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**FACILITY AIR CONTROL SYSTEMS  
DESIGN FOR A PILOT TRANSONIC  
WIND TUNNEL**

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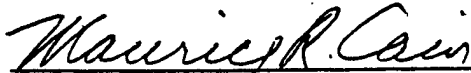
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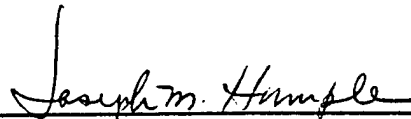
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# ABSTRACT

The pilot wind tunnel is a variable geometry wall transonic tunnel. It is constructed to use air from an existing bottle farm and blow down to atmosphere. The report presents the criteria for selection of control components and discusses in detail the design of the airflow control systems.

## FOREWARD

The work described herein is part of an on-going project to verify a concept for a wind tunnel flexible wall test section in a pilot scale facility. This effort, including the actual pilot tunnel which is approaching operational status, is being carried out at the Air Force Flight Dynamics Laboratory, Wright-Patterson Air Force Base, Ohio 45433. The work is an element to in-house Work Unit Number 24041303 titled "Develop New Technology Concepts for Aerodynamic Ground Testing" under Task Number 240413 titled "Aerodynamic Ground Test Technology".

Aerodynamic guidance and criteria for this effort are being supplied by Dr. Donald J. Harney, Assistant for Simulation to the Aeromechanics Division Chief.

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## I. INTRODUCTION

In late 1973 and early 1974 plans were being formulated in the Air Force Flight Dynamics Laboratory (AFFDL) for constructing a wind tunnel with a fifty inch square test section with variable geometry walls. The variable geometry of the walls would provide an adaptive test section which would allow testing larger models in a smaller tunnel than possible in a comparable sized conventional tunnel. The facility would provide support for in-house test programs when time in other, larger tunnels was not available, and provide this support at a lower cost. These economies would also permit a more thorough job of testing, which is important in a flight regime where discontinuities exists in modeling theories.

In order to assure that the design of the variable geometry is accurate, planning was initiated for in-house construction of a blowdown pilot facility with a nine inch square test section. The pilot facility would provide a means of testing the variable geometry concept. It would also provide a means of determining the degree to which the hardware realization would have to match the theoretical predictions of wall curvature requirements. Operating experience with the pilot facility should permit relaxing the curvature requirements and result in a test section of less complexity.

This report does not cover the aerodynamic design of the test section wall curvature. (Harney, 1973). The report is intended to document the design of the test leg air controls.



Design of test section is documented in an AFFDL Technical Report (TR)  
in preparation.

## II. DESCRIPTION OF FACILITY

A brief description of the tunnel airflow circuit and the test section is included herein. Test section details are described in this report. Airflow circuit details are described in a Technical Report in preparation. A schematic diagram is shown in Figure 1.

### A. TUNNEL CIRCUIT

1. Air Storage. The tunnel will operate from the same high pressure bottle farm as the AFFDL Mach 3 High Reynolds Number Facility (Fiore, 1975). This 15,000 Ft<sup>3</sup> of 3,000 psi air storage is shared by a number of wind tunnels and test rigs in the Bldg 450 complex. Due to line losses and other air usage, the useable air for this facility is approximately 15,000 pounds. The recharge rate varies from 15 to 2 lb<sub>m</sub>/sec depending on how many of four available reciprocating compressors are on the line. The air consumption/recovery ratio is approximately 100 to 1. A two minute run would require more than three hours of pumping time during which some facilities could not run. Therefore the wind tunnel design must maximize productivity for very short runs. The air circuit controls are designed to accommodate runs of twenty seconds or less.

2. Supply Piping. Supply piping to the test leg is shown in a schematic of the facility Figure 1. The perforated plate will act as a flow distributive throttling device to remove the control valve signature which sometimes affects wind tunnel data (Lowe and Cumming, 1970).

The piping downstream of the regulating valves is protected from over pressure by a rupture disk.

