A FLUIDIC CONTROL SYSTEM
APPLIED TO
ROCKET PAYLOAD ROLL CONTROL

A Thesis
Submitted to the Faculty of Graduate Studies
in Partial Fulfilment of the Requirements
for the Degree of
Master of Science
in the Department of Mechanical Engineering
and the Division of Control Engineering
University of Saskatchewan

by

Lawrence R. Stevens

Saskatoon, Saskatchewan.
April, 1967.

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ABSTRACT

The problem of automatic roll control of a partial payload of an upper atmosphere sounding rocket is discussed. A control system is proposed for accomplishing this function which is entirely fluidic, from sensing the necessary variables to applying the required correction torques. The design of the system, excluding the output power amplifiers and actuators, is described in detail.

The system is implemented in the form of a demonstration model to illustrate the design techniques. The results of experiments carried out on this model are presented and discussed.
ACKNOWLEDGEMENTS

The author wishes to express his gratitude and appreciation to Dr. J.N. Wilson for his assistance and guidance in the preparation of this thesis. The assistance of Professor R.W. Besant is also gratefully acknowledged.

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<td>$d_n$</td>
<td>nozzle diameter; in.</td>
</tr>
<tr>
<td>$e$</td>
<td>base of natural logarithms</td>
</tr>
<tr>
<td>$F$</td>
<td>force; lb.</td>
</tr>
<tr>
<td>$F_m$</td>
<td>maximum thruster force; lb.</td>
</tr>
<tr>
<td>$G(s)$</td>
<td>transfer function</td>
</tr>
<tr>
<td>$I_p$</td>
<td>roll moment of inertia of payload; in$.lb.\cdot sec.^2$</td>
</tr>
<tr>
<td>$K$</td>
<td>gain constant</td>
</tr>
<tr>
<td>$L$</td>
<td>nozzle to flapper quiescent gap setting; in.</td>
</tr>
<tr>
<td>$L'$</td>
<td>moment arm; in.</td>
</tr>
<tr>
<td>$\Theta$</td>
<td>angular displacement; radians</td>
</tr>
<tr>
<td>$\dot{\Theta}$</td>
<td>angular rate; radians/sec.</td>
</tr>
<tr>
<td>$\ddot{\Theta}$</td>
<td>angular acceleration; radians/sec.$^2$</td>
</tr>
<tr>
<td>$\Theta_m$</td>
<td>angular displacement value which causes thruster saturation $(\dot{\Theta} = 0)$; radians</td>
</tr>
<tr>
<td>$\dot{\Theta}_m$</td>
<td>angular rate value which causes thruster saturation $(\Theta = 0)$; radians/sec.</td>
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<tr>
<td>$P_n$</td>
<td>nozzle upstream pressure; psi.</td>
</tr>
<tr>
<td>$P_s$</td>
<td>supply pressure; psi.</td>
</tr>
<tr>
<td>$S$</td>
<td>laplace operator</td>
</tr>
<tr>
<td>$t_c$</td>
<td>time constant; sec.</td>
</tr>
<tr>
<td>$t_o$</td>
<td>transport lag; sec.</td>
</tr>
<tr>
<td>$T$</td>
<td>torque; lb.in.</td>
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<tr>
<td>$\omega_n$</td>
<td>undamped natural frequency</td>
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<tr>
<td>$X$</td>
<td>flapper to nozzle displacement from quiescent position; in.</td>
</tr>
<tr>
<td>$y$</td>
<td>cam to nozzle distance; in.</td>
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<tr>
<td>$\zeta$</td>
<td>damping ratio</td>
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1. INTRODUCTION

The purpose of the work outlined in this thesis is basic research into fluidic systems design. Such a purpose is immense in its scope, especially when one considers the relative lack of available literature in this area at the present time. For this reason a particular fluidic control system is chosen as a goal and an attempt is made to use what has already been learned as guidance towards a better understanding of the principles and problems involved in the design process. The result is implemented in the form of a demonstration model which is used to verify the validity of the design techniques and procedures developed.

Many of the phenomena discussed herein are not as yet fully understood; as a consequence, each could easily be pursued independently at great length. This thesis therefore, does not pursue any one phenomenon in detail, but instead is a balance between analytical prediction, basic gross understanding, and undeniable experimental facts; which, when integrated, yield the most fruitful insight into the overall design problem.

To the author's knowledge there have been several papers published on fluidic feedback control systems. The system developed in this thesis differs in its application and in the exclusive use of proportional elements. Also the design techniques are considered unique as applied herein.

Since the purpose of the present investigation is the development

* -- Numbered superscripts refer to List of References.
and study of the fluidics involved in a feedback control system, it is relatively immaterial as to what particular system it is applied. However, it is always more intuitively acceptable to have a specific function in mind during the research and development process. During the initial stages of planning this program, it was learned that the Division of Space Engineering of the University of Saskatchewan would be interested in the development of an automatic roll attitude control system for a partial payload of an upper atmosphere sounding rocket. This application was considered ideal for a fluidic system study. The advantages of fluidics in this application would be: low cost, light weight, inherent simplicity and good reliability. The system as proposed would be entirely pneumatic, from sensing the relative parameters, to controlling the outputs of cold gas thrusters. This thesis will deal with that portion of the system which senses the parameters and produces an output error signal with the correct relation to these parameters. The power gain stages, the thrusters and the power supply, are the subjects of another thesis 4.

The mode of operation of the proposed control system as applied to the probe vehicle is as follows:

1) The boost phase proceeds as normal, the payload being rigidly connected to the rocket and the control system quiescent. A normal boost phase begins with a high level thrust, rail guided launch. The rocket is guided strictly by aerodynamic stabilizing forces upon leaving the launch rail until some rate of spin is achieved\(^*\), whereupon the gyroscopic effect of the induced spin provides additional directional stability.

\(^*\) Spin is induced aerodynamically by fin angle set.
2) The angular position reference gyro is functioning and positioned to the final required orientation before launch. It would be uncaged immediately prior to launch thus maintaining the reference orientation. At burn-out the vehicle has attained some prescribed velocity and spin rate. It continues in a constant directional attitude with respect to inertial space, on a ballistic trajectory through the near vacuum of the outer atmosphere*. 

3) That portion of the payload which is to be position controlled would now be separated from the main rocket body except for a rigid low friction bearing connection whose axis is coincident with the longitudinal roll axis of the rocket.

4) The angular position control system would be activated, de-spinning the controlled portion of the payload and orienting it to the desired angular position.

The technique of roll controlling a portion of the payload while maintaining its longitudinal axis coincident with that of the rocket, simplifies the control system considerably. It means, in effect, that only one degree of freedom need be controlled. It is assumed that the spinning rocket has negligible coning and is stably oriented in space during the all-important coast phase. The coast phase is defined as the time between burn-out and re-entry into the atmosphere. It is during this phase that the major portion of the payload experiments are accomplished. The past performance of these rockets has shown that the above mode of operation is entirely feasible and consequently is assumed for the remainder of this thesis.

* -- See Appendix B for rocket trajectories.
The roll stabilization of the payload package as outlined above is undertaken for one reason; to provide a relatively stable platform from which photometric type instruments (or others) may view a particular area of the night sky. It is presumed that the area of interest contains an auroral display or similar phenomenon as determined immediately prior to launch. The instruments in the roll stabilized package would then be viewing the area of interest during the full coast phase of the flight rather than a fraction of the available time as would be the case if the payload remained spinning. Since an increase in experiment time is gained through the use of the roll control system, the economic justification of its use would be dependent upon a compromise between the control system cost and the cost of gaining the equivalent additional experiment time through the use of additional rockets. There are other considerations of course, but this one alone is sufficient to justify the development of a roll control system such that it would be available when circumstances favor its utilization. The above justification is independent of the fact that the primary object of the project is research into fluidic systems design. The accuracy requirements for the photometric type application of the system are fairly broad, they are: a maximum angular error of roughly ±2 degrees. This lenient performance demand is in accordance with the first-generation nature of the project.

It is informative at this point to insert a brief discussion of a typical rocket vehicle of the type being considered. The Black Brant IIA is the booster presently used by the Division of Space Engineering of the University of Saskatchewan for upper atmosphere studies. The standard Black Brant IIA is 17.2 inches in diameter, 27.5 feet long, weighs about 2775 pounds at launch and 995 pounds at burn-out. The nose cone including
payload and ballast is approximately 300 pounds. The maximum altitude
reached is of the order of 500,000 feet. For a more detailed
description of the vehicle and its characteristics see Appendix B.

In summing up the aims of this thesis it is sufficient to state
them in order of relative importance as follows:

1) To study the problems, and their solutions, encountered in the
design of a fluidic feedback control system.

2) To investigate the feasibility of the application of fluidics
to the roll control of a partial payload of an upper atmosphere sounding rocket.
2. INVESTIGATION OF PAYLOAD DYNAMICS

The mode of operation of the roll control system as outlined in the introduction, implies that missile stability is entirely dependent on aerodynamic and ballistic properties. In other words, the proposed control system does nothing to assist missile guidance or control, but functions independently as an integral part of the payload. It should be interjected at this point, that the despinning of the payload detracts negligibly from missile stability. This contention is upheld by a cursory examination of the rocket specifications as presented in Appendix B. Here it is found that the moment of inertia* for the burned out rocket is approximately 240 in.lb.sec², while that for the payload is only 7 in.lb.sec².