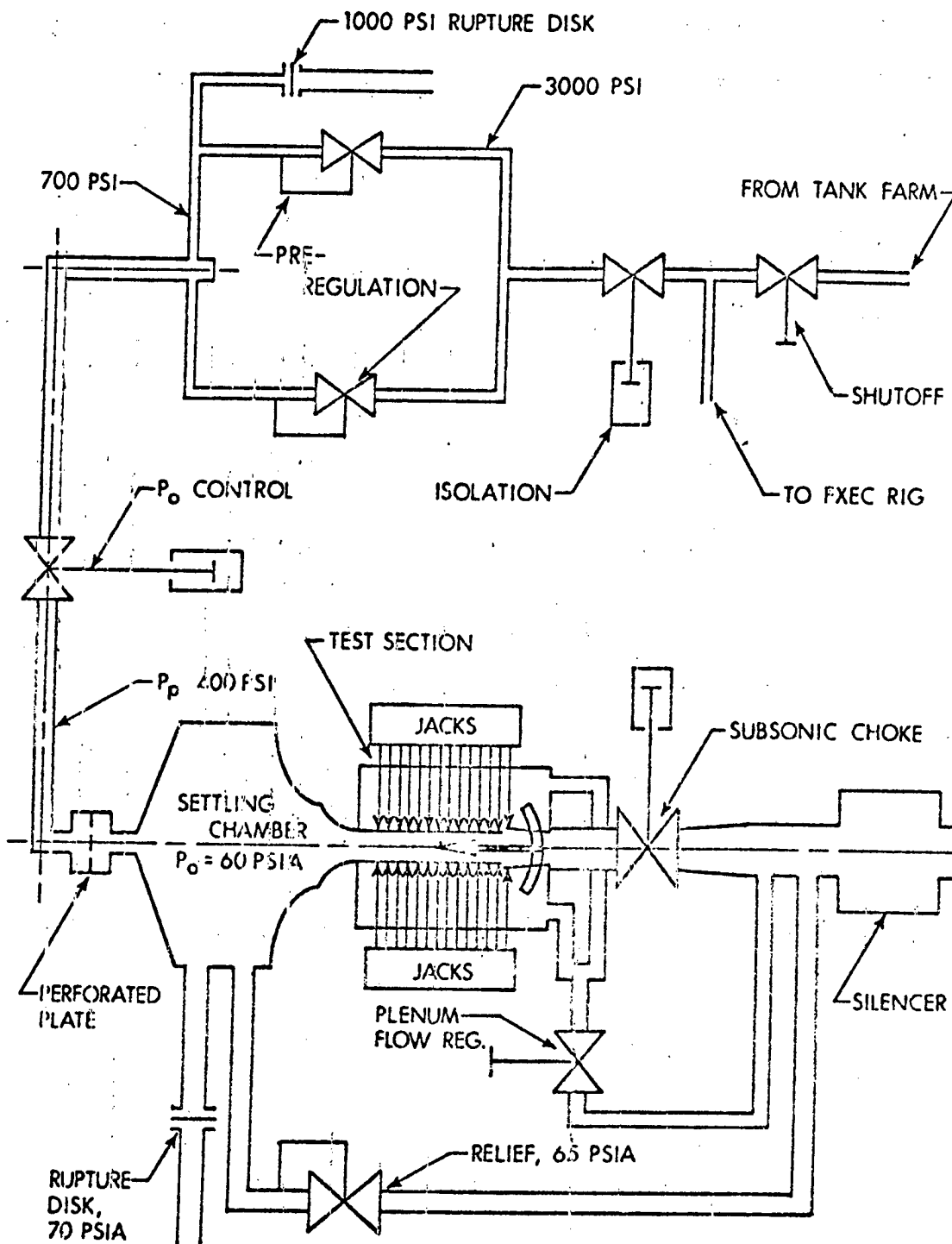


FIGURE 1 - Schematic of Facility

3. Test Leg. The test leg consists of a settling chamber, a three dimensional butterfly contraction section, a two dimensional nozzle, a test section, a rectangular to round transition section and a hydraulically operated butterfly style control valve. This control valve provides a means of choking the flow downstream of the nozzle for setting subsonic Mach numbers.

The settling chamber has a cross sectional area of thirty-one times that of the test section. It contains a conical baffle and screens to reduce turbulence in the flow.

Since numerous runs are planned with  $P_0$  close to the design pressure limit, two safety devices are incorporated in the settling chamber. The low pressure safety device is a relief valve which vents the settling chamber to exhaust line and silencer. Should the pressure continue to rise; a rupture disk, capable of handling a much larger mass flow, vents the settling chamber to atmosphere.

4. Exhaust Piping. The butterfly control valve is connected to a silencer with two foot diameter low pressure piping. For personnel safety, the stream exhausts vertically.

The plenum areas on each side of the test section are connected to a single manually operated butterfly style control valve. The control valve is used to modulate the plenum flow. The plenum flow can be returned to the stream at the silencer inlet when sufficient differential pressure exists. Lack of sufficient pressure drop would necessitate connecting the plenum line to existing vacuum source in the testing complex.

B. TEST SECTION. The test section wall deflection capability is shown diagrammatically in Figure 2 and is discussed further in Section III. The figure scale is not that of the pilot tunnel, and the deflection scale is exaggerated. The walls will be contoured by jacking to provide minimum wall interference in the model area at transonic speeds (Harney, 1973).