Thus the proposed control system is strictly a contribution to payload function. It will be assumed that activation of the control system has no effect on the missiles trajectory or motion. Also, such effects as missile "coning" and resultant "whip" or flexure of the connecting shaft between rocket and payload, are not considered. It is assumed that these effects are small enough to be neglected. These assumptions and those that follow are in accordance with the basic purpose of this thesis as outlined in the introduction.

The purpose of the investigation carried out in this chapter, is to determine by actual simulation on an analog computer, the required relationship between the control system sensors and thrusters, which provides the best overall performance. As will be shown, this amounts to

---

* -- Along the longitudinal axis.
selecting the most desirable values for the gains of the rate and
position feedback loops. The selection is based on the payload settling
times for some worst case initial conditions.

The assumptions made earlier simplify the simulation considerably.
The problem is reduced to solving a one dimensional second order
differential equation. The equation of motion for the roll axis of the
spinning payload is

\[
T = I_p \ddot{\theta} 
\]

where, \( T \) = applied torque
\( \theta \) = angular displacement from some reference
\( I_p \) = moment of inertia of payload about the roll axis

At this point an additional assumption is introduced. The dynamic
bearing friction acting between rocket body and payload, is assumed to
produce a negligible torque on the payload. The torque available from the
control system thrusters, as will be proposed later, has a maximum value
of 72 in.lbf. This value is several orders of magnitude above the high-
est frictional torques one could expect from a modern ball bearing.
Thus, any torques applied to the payload about its roll axis are due
e ntely to the control system thrusters, since there is no aerodynamic
drag effects during the coast phase of the flight. The resultant torque
applied to the payload can therefore be expressed as;

\[
T = F L' 
\]

where, \( F \) = tangential force due to thrusters
\( L' \) = moment arm of thrusters
The thruster force $F$ is required by the present system to be some function of angular displacement and rate. It is the choice of this function which constitutes the present problem. The form of the curve chosen to represent the relationship between $F$ and $\theta$, or $F$ and $\dot{\theta}$, must be of such a nature as to be reproducible by the control system. The most desirable choice and, as it turns out, most practical one is a straight line relationship. Therefore, a linear relationship is assumed; the justification being the relatively large linear regions in the describing characteristics of the amplifiers to be used in the control system.*

When the maximum output of the thrusters is reached ($F = F_m$), the system saturates, introducing a non-linearity into the loop. The assumed form of the $F$ vs $\theta$ and $F$ vs $\dot{\theta}$ relationship is shown in figure 2.1. These two curves are summed along the ordinate to obtain the required $F$ vs ($\theta + \dot{\theta}$) characteristic family shown in figure 2.2. This family of curves represents the $\theta$ vs $F$ relationship for various values of constant $\dot{\theta}$. It thus shows the required value of $F$ for any combination of $\theta$ and $\dot{\theta}$, and includes the saturation effects mentioned earlier. Since $F_m$ is a constant, the present investigation will consist of a study of the effects on performance due to variations in the parameters $\theta_m$ and $\dot{\theta}_m$, which determine the slope of the respective curves in figure 2.1. $\theta_m$ and $\dot{\theta}_m$ are defined as the values of position error and rate error, respectively, which alone (other error zero) will cause saturation of thruster output.

From the above assumptions, the following equation represents the linear region of the $F$ vs ($\theta + \dot{\theta}$) relationship:

* -- These amplifiers and characteristics are described in chapter 4.
FIG. 2.1 - 'ASSUMED RELATIONSHIPS BETWEEN THRUSTER FORCE "F" AND ERROR SIGNALS "\( \theta \)" & "\( \dot{\theta} \)"

FIG. 2.2 - 'ASSUMED RELATIONSHIP BETWEEN THRUSTER FORCE "F" AND SUMMED ERROR SIGNAL "(\( \theta + \dot{\theta} \))"
\[ F = f(\theta, \dot{\theta}) = (F_{m}/\theta_{m}) \theta + (F_{m}/\dot{\theta}_{m}) \dot{\theta} \quad 2.3 \]

where \((F_{m}/\theta_{m})\) and \((F_{m}/\dot{\theta}_{m})\) represent the gains of the position and rate feedback respectively and \(F\) is limited to \(\pm F_{m}\).

A short note is injected at this point concerning sign convention. It is immaterial to system performance as to which direction of angular rotation is considered to be positive. However, this choice must be consistent with the remaining variable's sign convention. That is, if clockwise rotation is chosen to produce positive angular displacement, then clockwise acceleration must be positive and it follows that positive \(F\) produces positive acceleration. The \(F\) defined by equation 2.3 therefore, represents a positive feedback. This situation is unacceptable in the control system for obvious reasons and necessitates the inclusion of the negative sign which appears in the following equation.

Combining equations 2.1, 2.2 and 2.3, one obtains;

\[ -\ddot{\theta} = (I/I_{p}) \left[(F_{m}/\theta_{m})\theta + (F_{m}/\dot{\theta}_{m})\dot{\theta}\right] \text{ limit to } \pm F_{m} \quad 2.4 \]

where the term in brackets is limited to \(\pm F_{m}\). Equation 2.4 represents the equation of motion of the payload about its spin axis under control of the proposed control system. In the linear region of operation (no limiting) this equation has the form;

\[ \ddot{\theta} + 2\dot{\omega}_{n}\dot{\theta} + \omega_{n}^{2}\theta = 0 \quad 2.5 \]

That is, equation 2.4 can be written as;

\[ \ddot{\theta} + 2(I/2I_{p})(F_{m}/\dot{\theta}_{m})\dot{\theta} + (I/I_{p})(F_{m}/\theta_{m})\theta = 0 \quad 2.6 \]

Thus, by comparison of equations 2.5 and 2.6, one obtains;
\[ f = \left( \frac{I_F}{I_p} \right)^{\frac{3}{2}} \left( \frac{2\dot{\theta}_m}{I_p} \right) \]

and;

\[ \omega_n = \left( \frac{I_F}{I_p} \right)^{\frac{3}{2}} \]

The above equations are useful for predicting the parameter changes necessary to achieve the desired results when solving equation 2.4 on the analog computer. They are also helpful in gaining an insight into the problem in general. It should be noted however, that equations 2.5 through 2.8 apply only in the linear region of operation.

Based on the physical characteristics of the Black Brant IIA vehicle, the following payload parameters were chosen as typical for the purpose of this investigation;

- Payload moment of inertia, \( I_p = 7 \text{ in.} \text{lbf.} \text{sec}^2 \).
- Thruster moment arm, \( L' = 9 \text{ in.} \).
- Maximum thrust (two nozzles), \( F_m = 8 \text{ lbf}. \)

Equation 2.4 was instrumented on the analog computer using a "hard limiter"* to achieve the required discontinuity. Normalized variables and computer time scaling were used. The maximum values of the variables for the purpose of scaling were set at;

\[ \dot{\theta}_{\text{max}} = 100 \text{ rad./sec}^2. \]
\[ \dot{\theta}_{\text{max}} = 6 \text{ rad./sec}. \]
\[ \theta_{\text{max}} = 3 \text{ rad}. \]

Substituting the assumed payload characteristics into equation 2.4 and scaling as above, the result is;

* -- See reference 8, page 372, for description of "hard limiter" used.
\[
- \left( \frac{\theta}{100} \right) = 0.103 \left[ \frac{3}{\theta_m(\theta/3)} + 6/\dot{\theta}_m(\theta/6) \right] \text{ limit to } +1 - 2.9
\]

where \( \theta_m \) and \( \dot{\theta}_m \) are parameters.

The computer diagram for the instrumentation of equation 2.9, is shown in figure 2.3. A Pace TR-48 analog computer was used for the simulation. The procedure for solution was to choose some worst case initial conditions, solve the equation for varying values of the parameters \( \theta_m \) and \( \dot{\theta}_m \) for these initial conditions and compare the solutions. The solutions were in the form of time base recordings of \( \theta \) and \( \dot{\theta} \). The recordings were compared on the basis of settling times and damping.