The upper and lower walls of the test section are each composed of nine very flexible rods, or elements. The elements begin at the entrance of the two dimensional nozzle and are carried through the test section to the transition section. A cross schematic of the test section is shown in Figure 3. The back side of the elements, away from the tunnel centerline, are sealed with triangular shaped strips which can be moved into close proximity with the elements. The seal strips are spring loaded against the elements to act as relief devices to protect against a high differential pressure on tunnel start. The seals can also be moved away from the elements to create a flow path into the plenum space behind the elements.

A crescent style model support is located at the downstream end of the test section. The upper and lower walls diverge at this point to compensate for the blockage of the model support.

At the model support area the seals pivot about a hinge at the downstream end of the elements and act as partial flaps. This provides a means for partially equalizing pressure across the elements. This also can provide a means of returning plenum flow to the stream.

Each element has ten individual jack points in the model area which are electro-mechanically operated. The walls will be deflected by jacks to provide



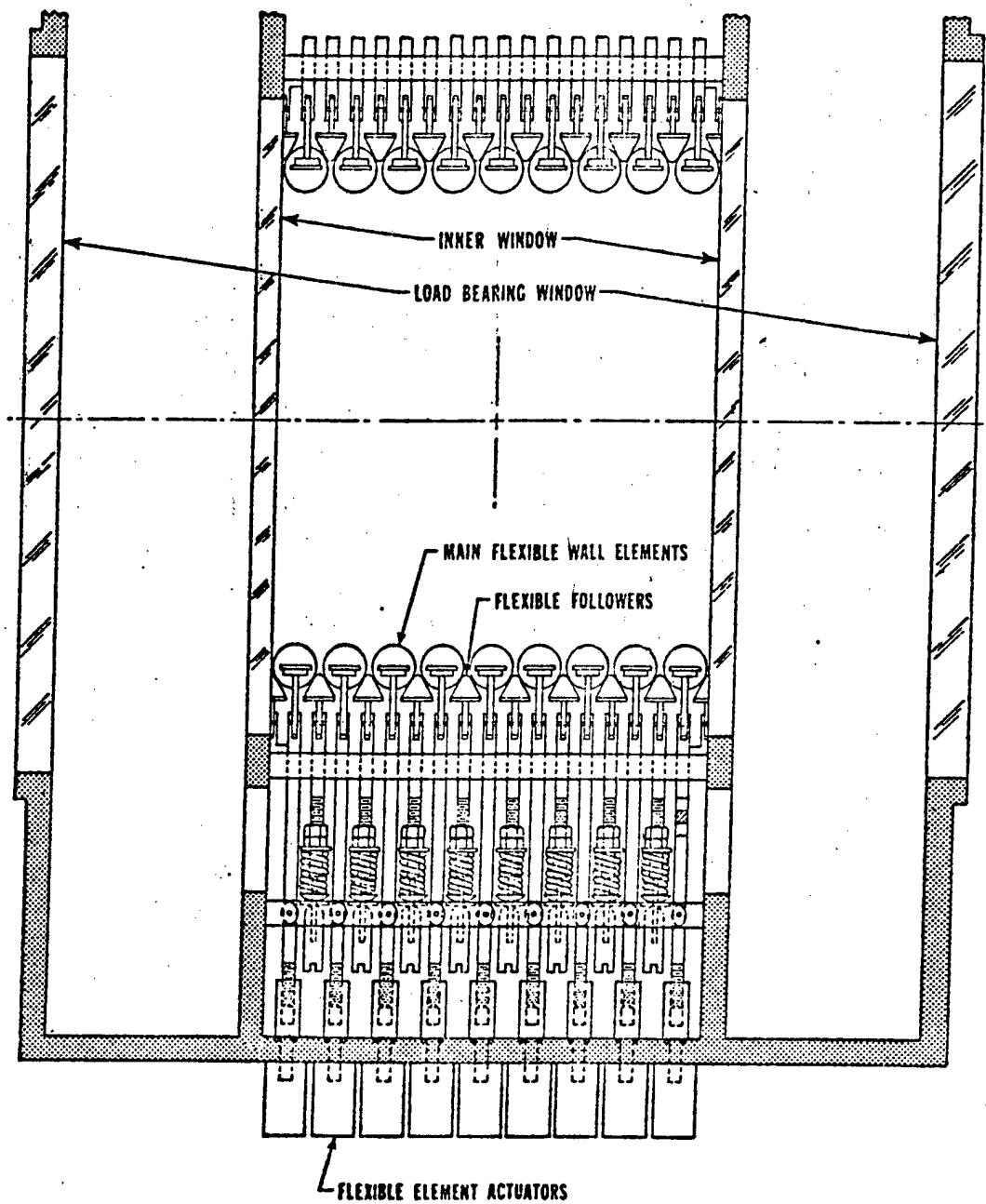


FIGURE 3 - Schematic Cross Section Through Test Section

both axial and lateral contouring. Between the nozzle throat and the model area the elements are connected together with three crossbeams per wall at manual jack stations. The manual jacks will be used to form a supersonic wall contour to develop Mach numbers up to 1.25.



### III. TUNNEL CIRCUIT DETAILS

#### A. AIR FLOW COMPONENTS AND CALCULATIONS

1. Components and Approach. Computations of flow and pressure drops for the tunnel begin at the settling chamber and nozzle and proceed upstream and downstream from that point. The maximum design settling chamber pressure, fixed by mechanical considerations at sixty psia, and the nozzle area, fixed by the aerodynamicist at nominally eighty-one square inches, determine the maximum tunnel flow. The maximum tunnel flow is then available for the sizing of pipe and valves. The temperature is not controlled and is assumed to be standard, 520°R.

Proceeding in the upstream direction, the inlet pressure to the perforated plate is computed first. This pressure becomes the stagnation pressure control valve exhaust pressure.

The size of the control valve and of components upstream of the valve were determined based on this pressure and the supply pressure. These components were originally sized for an installation of the tunnel in a different building. Therefore components and pipe sizes were selected to take maximum advantage of piping in place. The design also considered the need for a large flow, regulated air pressure source for another facility. Based on these constraints two regulating valves operating in parallel were selected. These valves reduce the supply pressure from 3000 psi to 1100 psi or less. A large size control valve which would pass the required flow with little pressure drop was selected. A low pressure air actuator was selected for this balanced control valve with the intention of pre-setting the valve and only fine tuning the control with flow on. However, since

other facilities use the stored air in the new location which was chosen, this valve is now being used to bring the tunnel on line directly and rapidly.

An economical design for the present installation, starting from scratch, would have dictated the use of a high pressure valve to control the flow with only one stage of control. However the use of suitable components which are already purchased is necessary for economy, and the original valves were retained. Since the piping from the regulators to the control valve is much shorter, the line losses are much lower and pressure fluctuations for flow versus no flow are much less, control with the valve is easier than it was for the original installation. The valve actuator has also been replaced with a hydraulic cylinder and servo-valve to provide a quick start capability which will conserve air.

With the control valve size fixed, its inlet pressure is determined by the desired valve opening at design conditions. A value of one-half open was selected to permit an adequate reserve of extra flow capacity for response.

The pressure setting for the regulating valves is equal to the sum of the control valve inlet pressure and the line losses between the regulators and the control valve.

The line connecting the regulators to the tank farm is existing and questionable in size. Its adequacy will be based on flow rate, i.e., the time required to reduce the source pressure below useable values. Should it prove inadequate a second line, in parallel, will be added.

Proceeding downstream from the settling chamber, at sixty psia total pressure, the flow will exhaust to atmosphere. With a pressure ratio of four, sonic flow will exist at some point in this section. To provide subsonic flow

at sixty psia total pressure in the test section, the sonic flow section must be located downstream of the test section. A butterfly control valve is used to choke the flow downstream of the test section. This style valve was selected because coarse valve configurations, such as gate valves, can show a dependence of subsonic Mach numbers on stagnation pressure, i.e., the valve flow coefficient varies somewhat with total pressure (Grunnet, 1974). Additionally the butterfly valve must not be choked with Mach 1.25 flow in the test section. This means that when the valve is wide open the flow area must be greater than the eighty-one square inches nozzle area or the expanded Mach 1.25 flow area of 84.8 square inches.

The valve was selected by examining the properties of butterfly valves ranging in size from twelve to sixteen inches. The flow through the valve in the design choke condition creates large forces on the disk. Holding the valve open against these forces requires a heavy shaft to provide the torque. The heavy shaft means a thick disk with less flow area in the full open position. It was necessary to use a sixteen inch valve with a class five shaft (two inch diameter). The full open area is approximately 84.7 square inches.

A silencer was incorporated to reduce the exhaust noise to below safe sound levels. The silencer is rated at an attenuation of 48 DB for noise at 134 DB and 100 Hz. Effectiveness diminishes with frequency to 5 DB attenuation at 31.5 Hz. The silencer is rated at a pressure drop of 0.3 psi at 102 lb/sec flow rate. The two foot diameter connecting line also has a low pressure drop.