The results from two of these solutions are given in figures 2.4 and 2.5. Those shown in figure 2.5 displayed the most satisfactory performance as determined by this simulation. Figure 2.5 corresponds to \( \theta_m = 0.4 \) radians, and \( \dot{\theta}_m = 0.6 \) rad./sec. These values are therefore chosen as guides for the design of the control system and will be referred to frequently in the following chapters.
Note: \( n = \text{time scale factor} \)

FIG. 2.3 - ANALOG COMPUTER DIAGRAM OF CONTROL SYSTEM SIMULATION
\[ -\dot{\theta} = \frac{9}{7} \left( 4\theta + 20\dot{\theta} \right) \quad \text{Limited } t_0 \pm 8 \]

**FIG. 2.4 - SIMULATED RESPONSE OF PAYLOAD FOR**

- \( \theta_m = 2.5 \text{ radians} \)
- \( \dot{\theta}_m = 2.0 \text{ radians/second} \)

Note:
- \( F_m = 8 \text{ lbf.} \)
- \( I_p = 7 \text{ lbf. in. sec.}^2 \)
- \( L = 9 \text{ in.} \)
- \( \theta_m = 2.5 \text{ rad.} \)
- \( \dot{\theta}_m = 2 \text{ rad/sec} \)
$-\ddot{\theta} = \frac{9}{7} [12\theta + 20\dot{\theta}]$ Limited to $\pm 8$

Note: $F_m = 8 \text{ lbf.}$
$I_p = 7 \text{ lbf. in. sec.}^2$
$L = 9 \text{ in.}$
$\theta_m = 0.4 \text{ radians}$
$\dot{\theta}_m = 0.6 \text{ rad./sec.}$

FIG. 2.5 - SIMULATED RESPONSE OF PAYLOAD FOR
$\theta_m = 0.4 \text{ radians}$
$\dot{\theta}_m = 0.6 \text{ radians/second}$
3. FLUIDIC IMPLEMENTATION METHODS

In any system design, a major decision which affects the resultant configuration to a very large extent, is the choice of active components. Such a decision is in turn controlled by the implementation methods utilized. As a consequence, these decisions are of paramount importance in the systems design process, and will therefore be discussed in some detail in this chapter.

3.1 Mode of Operation

The choice of analog vs digital implementation is dependent upon which method provides the least complicated and most reliable system, while still meeting the accuracy requirements. The accuracy requirements as outlined in the introduction, are a maximum angular error of roughly ± 2 degrees.

Digital implementation is considered first. A digital system can theoretically provide any degree of accuracy at a cost of more components and system sophistication. A block diagram of a basic position control system, with rate feedback, is shown in figure 3.1. In order to execute such a system digitally, the first requirement would be digital rate and position sensors. Since gyros are used as rate and position references*, then some manner of digital pickoff must be used for sensing the respective gimbal positions. This could consist of a pulse generator which produces a pneumatic pulse for every degree of angular change. A polarity signal must accompany the resulting pulses, so that they may be stored in an up-down binary counter. The counter must be capable of

* -- As outlined in chapter 5.
FIG. 3.1 - BASIC POSITION CONTROL SYSTEM
storing a count of 360 and have the ability of going "around the end" in either direction. That is, on the 361st pulse, the counter must count 1, or on the -1st pulse it must count 360. Such a counter is necessarily complex, as is the pulse generator that drives it. The position sense counter must have a minimum of nine bistable elements in order to store a count of 360. The number of elements rapidly swells as more of the required circuitry is considered. A similar but even more sensitive device is required for sensing the rate. The complexity of digital implementation, as implied here, provided motivation for an examination of analog techniques at this point.

For the analog implementation, the position and rate sensing would be accomplished using nozzle-flapper techniques. The output signals from the nozzle-flapper sensors, would be amplified by proportional fluidic amplifiers. The summation of the resulting two signals would be accomplished by a third proportional amplifier. The output of this amplifier is the required error signal; assuming that the sensitivity of the sensors is in accordance with theoretical demands. The error signal is then amplified to a level sufficient to drive a pair of cold gas thrusters directly.

Either the digital or analog method is feasibly applicable to the fluidic roll control system. However, the analog techniques produce a much simpler system than the digital. Thus, if analog techniques can be successfully applied to this problem, with performance within the limits stated earlier, the resulting system would be much superior from an engineering standpoint than an equivalent digital system. It is determined therefore, that analog devices be used throughout.
3.2 Active Fluidic Component

With the decision being made to design an entirely proportional system, the choice of an active fluidic device is considerably narrowed. Even so, there are several devices to be considered. The vortex amplifier* can be ruled out immediately due to its characteristic low pressure gain. Pressure gain is important in the roll control system application due to the large expected discrepancy between input and output pressures. In this respect, the transverse impact modulator has very desirable characteristics for the present system, but was unavailable from the manufacturer. The development of a transverse impact modulator was considered, but soon rejected as too ambitious a project to be undertaken as a part of this thesis. Continuing with this manner of reasoning, the field of proportional amplifiers was narrowed to one. The beam deflection fluid amplifier was both readily available and had good operating characteristics for the present system.

A variety of beam deflection fluid amplifiers were commercially available at the time this project was undertaken. After a survey of the various configurations, the author decided to utilize one which shall be hereafter referred to as the 1300POI proportional amplifier (manufactured by Aviation Electric Ltd., Montreal). The characteristics of this device are crucial to the design of the entire system; consequently, they are discussed in detail in the following chapter.

* -- See reference 5 for a description of the various fluidic devices.
The proportional beam deflection amplifier was considered desirable for this project due to both its availability and performance. The empirically determined characteristics of this amplifier affect the design to such an extent as to merit a chapter devoted to their description.

This amplifier was found to provide equivalent, and in some cases superior, pressure recovery and gain as compared to other similar available devices. Also, tests indicated satisfactory performance with supply pressures as high as 50 psi. This latter characteristic is advantageous, since, with an entirely fluidic system, it is desirable to have as high an output pressure from the control system as possible, to facilitate the driving of the power stages controlling the torquing thrusters*. This amplifier also has the advantage of a unique vortex vent in the output ports. This feature effectively decouples the input ports from the outputs, providing excellent isolation of input characteristics from downstream loading. Thus, the vortex vents make staging or matching to various loads a relatively straightforward proposition. A photograph of the amplifier, showing the vents is shown in figure 4.1.

A static transfer curve as provided by the manufacturer, is shown in figure 4.2. The common practice of scaling device characteristics in a nondimensional manner**, holds only for low supply pressures (eg. less than 10 psi.). At higher pressures the characteristics vary with supply pressure (Ps). As a consequence, it is necessary to empirically determine

* -- The power stages and thrusters and power supply for the proposed system are the subject of another thesis -- see reference 4.

** - ie. by normalizing with respect to supply conditions.
Fig. 4.1 - The 1300P01 Proportional Amplifier
Differential Control Pressure (% of supply)

FIG. 4.2 - TRANSFER CHARACTERISTICS OF THE 1300P01 PROPORTIONAL AMPLIFIER (as provided by manufacturer)
the performance for each of the supply pressures anticipated. The input and output characteristics of the 1300PO1 proportional amplifier are given in figures 4.1, 4.5 and 4.6, for supply pressures of 10 and 50 psi. These curves were found to be sufficient for static design of the overall system. A supply port characteristic is also shown in figure 4.3 and is useful for estimating fluid power requirements.

The 1300PO1 proportional amplifier displays an interesting (and as yet unexplained by the author) phenomenon in its output characteristics, for a supply pressure of 50 psi. As seen from the output characteristic family for this amplifier and supply pressure, (figure 4.6) there is a distinct negative resistance region near the center of the zero differential input curve. It is beyond the scope of the present work to investigate the cause of this effect. However, the existence of the phenomenon was verified and the two quasi-stable states demonstrated by constructing an oscillator through the simple expedient of loading an active amplifier in the negative resistance region. Needless to say, in all of the amplifiers used in the present system, this region of the characteristic is meticulously avoided.

The dynamic characteristics of the 1300PO1 proportional amplifier are shown in figure 4.7. This figure depicts the frequency response in the form of pressure gain and phase lag vs frequency. The format is the conventional logarithmic scales for frequency and linear scales for gain in decibels and phase lag. Judicious curve fitting to match the experimental points, provided the following transfer function:

\[
G(S) = \frac{K e^{-t_c S}}{1 + t_c S}
\]
FIG. 4.3 - SUPPLY CHARACTERISTIC FOR THE 1300P01 PROPORTIONAL AMPLIFIER
FIG. 4.4 - INPUT CHARACTERISTICS FOR THE 1300P01 PROPORTIONAL AMPLIFIER

Note: - curves include only points within operational range
      - experimental points

Flowrate, $Q_{c2}$ (ft$^3$/sec $\times 10^{-4}$)

Pressure, $P_{c2}$ (psi)

$P_s = 10\text{ psi}$, $P_{c1} = 2.11\text{ psi}$

$P_s = 50\text{ psi}$, $P_{c1} = 5\text{ psi}$
FIG. 4.5 - OUTPUT CHARACTERISTICS FOR THE 1300P01 PROPORTIONAL AMPLIFIER
FIG. 4.6 – OUTPUT CHARACTERISTICS FOR THE 1300 P01 PROPORTIONAL AMPLIFIER
FIG. 4.7 - FREQUENCY RESPONSE
of the
1300PO1 PROPORTIONAL AMPLIFIER
with, \[ t_0 = 0.25 \times 10^{-3} \] seconds,
\[ t_c = 2.27 \times 10^{-3} \] seconds.

Considerable reference is made to the above static and dynamic experimental results in the remainder of this thesis. The experimental methods employed for obtaining this information are outlined in appendix A.
5. **PNEUMATIC SENSOR STUDY**

The basic control system under present investigation, consists essentially of three main sections; the sensors, the computational facility and the actuators. It is the sensors which are of concern in this chapter. To be sensed, in this instance, are the parameters necessary for adequate angular position control of the payload package. As indicated in the previous chapter, it is sufficient for the present system to have an instantaneous knowledge of angular position and rate, with respect to the earth*. This requires the payload to have either an internal inertial reference, or some means of observing an external reference (ie., sun, star, horizon, etc.). Since the system is to be entirely pneumatic, the observation of an external reference is ruled out** as it would involve electromagnetic detectors of some description. Also, a very undesirable electrical to pneumatic interface would be required which would almost certainly involve some inertia terms. The use of an internal reference is more practical, from a fluidics standpoint.

Fluidic vortex angular rate sensors have been developed and shown to be highly successful. However, to obtain angular position using such a device, an integration would be required. Integration by fluidic means is a complex and not too accurate process by the best available techniques. When it is considered that the mode of operation outlined in the introduction, calls for the payload to spin with the missile during the boost phase, the

---

* -- Actually, an inertial reference is used, but considering the short duration of the flight it is not improper to consider it as an "earth reference".

** - Within the scope of present day fluidics state-of-the-art.
resultant large integration and associated errors make this approach unfeasible.

The only practical method remaining for achieving a reference is the use of pneumatically driven gyros with some manner of pneumatic sensing for determination of the gyro axis position with respect to the payload. This last approach is the one used in this thesis.

The use of gyros, as justified in the foregoing paragraphs, presents a major problem in the fluidic sensing of the relative gimbal positions. Also, two gyros must be used, one for rate, and one for position, to avoid the necessity of an integration or differentiation. The relative high cost of a modern gyro adds an economic disadvantage. This is somewhat lessened however, if it is recalled that the entire flight length is approximately six minutes, thus placing a relatively low demand on drift rate and consequential gyro sophistication. There remains the problem of sensing the relative gimbal positions.

The original approach to the position sensing problem, was to use electrical readout from the gyros and an electrical to pneumatic interface. This approach was abandoned as being unnecessarily complex. The system under present investigation involves control of one axis. Thus, since the position information required for single axis control can be obtained from the outer gimbal of the gyro, no slip rings are required. As a consequence, a pure fluidic sensor becomes feasible. It should be stressed here that pneumatic sensing of the inner gimbal position is not considered impossible but would likely be of such mechanical sophistication as to be beyond the capabilities of the fabrication facilities available to the author.

The nozzle-flapper principle was employed for the pneumatic sensing
of both position and rate. This principle is based on the variable restrictive properties of a nozzle placed varying distances from a plane obstruction (flapper). The idea has been used to good advantage for several decades in hydraulic and pneumatic servomechanisms and also in pneumatic process control instrumentation. The principle is also used in pneumatic micrometers or inspection gauges. The nozzle-flapper lends itself ideally to the position and rate sensing problem due to its flexibility, low power consumption and high sensitivity. Although much used, there is little analytically that one can accomplish in assisting the design of this or any but the simplest of fluidic devices. Pure fluid technology is based on some of the most sophisticated fluid dynamic phenomena encountered in the scientific world. The basic equations of motion of a viscous time-dependent flow, the Navier-Stokes equations, are non-linear and difficult to solve analytically even for the simplest two dimensional, incompressible case. The result is, that a number of assumptions and approximations are usually made, consistent with empirical data, to simplify the descriptive equations of any particular fluid-state phenomenon. The case at hand, the nozzle-flapper principle, is no exception. The purpose of this thesis is basic research into the design problems associated with the development of a particular fluidic control system. Consequently, no significant effort is made to analytically describe the system. Some graphical design techniques used in the following chapter will be familiar to those experienced in electronic design and are shown to be equally applicable to fluidics.

A sketch of the basic nozzle-flapper configuration is shown in figure 5.1. As can be seen the nozzle-flapper acts as a variable restrictor, which when combined with the upstream restrictor, provides a
Restrictor

Note: $d_n =$ Nozzle Diameter

FIG. 5.1 - NOZZLE - FLAPPER CONFIGURATION

Flat-Faced

FIG. 5.2 - NOZZLE GEOMETRIES
variable signal pressure $P_n$, the amplitude of which is some function of the flapper to nozzle distance $(L + X)$. There are basically two nozzle geometries commonly recognized; they are, sharp edged and flat faced*. A sketch of these two geometries is shown in figure 5.2. In the present instance, the former configuration is used due to its ease of manufacture and repeatability of flapper force characteristics.

The operational range of the flapper in figure 5.1 is from, $X = -L$, to some saturation value, where further increases produce little useful effect and can be said to be impractical. A good criteria for the determination of this saturation point, is that it occurs for that value of $X$ where the curtain-shaped gap area between nozzle and flapper equals the nozzle area$^6$. That is, for:

$$\pi d_n(L + X) = \pi l (d_n)^2$$

or,

$$L + X = \frac{d_n}{4}$$.  - - - - - - - - - - - - - - - - - - - 5.1

A typical pressure ($P_n$) vs flapper distance $(L + X)$ characteristic is given in figure 5.3. The nozzle diameter ($d_n$) in this case is 0.040". Thus equation 5.1 predicts saturation at, $L + X = 0.010"$. A cursory inspection of the characteristic in figure 5.3 shows this to be essentially the case. It is also noted that the curve is non-linear except for small values of $(L + X)$. For this example, linearity can be assumed in the range, $0 < (L + X) < 0.003"$, or 30% of the above predicted saturation value. Also, the sensitivity$^{**}$ is a maximum in this range. These

* -- See reference 11, page 244.

$^{**}$ Defined as $dP_n/d(L + X)$. 

Note: - nozzle diameter $d_n = 0.040''$

FIG. 5.3 - PRESSURE ($P_n$) vs FLAPPER DISTANCE ($L + X$) FOR 0.040'' DIA. NOZZLE
observations are used in later design considerations.

Before continuing with a description of the design procedures used for the pneumatic sensors, some consideration must be given to defining the operating specifications which the devices must satisfy. Also, any environmental constraints must be outlined. It is found that these factors alone, to a large extent, determine the final design. The operating specifications include the signal load (or load impedance), the available flapper movement, the sensitivity, and the output pressure swing required. The environmental constraints are; saturation of the fluid amplifiers, maintaining as low a power consumption as possible, and maintaining the forces on the flapper due to the nozzle jet impact, as low as possible. These last two restrictions are both satisfied by a low nozzle upstream stagnation pressure $P_n$.

The load of either the position or rate sensor can be safely defined at the beginning of the analysis as the input port of a 1300PO1 proportional amplifier*. This contention is based on the reasonable assumption that the nozzle-flapper output signal $P_n$ must be amplified before it can be used for summation; an assumption in keeping with the earlier mentioned desirability of a low value for $P_n$.

The available flapper movement is fixed, in this case, by the rate gyro. The rate gyro used for this project was a modification** of a unit manufactured by "Graseby Instruments of Tolworth, Surrey, England". A photograph of this gyro is shown in figure 5.4. The instrument was

* -- As described in chapter 4.

** -- The modification consisted of decreasing the spring rate of the gimbal retaining springs, so as to increase the sensitivity.
FIG. 5.4 - RATE GYRO - SHOWING NOZZLE-FLAPPER SENSING APPARATUS.
originally designed to measure angular rate in the range ±10 radians per second. The rotor, which was governed at 9600 R.P.M., was driven by 48 volts D.C. This gyro was used for the demonstration model only, it being assumed that a pneumatically driven unit would be obtained for a final system. A plot of flapper displacement vs angular rate for the gyro is shown in figure 5.5. The flapper in this instance, consisted of an attachment to the inner gimbal frame. Thus the flapper actually moved in an arc. However, the flapper displacements encountered are very small compared to the radius of arc and for all intents and purposes, the flapper movement can be assumed linear. The limits of the proportional range of rate signal required by the system* are ±1.2 radians per second. From figure 5.5 it is noted that this corresponds to a flapper movement of ±0.005". Since a high sensitivity is desirable, then 0.005" should be 30% of the saturation value of (L + X) as shown earlier. This also provides a relatively linear operating characteristic. Simple algebra then leads to a value for the nozzle diameter -- i.e., \( d_n = 0.060" \). Thus, the nozzle diameter has been determined entirely on the basis of available flapper movement. A larger diameter nozzle would also operate in the linear region noted above but would consume more power than necessary.

The only parameter remaining in the nozzle-flapper assembly, is the value of the input restrictor \( R_s \). This value will be determined by the load requirements and consequently is considered in the system static design procedures of the next chapter. A detail drawing of the nozzles used is shown in figure 5.6.

* -- As determined in the computer simulation, chapter 2.
FIG. 5.5 - FLAPPER POSITION vs ANGULAR RATE FOR RATE GYRO
Note: nozzle length $D = 2''$ for rate sensor
    $= 1''$ for position sensor

FIG. 5.6 - NOZZLE DETAIL
For the angular position sensor on the attitude reference gyro, the nozzle-flapper principle is utilized but in a slightly different manner than the conventional configuration as described above for the rate gyro. A cam is used in place of the flapper. This results in the nozzle output pressure being a function of the relative cam angle; the function being determined by the shape of the cam. A detail of the proposed cam for this purpose is shown in figure 5.7. The particular shape shown, was chosen mainly for ease of manufacture. It was assumed at the time that non-linearities in the $\Theta$ vs $P_n$ relationship, due to the shape, could be compensated for in the final system. It is quite feasible to develop a cam shape that would produce a linear output characteristic, but such a cam would be necessarily complex and add little to the system as a whole. As will be shown later, the chosen shape for the cam actually assists in the design by improving the angular sensitivity of the sensor in the range of interest.