The auxiliary flow lines must also be adequately sized. Most of these will be discussed under safety relief flow. Plenum flow results from moving the seal strips away from the elements creating a flow passage through which a low pressure downstream will cause flow. The maximum flow desired is only 12.5% of the main line

flow, and an existing ten inch butterfly valve was found to be adequate for throttling the flow. Manual presetting of the valve is presently planned but future operations may be under servo control.

A jury rig set up was fabricated and tested to verify the airflow quantities. The flow versus the seal motion, normal to the test section rod wall, is initially nonlinear as the seal begins to move away from the rods. After about 10% travel the characteristics become fairly linear until about 95% of maximum flow is reached. At maximum flow the spacing between the rods is the limiting factor rather than the seal travel.

2. Equations and Calculations. Flow through pipes is assumed to follow the Fanno Line (VanWylen, 1966) for compressible adiabatic flow with friction. This method of analysis was chosen for pipe sizing because it considers that compressible flow in a duct can enter at subsonic velocity and choke before the end of the duct is reached.

Flow through valves which have large pressure drop and low recovery was assumed to follow the standard valve equations which employ a capacity factor,  $C_v$ . The Fisher Controls equation for high recovery valves was used for the butterfly style subsonic choke valve.

Flow through the nozzle is assumed to result from still air in the settling chamber expanding isentropically through the nozzle. This is a good assumption with a well made nozzle and with the area ratio of settling chamber to test section of thirty to one.

(a) Supply line flow - The regulator inlet pressure which results from the four inch supply line losses is shown in Figure 4. The line is approximately 300 feet long, has an inner diameter of 3.5 inches, with six ninety degree elbows and two gate valves. The wide open regulator flow, no regulation, is shown in Figure 5. Data for the latter figure was calculated using characteristics provided by the valve manufacturer (Grove, 1969) and by extrapolating for subsonic flow.

Figure 5 shows that the minimum regulator inlet pressure to obtain the maximum continuous tunnel flow of 115 lb/sec is 1300 psi. Figure 4 shows that the minimum tank farm pressure to achieve 1300 psi regulator inlet pressure at that mass flow is 2100 psi. With a beginning pressure of 2600 psi, available storage of 8250 cubic feet, the run time at 115 lb/sec is 134 seconds assuming adiabatic expansion of remaining air. This is adequate since the desired run time is ten seconds to sixty seconds.

The pressure drop in the two four inch regulated pressure lines is 20 psi at 115 lb/sec and 700 psi inlet pressure, the regulator setting.

(b)  $P_o$  controlling flow - The equations for sonic and subsonic flow through the control valve are (Beard, 1969):

sonic

$$\dot{m} = (.00002122) K_1 \frac{C_v}{\sqrt{GT}} \frac{x}{2.5} P_1$$

$$\dot{m} = .142 \times P_1$$

(1)<sup>1</sup>

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1. Equations are numbered for use in Section III-C