The input characteristics of a 0.060" diameter nozzle, when paired with a cam of the type shown in figure 5.7, are shown in figure 5.8. For these characteristics, the gap between the high side of the cam and the nozzle, was set as close to zero as practically possible, without actual contact. The angular range covered corresponds to that segment of the cam for which the radius is a function of the angle. That is, for $0 < \Theta < 90^\circ$, as defined in figure 5.7. Using the characteristics of figure 5.8, plus those of the load, the characteristics of the required input restrictor can be determined.

It is informative to compare the nozzle-cam characteristics of figure 5.8, with the nozzle-flapper characteristics of figure 5.9. The latter are for a 0.060" diameter nozzle, identical to the one used for figure 5.8.
Note - Cam 3/16" Thick.

- For $0 \leq \theta < \pi/2$,
  \[ y = 1 - 0.015 \cos \theta - \sqrt{1 - (0.015)^2 \sin^2 \theta} \text{ in.} \]
  or, \(|y| \approx 1 - 0.015 \cos \theta \text{ in.}\)

- For $\pi/2 < \theta < \pi$,
  \[ y = 0.015 \text{ in.} \]

- For $\pi < \theta < 2\pi$,
  \[ y = 0. \]

FIG. 5.7 - CAM DETAIL
FIG. 5.8 - NOZZLE - CAM

INPUT CHARACTERISTICS
FIG. 5.9 - NOZZLE - FLAPPER OUTPUT CHARACTERISTICS
As can be seen the correspondence between the curves is nearly perfect; indicating that the curvature of the cam has little effect on performance.

The attitude reference gyro used for the demonstration model is shown in figure 7.2. This unit is a two degree of freedom vertical reference gyro. Modifications included, removing the selsyns used for gimbal position information and adding the nozzle-cam apparatus designed for pneumatic sensing of the outer gimbal position with respect to the frame (photograph of the nozzle-cam apparatus is shown in figure 7.3). The rotor is driven at 20,000 R.P.M. by a 115 volt, 400 cycle source. As with the rate gyro, this unit was used strictly because of its availability. In a final system, a pneumatically driven gyro would be used.
6. FLUIDIC CONTROL SYSTEM PROPOSAL

The control system in question is relatively basic. As a consequence, the series of functions that must be carried out in order to achieve the desired results are straightforward and can be immediately written down as follows:

- Pneumatically sense position error and provide a usable signal proportional to same.
- Pneumatically sense rate error and provide a usable signal proportional to same.
- Sum the position and rate errors in such a manner as to provide the required output function of same (as determined in chapter 2).
- Amplify the resultant output signal to facilitate coupling to the thruster-power amplifier stages.

It is assumed in the above that dynamic considerations with respect to the fluidics, are relatively unimportant at the frequencies of concern (10 cps and below). Continuing with this assumption, a fluidic control system is proposed (not including actuator and power stages which are the subject of another thesis) which is the simplest possible system required to carry out the above functions. The proposed system is shown in schematic form in figure 6.1. The simplicity of the system and the low frequency requirements of the system, will, it is hoped, combine to make dynamic considerations negligible. Actually, it was not known at the start of this project if the fluidic dynamics could be ignored. It is simply more practical to construct the system as proposed, test the resultant dynamic properties, and apply any indicated compensation.

The first step in the design is the choice of the supply pressure $P_s$. 
FIG. 6.1 - PROPOSED FLUIDIC CONTROL SYSTEM
This is dictated primarily by the pressure requirement of the most powerful stage in the system. Since the highest practical supply pressure for the 1300POI proportional amplifier appears to be approximately 50 psi, this value was chosen for $P_s$.

The supply jet pressure for the position sense amplifier is next. This pressure must be as low as possible, so that the required differential control pressures are small. By keeping the latter small, good angular sensitivity is assured, since the cam-nozzle input impedances $R_1$ and $R_2$ are high. The supply jet pressure must not be too low however, since this would make the system more sensitive to noise and demand more active stages after the summing amplifier to achieve the required output signal level.

A reasonable compromise with due reference to the above defined value of $P_s$ and the characteristics of both the proportional amplifiers and the cam-nozzle, would be 10 psi. Thus, $R_5$ must be of such a value as to provide 10 psi at the supply jet port of $A_1$. From the supply characteristics of the proportional amplifier, figure 4.3, it is noted that the input flow at 10 psi is $31 \text{ ft}^3/\text{sec} \times 10^{-4}$. If it is assumed that linear fluid restrictors are available, then a straightforward calculation gives;

* -- All component designations in this chapter refer to figure 6.1 unless otherwise stated.

** -- Actually, such a level has not been defined as yet since the input requirements of the thruster-servo system were not known at the time of this design; consequently, it is assumed to be the output of an amplifier with 50 psi at the supply jet.
However, at the time this work was done suitable linear restrictors were not available. As a consequence, needle valves are used throughout as fluid resistors. The pressure-flow characteristics of a needle valve are highly non-linear as shown in figure 6.2. Thus, rather than calculate fluid resistances, it is much simpler to install the valve in the circuit and set the required pressure drop "in situ". This situation is true for all the fluid restrictors in the system. Therefore, the design procedure consists of obtaining the quiescent static pressure drops required of the restrictors; the incremental fluid impedance at any given operating point being obtainable from the needle valve characteristics in figure 6.2.

For a supply nozzle input of 10 psi, the 1300P01 proportional amplifier requires an input differential pressure swing of ±0.7 psi to drive it fully through its useful range. The useful range is defined as that portion of the output characteristic which displays relatively linear operation. This information is readily obtained from the output characteristics given in figure 4.5. Also, from the input characteristics in figure 4.4, it appears that the incremental input impedance is of the order of 1600 fluid ohms (fluid ohms has units of psi/ft³/sec as defined here). It should be stressed here that these fluid impedances cannot be used in an ohms law manner for static calculations. This is due to the fact that although the curve in question may display a substantial linear range, a linear extrapolation rarely intercepts the origin. The procedure used to overcome this problem
 wided

Note:
- experimental points ** experimental points
- valve used was "Alkon"
Series "J", No. JN1

FIG. 6.2 - NEEDLE VALVE IMPEDANCE CHARACTERISTICS
throughout the system design is to simply superimpose the curves in question to determine operating points.

Continuing with the design of the position sense circuitry, the next problem consists of determining the pressure drops across R1 and R2. The nozzle-cam configuration used for position sensing is assumed to be fixed, it being previously designed from practical limitations as well as functional performance considerations. The finalized design of the nozzle-cam system with physical dimensions and reasons for same, were given earlier*. The resultant input characteristics are shown in figure 5.8.

At this point an assumption is introduced which will be repeated in all future amplifier calculations. The assumption is that of ideal isolation between the input ports and the output port loading conditions in the 1300pol fluid amplifiers. Thus, the nozzle-cam position sense circuit can be schematically represented as shown in figure 6.3.

The cam detail drawing, figure 5.7, should be recalled at this point to fully appreciate the mode of operation of this circuit. As indicated in figure 5.7, the nozzle-cam distance "y" varies from zero** at θ = 0, to 0.015" at θ = 90°. The gap then remains at 0.015" until at θ = 180° a step change takes place to y = 0. From θ = 180° to θ = 360° the gap remains at zero.

In operation two cams are used; one for each nozzle. Both are mounted on the same shaft but one is the mirror image of the other. The resultant

* -- See chapter 5.

** - Actually, of course, y = ε at θ = 0, ε being as small as practical without contact.
FIG. 6.3 - SCHEMATIC REPRESENTATION OF NOZZLE-CAM POSITION SENSE CIRCUIT
gap vs angle relationship for both nozzles is therefore as shown in figure 6.1. The relationship is shown as linear in figure 6.1 for clarity, although in fact, it is a trigonometric function as indicated in figure 5.7. Also, the angle is shown as varying from 0 to $180^\circ$, in accordance with the definition of angular error. Thus it is seen that each nozzle has a distinct range of effectiveness. One nozzle represents positive error; increasing to $90^\circ$, then remaining saturated until $180^\circ$. The same nozzle gives no signal from $\theta = 0$ to $\theta = -180^\circ$. The other nozzle provides the exact opposite output with respect to the angular error. The discontinuity at $\theta = 180^\circ$ introduces no problems during despinning of the payload since the initial high rates involved, will be more than enough to overcome the fluctuating position signal and thus maintain maximum deacceleration torque. When the rate is low enough for the position signal to have effect (see figure 2.2), then the payload will come to rest without making another complete revolution.

The use of two cams, as described in the above manner, provides for a definite polarity to the error signal while allowing for active operation over the full $360^\circ$. A single cam design is feasible and in fact has been considered; but such a design would limit the useful range to $\pm 90^\circ$.

Returning to the nozzle-cam position sense circuit, figure 6.3; the only parameters yet to be determined, are the quiescent bias levels at the A1 input ports. This bias level is determined by the values of R1 and R2; all other fluid impedances being fixed. The input impedance characteristics of the cam-nozzle configuration are shown in figure 5.8. Recalling that 0.7 psi differential pressure is required for maximum
Nozzle 1

Nozzle 2

FIG. 6.4 - ANGULAR ERROR vs NOZZLE-CAM GAP DISTANCE "y"
output of A1 and that only one nozzle is active for any given error; then either side of the nozzle-cam sensor must change* the pressure by 0.7 psi, at the respective input port, for a change of position of 90°. To obtain the required value of R1 or R2 to accomplish this, the cam-nozzle input characteristics (figure 5.8) are added to the A1 input characteristics (figure 4.4) along the flow axis as shown in figure 6.5. The resultant curves give the combined characteristics of the cam-nozzle and A1 input port in parallel. A load line with a pressure axis intercept at Ps = 50 psi and slope of -R1, would represent the supply resistor R1. It is seen from figure 6.5 that such a load line, giving the required 0.7 psi change in pressure, is almost horizontal. The impedance represented by the slope is:

R1 = (50/8.5) x 10^4 = 58,900 fluid ohms.

The quiescent pressure, or bias level, is 0.8 psi. The A1 input impedance characteristic has been taken as representing the combined impedance for Θ = 0°. It is noted that on the load line, the pressure sensitivity, dP/dΘ, is a maximum at Θ = 0, and drops off rapidly after the Θ = 40° characteristic. This type of performance is precisely what is required here, and is readily predicted from both the pressure vs gap curves of the nozzle-flapper combination, and the y vs Θ curves for the cam. Both these curves show maximum rate of change for small gap distances.

From the analog simulation of the rocket roll characteristics, it

* -- Decrease in this case.
FIG. 6.5 - CHARACTERISTICS OF $A_1$ INPUT AND CAM-NOZZLE IN PARALLEL
was noted that the torquing thrusters required a saturation signal for a position error of 0.4 radians or 23°. Such performance seems reasonable from the above design.

Before the design can proceed further the required supply pressure for the rate sense amplifier, A2, must be determined. It will be recalled, that for the position sense amplifier A1, the supply pressure was chosen to be 10 psi. If direct flow summation is used*, as implied by the system proposed in figure 6.1, then for A1 and A2 operated with the same supply pressure and over the same output range, the summed signal for maximum outputs of opposite phase will be zero. This is not in accordance with the mode of operation assumed for the analog computer simulation of the system**. One of the assumptions in this study was that the limiting of the signal took place after the summation of the rate and position error signals. However, the position error cannot ever be greater than ±180°. Consequently, it should be possible for the rate error to completely over ride the position error for rates greater than the maximum of 

\[ \hat{\theta}_m = \pm 0.6 \text{ rad./sec.} \]

To achieve this, the supply pressure to the rate sense amplifier must be greater than the 10 psi used for the position sense amplifier. Also, the sensible range of rate for the rate gyro must be ±1.2 rad./sec. The output from A2, for a rate of ±0.6 rad./sec., must be equal to the maximum output of A1. The output from A2, for a rate of ±1.2 rad./sec., must be capable of completely inverting a maximum signal of opposite phase from A1. These specifications are by

* - For summing the outputs of A1 and A2.

** - See chapter 2.
no means infallible, since ideal flow summation is assumed, but they should be enough for "ball park" analysis; some trimming likely being required in the final system.

The transfer characteristic of the 1300P01 proportional amplifier as provided by the manufacturer (figure 4.2) implies a constant slope for varying supply pressures. Capitalizing on this fact, the supply pressure for A2 necessary to meet the specifications of the last paragraph, is seen to be 20 psi*. In other words, it is assumed that double the supply pressure, will give double the attainable differential output. Likewise, double the differential input pressure is required for control.

The circuit design of the rate sensor is complicated by the fact that the gap for both nozzles varies simultaneously (but in opposite directions) with flapper displacement. This is in contrast to the position cam-nozzle system, where only one gap was active at any given angular displacement. A plot of rate vs flapper displacement, for the rate gyro, is given in figure 5.5. Using this plot in combination with the general nozzle-flapper (0.060" dia.) input characteristics, figure 5.9, the input characteristics for the rate gyro nozzles can be obtained with rate as a parameter. However, before this can be done, the quiescent or centered-flapper gap setting must be determined.

As was shown earlier, the maximum differential output swings from the

* -- Actually, as will be seen from the experimental results, the assumption of ideal summation is a poor one; ie, it will be found necessary to raise the supply nozzle pressure on A2 to 35 psi in order to meet the summation requirements.
rate sense amplifier should be attained for a rate maximum of $\dot{\theta}_m = \pm 1.2$ rad./sec. From the flapper displacement vs rate curve, figure 5.5, it is seen that $\pm 1.2$ rad./sec. corresponds to approximately $\pm 5.0$ mil's of flapper movement. This suggests a quiescent gap setting of $5.0$ mil's. With this setting, any rate above $\pm 1.2$ rad./sec. will produce a gap of zero on one side and 10 mils on the other. Such a situation must, by previous requirements, provide a differential input pressure to A2 sufficient to drive it fully to one side. It has been assumed here that the hysteresis in the rate gyro is negligible*.

Using the above determined quiescent gap setting and combining the information contained in the rate-displacement curve and the nozzle-flapper input characteristics, the rate sensor input characteristics are obtained as shown in figure 6.6. These characteristics are then re-drawn, after adding them (flow summation) to the A2 input impedance characteristic, to give the combined parallel impedance characteristic family of the A2 input and rate sensor nozzle-flapper (figure 6.7). As with the position sense circuit, a load line with a slope representing R3 or Rh, and a

* -- For a working system to be used in a rocket, the hysteresis shown in the rate gyro characteristic, figure 5.5, would not be negligible; however, for the purpose of demonstration, certain inadequacies in the components can be tolerated, since they detract from performance rather than improve it; thus, providing greater confidence in the design when the model performs satisfactorily.
Note: For quiescent gap setting of 0.005"
- Dashed curves are extrapolated

FIG. 6.6 - RATE SENSOR INPUT CHARACTERISTICS
Note: For quiescent gap setting of 0.005"
pressure axis intercept of 50 psi, is now chosen. The choice is based on obtaining the required pressure swing for the range of rate of interest. The characteristics of figure 6.7 represent one nozzle only. Nevertheless, due to the symmetry of the device, the characteristics of both nozzles appear on the diagram for any given rate. The instantaneous operating point of one nozzle being found at the load line intercept, with the rate characteristic of identical magnitude but opposite sign, to that rate characteristic representing the other. Thus, the pressure differential seen by the amplifier input for any load line, is given by the difference of the pressures representing the load line intercepts with the appropriate two characteristics of the same rate magnitude.

With the above factors in mind the load line intercept on the flow axis is chosen to be 28 ft³/sec x 10⁻¹. The resultant load line, shown on figure 6.7 indicates a differential pressure swing of slightly over 1.6 psi. This value is greater than double the 0.7 psi value used for Al. However this exaggeration was done purposely to account for the fact that the \( \theta = 1.20 \text{ rad/sec} \) characteristic shown cannot be reached in practice; i.e., it is doubtful that the flapper will cut off the nozzle flow completely. If experiment shows this reasoning to be false, only a slight trim adjustment is required to correct the situation.

The final operating point, as shown in figure 6.7, gives a quiescent pressure setting (for \( R_3, R_4 \)) of 1.1 psi. Thus, the design of the two sense amplifiers is essentially complete.

The design of the summing and final amplifier stages follows in an identical manner to the above. The problem here is essentially one of matching. It will be noted from the amplifier characteristics, chapter 4, that an amplifier operating with a supply of 10 psi, over its useful range,
will adequately drive an amplifier with a supply of 50 psi, over its useful range. These supply pressures were therefore chosen for the summing and final amplifiers respectively.

The matching between the last two amplifiers is thus complete; it remains to match the summed output of the sense amplifiers, to the input of the summing amplifier. This is accomplished through the use of the variable restrictors R7 and R9, which are adjusted to provide the same quiescent bias level at the input to the summing amplifier (A3) as was used for A1 (0.8 psi). The reasoning behind this decision, is that assuming the restrictors, R7 and R9, are linear over the limited range of operation, then the incremental changes that appear at the inputs to the sensing amplifiers, will appear also at the input to the summing amplifiers. Such an assumption was found to be reasonable from the performance tests on the demonstration model.