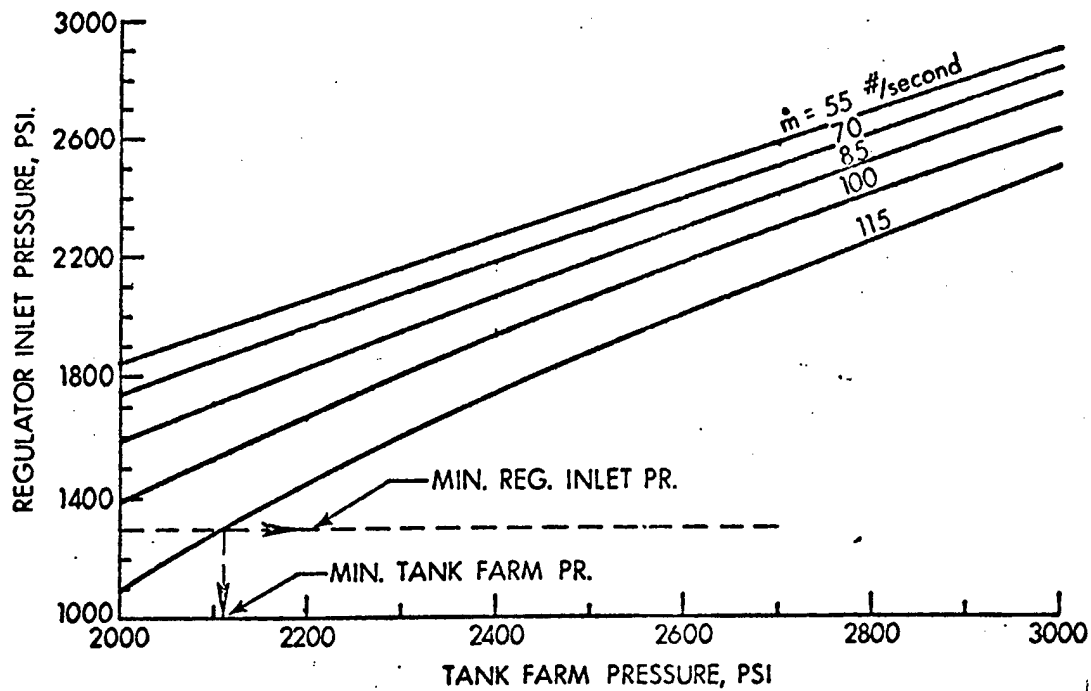


FIGURE 4 - High Pressure Supply Line Pressure Drop

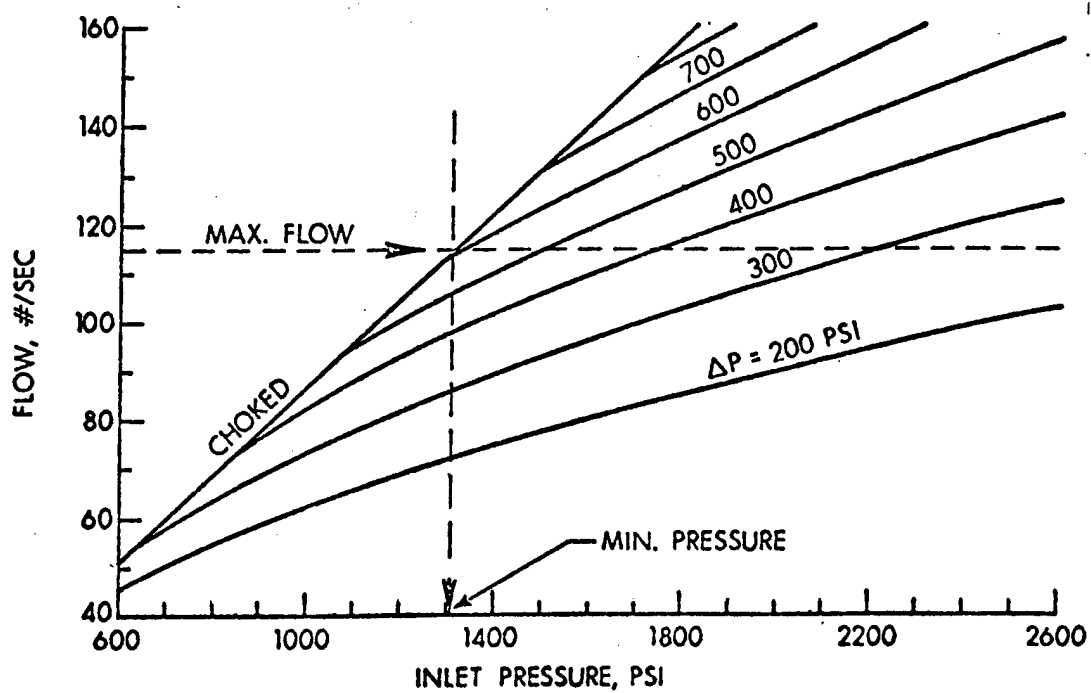


FIGURE 5 - Flow Capacity of Two Parallel Regulating Valves

subsonic

$$\dot{m} = (.00002122) K_2 \frac{C_v}{\sqrt{GT}} \frac{x}{2.5} \sqrt{P_1^2 - P_p^2}$$
$$\dot{m} = .167 x \sqrt{P_1^2 - P_p^2} \quad (2)$$

where .00002122 converts scfh to lb/sec

$$K_1 = 834$$

$$K_2 = 963$$

$$C_v = 466$$

$$G = 1$$

$$T = 520^\circ R$$

$$x = \text{stroke, } 2.5 = \text{full stroke}$$

$$P_1 = \text{inlet pressure} = 680 \text{ at design conditions}$$

$$P_p = \text{outlet pressure} = 400 \text{ at design conditions}$$

The flow through the perforated plate may be found with that choked orifice equation:

$$\dot{m} = K C \frac{AP}{\sqrt{GT}}$$
$$\dot{m} = .287 P_p \quad (3)$$

where C = a constant, taken as .6

$$K = .53 \text{ for air}$$

$$A = .41 (.785) (8)^2 = 20.6 \text{ for 41\% open holes}$$

$$P_p = 400 \text{ at design conditions}$$

(c) Test section flow - The test section mass flow may be found with the use of Chart 1, NACA 1135 (Ames Research Staff, 1953) or it may be found, as a function of Mach number and stagnation conditions, by combining certain of the equations in the report to give:

$$\dot{m} = P_o g \frac{A}{144} \frac{\rho_{std}}{P_{std}} \frac{T_{std}}{\sqrt{T_o}} 49M(1 + \frac{M^2}{5})^{-3}$$

$$\dot{m} = 3.271 P_o M (1 + \frac{M^2}{5})^{-3} \quad (5)$$

where  $P_o$  = stagnation pressure, psi

$A$  = area, square inches

$T_o$  = stagnation temp, 520°R

$M$  = Mach number

$g$  = gravitational constant

$\rho_{std} = .002378$  slug/ft<sup>3</sup>

$P_{std} = 14.7$  psia

$T_{std} = 520^\circ R$

At  $P_o = 60$  psia and  $M = 1$  the mass flow is 113 lb/sec. A nominal figure of 115 lb/sec is used.

The flow through the subsonic choke valve is (Jury, 1974):

$$\dot{m} = .00002122 \sqrt{\frac{T_{std}}{T_1}} C_g P_1 \sin \left[ \frac{3417}{C_1} \sqrt{\frac{\Delta P}{P_1}} \right]$$

$$\text{since } \sin \left[ \frac{3417}{C_1} \sqrt{\frac{\Delta P}{P_1}} \right] = 1 \text{ for choked flow} \quad (5)$$

and  $C_g = f(\theta)$  where  $\theta$ =valve position (Jury, 1974). Since standard temperature was assumed, then  $\dot{m} = .00002122 P_1 f(\theta)$ .

Since the expansion of air in a well made nozzle is a highly efficient process, then  $P_o = P_1$  and  $T_o = T_1$ . Setting Equations 4 and 5 equal to each other yields:

$$3.271 M(1 + \frac{M^2}{5})^{-3} = .00002122 f(\theta) \quad (6)$$



Equation 6 shows that the test section Mach number is a function of the valve position only.

(d) Plenum flow - Treating the plenum flow passages as valves, the flow coefficient,  $C_v$ , was found to be a function of the seal stroke,  $S$ , and the stroke length,  $L$ , times a constant experimentally determined to be 26. There are sixteen spaces between rods, each with a length of about sixty-four inches. The stroke for maximum flow is approximately .1 inch. The flow, using a subsonic orifice equation is:

$$\dot{m} = .00002122 K_2 C_v \sqrt{\frac{P_1^2 - P_2^2}{GT}}, \text{ where } C_v = 26 SL$$

For maximum total flow,  $S = .1$ ,  $L = 16$  spaces at 64 inches. Using  $G = 1$ ,  $T = 520^\circ R$  and  $K_2 = 963$  the total flow is

$$\dot{m} = 2.386 \sqrt{P_1^2 - P_2^2}$$

If  $P_1$  is taken as the free stream static pressure of 33.5 psia (at 60 psia  $P_0$  and .95 Mach number), and  $P_2$  is the pressure downstream of the subsonic choke valve, 18.2 psia, then the maximum flow disregarding line losses is 67 lb/sec. However, since the differential pressure across the rods at this flow is much greater than can be tolerated and the flow requirements are only about 11 lb/sec, it follows that the flow should be throttled. A butterfly control valve will be used for this purpose. This is the plenum flow regulating valve shown in Figure 1. The differential pressure required for full flow is about .5 psi and all seals opened fully. This is based on a constant  $P_1$  of 33.5 psia and  $M_{TOT}$  of twelve percent of 115 lb/sec main line flow. The plenum flow desired is not uniform down

the length of the tunnel. One of the advantages of this tunnel is that variable geometry can be combined with a form of variable porosity. Therefore the seals will not be uniformly opened and the anticipated differential pressure is 1 to 5 psi.

3. Safety Relief Flow Rates. The safety relief flow rates are based on static line conditions at maximum pressure and assumed sudden malfunction of valves or controls or system blockage. These flow rates are very short transients which decay rapidly as flow is established and line pressures drop. They represent worse conditions than can actually be realized.

(a) Regulated pressure lines - The nominal line pressure is 700 psi. The lines are four inch schedule forty and eight inch schedule eighty steel pipe. The safety factor is four at a maximum operating pressure of 1157 psi for Grade A steel pipe (T.O. 00-25-223). The lines are protected by a 1000 psi rupture disk, Figure 1. The disk flow is vented to atmosphere above the building. A steel plate above the vent line prevents rain water from entering and prevents debris from blowing into the air. An eight inch vent line is used because a four inch line would reduce the flow rate by either choking in the vent or causing subsonic flow across the ruptured disk.