Thus the static design of the roll control system is complete.
7. EXPERIMENTAL RESULTS

The roll position control system described in the previous chapter of this thesis, was constructed in a bread-board manner on a test stand, for the purpose of demonstration. A photograph of the resulting assembly is shown in Figure 7.1. As discussed in chapter 5 of this thesis, the gyros shown in the aforementioned photograph were modified to meet system requirements. The compound angular motion stand used for test purposes, contained eleven channels of electrical slip rings. These channels were used to provide the necessary power for gyro operation, and for the instrumentation used for performance tests. Photographs showing a close-up of the bread-boarded control system and the cam-nozzle assembly are shown respectively in figures 7.2 and 7.3. There is nothing in the fabrication of the fluidic sensors that represents a relatively high degree of machining capability. As a consequence, the construction of the demonstration model was straightforward.

The variable restrictors used on the model are of the needle valve type. The needle on these particular units employed a three degree taper for fine control of flow. Also, a micrometer type adjustment was provided. The needle valve impedance characteristics were given in figure 6.2.

The compound test stand on which the model was mounted, incorporated three rotationally independent, co-linear shaft segments. Each shaft is free to rotate independent of the others. The lower two segments have electrical drive mechanisms. The bottom segment is driven with respect to the stand. The center segment is driven with respect to the bottom segment. The top segment, which contains the model, is free except for
FIG. 7.1 - FLUIDIC CONTROL SYSTEM DEMONSTRATION MODEL ON TEST STAND.
a remote operated lock which can be used to connect it to the second or center segment. The axis of the upper two segments can be tilted with respect to the lower shaft to simulate the motion of rocket coning.

Tests of the demonstration model operation on the compound test stand showed satisfactory performance when some limitations are taken into account. First and foremost, the actuator system consisting of high gain fluid amplifiers and torquing thrusters could not attain the required 28 lbf. thrust. The highest available thrust with the finished system was 0.09 lbf. Also, the friction of support bearings, electrical slip rings and a pneumatic slip ring added up to a constant dynamic resistance torque of 1.21 lb.in. To partially compensate for the low available torquing thrust, the moment arm was increased to 23 inches. The result of these changes was a value for maximum control torque of 0.86 lb.in., rather than the proposed figure of 72 lb.in. This is a severe reduction in controlling torque and in fact, is equivalent to using a value of \( J_m = 0.096 \text{ lb.in.} \) in the simulation equations of chapter 2. However, even with these limitations, the model displayed more than adequate damping and a deadband of less than the required 22 degrees.

The step response of the model is shown in figure 7.1. This response compares favorably with those given in chapter 2 for the simulated model. It is apparent from these results that the factor which has suffered most from the reduced actuator power, is the settling time. This time has been increased by a factor of four.

The response of the demonstration model in figure 7.1, is similar to

* -- Assumed constant over angular velocities of interest.
FIG. 7.4 - EXPERIMENTAL RESPONSE OF DEMONSTRATION MODEL

Note:

- $F_m = 0.09$ lbf.
- $I_p = 7.05$ lbf. in. sec$^2$
- $L = 23$ in.
- $\theta_m = 0.4$ radians
- $\dot{\theta}_m = 0.6$ radians/sec
that of the simulated response shown in figure 2.4 of chapter 2. This is an indication that the assumptions made in chapter 2 are valid ones and that the principles and methods utilized in the design procedures for the fluidic control system are basically sound.

To gain some insight into the dynamic characteristics of the fluidic system a series of frequency response tests were carried out on the bread-boarded model. The results of these tests are in the conventional form of gain in decibels vs log frequency and phase lag vs log frequency. The resultant curves are shown in figures 7.5 through 7.7, representing respectively, the position sense circuit, the rate sense circuit and the summer and output amplifier circuit. It should be noted, that the gain as defined in these results is the pressure gain and therefore is a function of load at the point of measurement. The experimental methods used in determining these curves and some additional curves are discussed in Appendix A.

An examination of the frequency response curves indicates that, as expected, the order of the transfer function for a particular fluidic sub-system is directly related to the number of active elements involved. Thus, the rate and position sense circuits are both first order systems while the summation amplifier and output amplifier in cascade represent a second order system. Perhaps the most important factor to be learned from these results, however, is the relatively low attenuation for the previously defined* frequency range of interest of 0 to 10 cps. This fact reinforces the assumption made in chapter 6; that dynamics can be

* -- See chapter 6.
FIG. 7.5 - FREQUENCY RESPONSE
of POSITION SENSE CIRCUIT
Fig. 7.6 - Frequency response of rate sense circuit

NOTE: Experimental points are gain = S, phase = X.
Curves represent
\[ G(s) = \frac{0.8 e^{-10s}}{1 + 10s} \]
for time = 3.18 x 10^{-3} seconds,
lag = 2.66 x 10^{-3} seconds.
FIG. 7.7 - FREQUENCY RESPONSE of SUMMER and FINAL AMPLIFIER
neglected in the design of the system for the above mentioned frequency range. However, it should be noted that the phase lag is not necessarily negligible at 10 cps. This is due to the transport lags involved. The transport lags arise almost entirely as a result of the long connecting lines used between the various elements*. A more physically sophisticated model would use considerably shorter interconnections and virtually eliminate this problem.

A block diagram of the control system is shown in figure 7.8 and effectively summarizes the results of the frequency response tests.

To sum up, it is sufficient for the purposes of this thesis to state that the experimental tests of performance and response of the demonstration model reinforce the original assumptions and provide considerable confidence in the design techniques used.

* — See figure A.1, for actual lengths involved.
FIG. 7.8 - BLOCK DIAGRAM OF ROLL CONTROL SYSTEM

Note:
- Abscissa represents differential input pressure (psi)
- Value of $K$ is dependent on power amplifier gain
8. SUMMARY AND CONCLUSIONS

The control system described in this thesis is developed primarily as a means of studying fluidic system design techniques. A secondary function is the determination of the feasibility of the application. The application consists of a single axis position control system used to stabilize and position (about the roll axis) the partial payload of an upper atmosphere sounding rocket.

The system, a schematic of which is shown in figure 6.1, is implemented using all fluidic elements, from sensing roll position and its first time derivative, to generation of the required error signals. The active fluidic component was chosen to be a beam-deflection proportional amplifier.

Nozzle-flapper techniques are used to sense the position and rate error, as provided by a two degree of freedom gyro and a rate gyro respectively. The resulting pressure signals are amplified and summed by the fluid amplifiers and the output signal used to drive the power amplifiers and thrusters which are the subject of another thesis. The gain of the various components is adjusted so as to provide an output signal which has the correct relation to the position and rate errors as determined by an analog computer simulation of the system.

The graphical design techniques developed in chapter 6 to solve for the complex matching conditions between components are, to the author's knowledge, original as applied here. Graphical design is not new and in fact has been used for some time in electronics for matching electronic devices with nonlinear characteristics. Experimental results of tests carried out on a demonstration model, constructed for this purpose, verified that these techniques are an accurate and powerful tool for systems design in the field of fluidics.
The nozzle-flapper principle is successfully applied to the problem of converting the information available from the two gyros (in the form of relative positions of mechanical components) to a useful pressure signal proportional to same.

To summarize, the conclusions which are derived from the work presented in this thesis can be listed as follows:

- The graphical design techniques employed are an accurate and useful tool for systems design in the field of fluidics.

- The nozzle-flapper principle is an excellent device for providing a mechanical interface between relative motion and fluidic circuitry.

- From test results obtained on the model, to date, it appears that fluidics could be successfully applied to the roll position control of a partial payload of an upper atmosphere probe.
9. DISCUSSION AND RECOMMENDATIONS

There are many ways of achieving the roll control function outlined in this thesis. Even limiting the components to pure fluidics does little to alter the almost infinite choice of possibilities. The system described in this thesis is but one of this field. If the goal of this research had been the development of a practical working system, then the work contained herein would constitute a necessary and major step towards this goal. However, the aim here was basic research into the techniques of fluidic systems design. As a result the finished system described here, although adequate for this study, requires further development to make it truly practical. The purpose of this section is to discuss the areas where further development is required to obtain this end and also, to recommend areas of further research in this subject beyond the scope of the present endeavor.

The large physical size of the system as laid out on the demonstration model is perhaps the most obvious area where improvement is required. The entire system, excluding the gyros and associated sensors, could easily be integrated into a much smaller form. The result would contain two sandwiched two-dimensional layers of circuitry with very short interconnecting lines between elements. The variable restrictors could consist of set screws which would partially block the appropriate channels; the amount of restriction being a function of their position. The reason for including adjustment in a final system is that there will always be some variation of symmetry and amplification properties in the fluidic elements, due to manufacturing tolerances. The resulting integrated fluidic circuit would be placed as close as possible to the respective
signal sources to keep line lengths to a minimum. The gyros would be pneumatically driven to eliminate the necessity of electrical power supplies. The resulting compact system would more than satisfy the requirements of the partial payload roll position control application.

Some further refinements which would facilitate the use of this system, are worth mentioning. While the missile is on the launch rail, it is necessary for the ground crew to have some control over the position gyro. This could be accomplished through the umbilical cord and would consist of remote caging and uncaging of the gyro gimbals and remote control of the angular position reference on the gyro. The latter function could be realized by rotating the entire missile to the final required orientation, or by remote control of the caged position of the gyro rotor.

It is fitting to terminate this thesis with a discussion of the areas where further research is desirable, as indicated by the present work.

- The nozzle-flapper principle.

This is a most outstanding area for further work; the reason being the comparative lack of available literature (to the author's knowledge) concerning the basic theoretical flow mechanisms upon which the operation of the device is based. For instance, it would be very convenient, in designing components utilizing this principle, to have an analytical expression describing the upstream nozzle pressure in terms of the nozzle-flapper distance, with nozzle diameter, supply restrictor and load restrictor as parameters.

- Fluid circuitry (pneumatic) dynamics.

There is a need for further research into the dynamics of fluid
circuitry, both active and passive.

- Pneumatic slip ring.

In a more practical vein, there is need for development of very low torque pneumatic slip rings (i.e., for two degree of freedom gyros).

Each of these areas of study involves a considerable amount of work. It is the hope of the author, that the particular areas of research and the resulting conclusions, contained in this thesis will be found useful in furthering the present state of knowledge of fluidic control systems design and provide incentive to continued research in this and related topics.
LIST OF REFERENCES


APPENDIX A

EMPIRICAL METHODS

A.1 Measurement Methods Used for Static Characteristics

The methods used for measuring the static characteristics of the various devices described in this thesis are relatively straightforward. In all cases the variation of parameters was accomplished through the use of needle valves as variable restrictors. Pressures were measured with mercury or water filled manometers; the choice being dependent on the pressure range involved. The manometer readings were converted to "pounds per square inch" (psi) as these units are considered to be the most universal and convenient for the present application. Flow was measured using tri-flat variable area flowmeters*. The readings from these flowmeters were taken directly in "cubic centimeters per minute", however correction was required for metering pressure errors as outlined in reference 12. These results were then converted to "cubic feet per minute", also for convenience.

To assist in understanding the various experimental static characteristics presented in the text, a schematic representation of the experimental arrangement utilized in their derivation is shown alongside the respective curves.

A.2 Measurement Methods Used for Dynamic Characteristics

The dynamic characteristics presented in this thesis in the form of

* -- as manufactured by Fischer & Porter (Canada) Ltd.
frequency response test results, were obtained through the use of a torque-motor driven pneumatic sine wave generator. This generator is simply a nozzle-flapper device with an electromagnetically driven flapper. The nozzle diameter was chosen as 0.060 inches so that the device could be used as a direct substitute for the sensing nozzles of the position and rate sensors. In all of the frequency response tests the input signal amplitude as produced by the pneumatic sine wave generator was maintained constant at 0.1 psi, peak to peak, throughout the frequency range. This was done in order to eliminate any variations in gain due to nonlinearities in the system.

Measurements of input and output pressure waveforms were made with piezoelectric pressure transducers* and associated charge amplifiers. The outputs of the charge amplifiers were displayed simultaneously on a double beam storage oscilloscope. The empirical data was obtained directly from this display.

For the case of the frequency response of the demonstration model, it was considered imperative that all tests be made with the system intact; thus maintaining unchanged all loading conditions. This is necessary since pressure gain is used in the frequency response, and is very much a function of load. The input signal was the pneumatic sine wave generator used to supplant one nozzle of either the rate or position sensor. The output signal was taken at the input to the summing amplifier or the output ports of the final amplifier, depending on the circuitry under test. Four frequency response tests were made. One for each sensing circuit alone (i.e., the output was the input to the summing amplifier) and

* -- Kistler model 606L transducer & model 5044 charge amplifier.
one for each complete sensing loop including the common summing and final amplifiers (ie., the output was the output of the final amplifier). The results of the former are given in chapter 7, figures 7.5 and 7.6. The results of the latter are shown in figures A.1 and A.2. The response of the summer and final amplifiers was then determined by subtracting the sense circuit response from the respective loop response. This provides the response of the summer and final amplifier as seen by each sensing circuit. The fact that there is a difference is important and justifies the indirect procedure used to obtain this response.

The response of the summer and final amplifiers as seen by the position sense circuit is shown in figure 7.7 of chapter 7, while the same as viewed by the rate sense circuit is shown in figure A.3. It is noted from these curves that they differ only by a constant value of gain, both having the same corner frequency as expected. Also the transport lag of the summer and final amplifiers appears different to the two sense circuits, as determined from the phase lag characteristics of the respective plots from which the above mentioned curves were derived. These differences are summerized by the blocks marked "position sum" and "rate sum" shown on the block diagram, figure 7.8 of chapter 7. The reason for the differences noted above is the disparity in the interconnecting line arrangements (line lengths, placement of "Tees" etc.) between the sense circuits and following stages. It is expected that these variations would be materially reduced, if not eliminated, in an integrated type circuit arrangement. A schematic diagram of the demonstration model interconnecting line arrangement, including lengths, is shown in figure A.4.

An interesting corollary to the above response tests is displayed in
FIG. A.1 - FREQUENCY RESPONSE
of POSITION LOOP CIRCUIT
FIG. A-2 - FREQUENCY RESPONSE of RATE LOOP CIRCUIT
FIG. A.3 - FREQUENCY RESPONSE
of SUMMER and FINAL AMPLIFIER
FIG. A.4 - FLUIDIC ROLL POSITION CONTROL SYSTEM INTERCONNECTION DATA
the curves of figure A.5. These curves were obtained by plotting the
difference in gain between the actual experimental points and the
derived first order approximation for the position and rate sense circuits
respectively. Some component of these curves is undoubtedly due to
experimental error; a variable factor which is difficult to estimate.
However, the shape of these curves is unmistakably due to transmission
line effects in the interconnecting lines of the respective circuits.
A detailed treatment of this phenomena is beyond the scope of the present
work. The interested reader is referred to references 13, 14, 15 and
16, in the bibliography.
FIG A.5 - FREQUENCY RESPONSE ERRORS ASSOCIATED WITH SENSE CIRCUIT RESULTS
APPENDIX B

ROCKET VEHICLE CHARACTERISTICS

The control system developed in this thesis was justified by the application outlined in chapter 1 (the automatic roll position control of a partial payload of an upper atmosphere sounding rocket). Since the major component affecting this application is the rocket vehicle, a detailed description of its characteristics and typical performance is given here.

The rocket vehicle, whose past performance was used to assist in decisions affecting the control systems mode of operation, was the Black Brant IIA. Each of these vehicles used, carries a different payload configuration and consequently, each is an individual with respect to physical parameters such as weight, location of C.G., moments of inertia, etc. For this reason, a particular set of flight test results which are considered as typical for the purpose of this thesis, are presented here for reference.

Table B.1 provides a detail listing of the vehicle physical data. Figure B.1 shows a drawing of the vehicle giving major dimensions and component section definitions. Figures B.2 through B.5 show the vehicle performance data deemed pertinent to the present application. For more information, refer to reference 17.
TABLE B.1

VEHICLE DATA

Weights and Centers of Gravity

<table>
<thead>
<tr>
<th>Items</th>
<th>Weight (lb.)</th>
<th>C of G Sta. (in.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nose cone and payload</td>
<td>248.5</td>
<td>74.19</td>
</tr>
<tr>
<td>Motor (unburnt)</td>
<td>2261</td>
<td>204.32</td>
</tr>
<tr>
<td>Motor (burnt estimated)</td>
<td>481.2</td>
<td>214.38</td>
</tr>
<tr>
<td>Fin assembly</td>
<td>156.0</td>
<td>307.05</td>
</tr>
<tr>
<td>Complete vehicle (unburnt)</td>
<td>2668.1</td>
<td>198.17</td>
</tr>
<tr>
<td>Complete vehicle (burnt)</td>
<td>888.5</td>
<td>191.27</td>
</tr>
</tbody>
</table>

Vehicle Parameters

- Overall length, inches: 321.26
- Body diameter, inches: 17.21
- Body area, ft²: 1.614
- Length of conical section, inches: 86.0
- Length of cylindrical section, inches: 24.0
- Cone semi apex angle, degrees: 5.73
- Gross payload volume, ft³: 6.00
- Parallel length of casing: 184.0
- Fin gross span, inches: 66.2
- Fin net span, inches: 49.0
- Fin root chord, inches: 36.10
- Fin tip chord, inches: 10.80
- Fin leading edge sweep back angle, degrees: 50.0
- Fin trailing edge sweep back angle, degrees: 10.0
- Fin planform area (each), ft²: 4.0
- Nozzle length, inches: 21.0
- Nozzle throat diameter, inches: 5.25
- Nozzle exit diameter, inches: 13.46
- Nozzle exit area, ft²: 0.988
- Pitch moment of inertia (unburnt) slugs ft²: 2770.0
- Pitch moment of inertia (burnt) slugs ft²: 1600.0
- Roll moment of inertia, (unburnt) slugs ft²: 36.0
- Roll moment of inertia, (burnt) slugs ft²: 20.0
FIG. B. 1 - ROCKET PHYSICAL CONFIGURATION
FIG. B.3 - ROLL MOMENT of INERTIA vs TIME
FIG. B. 4 - CALCULATED ACCELERATION vs TIME

$W = 2675 \text{ lb.}$

$\phi = 80^\circ$
FIG. B.5 - TYPICAL FLIGHT TRAJECTORY