Assume that the  $P_0$  control valve is closed and that both regulators fail to open exposing the system to the static supply pressure of 3000 psi. Assume that the disk ruptures, leaving an equivalent of a three inch round hole with a coefficient of .8. Simultaneous solution of the regulator and disk flow equations shows that the regulator flow is subsonic, the regulator and disk flow is 240 lb/sec, the line pressure in the 700 psi line is 1830 psi and the safety factor is 2.5.

(b) Settling chamber and test section - The settling chamber and the original test section components were designed by NACA, Langley Field, Virginia in the 1950's for a pressure of 45 psig, and were tested to a pressure of 90 psig. New components have been designed for the same pressure, with safety factors of four or more.

The maximum flow into the settling chamber occurs when the regulated line pressure is approaching 1000 psi, the four inch disk has not ruptured and the  $P_0$  control valve fails wide open. Simultaneous solution of the control valve and perforated plate flow equations shows that the pressure upstream of the perforated plate is 810 psi and the flow is 225 lb/sec. The worst case is when the subsonic choke valve is also closed and all the flow must be exhausted through relief devices. Two such devices are in use. One is a relief valve set at 65 psia. The valve has an eight inch inlet, twenty-six square inch flow area, ten inch outlet and expands into a twelve inch pipe which routes the flow through the tunnel exhaust and silencer. The second is a twelve inch, 70 psia rupture disk. An iterative solution shows that the pressure rises to 120 psig for a safety factor of 1.5. Flow through the ruptured disk is subsonic at 175 lb/sec. Flow through the relief valve is sonic at 50 lb/sec. The safety factor is inadequate, however, it was calculated for a line pressure of 300 psi above the set point and for simultaneous failure of two valves. Interlocks will be used to increase the safety of the facility.

(c) Interlock system - The interlock system is designed to shut off the high pressure air supply, and hence shut the facility down, in the event of: overpressure of the regulated pressure lines or settling chamber, operation of

relief devices, hydraulic system failure, or the choke valve closed beyond normal operation. The devices chosen to shut the facility down in an emergency are the dome loaded regulating valves and the isolation valve because they can be made to close in the absence of electrical or hydraulic power. The isolation valve is operated via a normally closed pilot valve and air pressure. The domes of the regulators are pressurized with an electrically operated dome loading regulator which bleeds off the dome pressure with no electric power applied.

In the design of the system, consideration was given to the fact that in protecting against one hazard, another hazard can be created. An example is possible damage to the model if the model is not at zero degrees pitch and if the tunnel is running supersonically and goes out of flow. This occurs when the normal shock passes through the test section. Because of this hazard to the model, the interlocks for most events incorporate a short time delay to allow the model to pitch to zero and also start the pitch motion. Closing of the choke valve however initiates immediate tunnel shutdown because, as seen in Figure 8, the tunnel will run subsonically with the valve less than fifty-three degrees open.

#### B. HYDRAULIC VALVE POSITIONING SYSTEMS

Hydraulic power was selected for valve positioning because of the need for high performance in a short running time blowdown tunnel. Two valves, the stagnation pressure control valve and the subsonic choke control valve, will be hydraulically operated. A third valve, the plenum chamber suction control valve, may be driven in the future although initially the valve will be manually preset.

The stagnation pressure,  $P_0$ , control valve will be initially set up for closed loop valve position control. This is the checkout and pre-regulator setting

purposes. The subsonic choke valve will be closed loop position controlled for purposes of open loop control of subsonic Mach number based on prior tunnel calibration data.

A block diagram for the servo valve and load is shown in Figure 6. A Moog Series 73 valve, rated at 5 gpm is used for each system. The servo valve second order model, shown in Figure 6, is accurate to about 300 radians/second with a natural frequency,  $\omega_{sv}$ , of 880 radians per second and a damping ratio,  $\omega_{sv}$ , of 0.9 (Moog Catalog). The flow gain is  $K_1$ , with units of cubic inches per second (cis) per ma, and is discussed as a part of load dynamics.

The viscous damping factor,  $B$ , usually low, is assumed to be zero. Applying block diagram reduction techniques to the load portion of Figure 6 gives a transfer function of

$$\frac{x}{Q}(s) = \frac{1/A}{\frac{K_3 M}{A^2} s^3 + \frac{K_2 M}{A^2} s^2 + \frac{A^2 + K_3 K_1}{A^2} s + \frac{K_2 K_1}{A^2}}$$

where  $x$  = Piston travel, inches (load travel=piston travel or is referred to piston travel)

$Q$  = Load flow, cis

$A$  = Net piston area, square inches

$K_3$  = Compressibility coefficient,  $V/B$ , in cubic inches/psi

(Johnson and Schmid, 1956)

$V$  = Volume under compression,

$B$  = Bulk modulus, 170,000 psi for mineral based oil

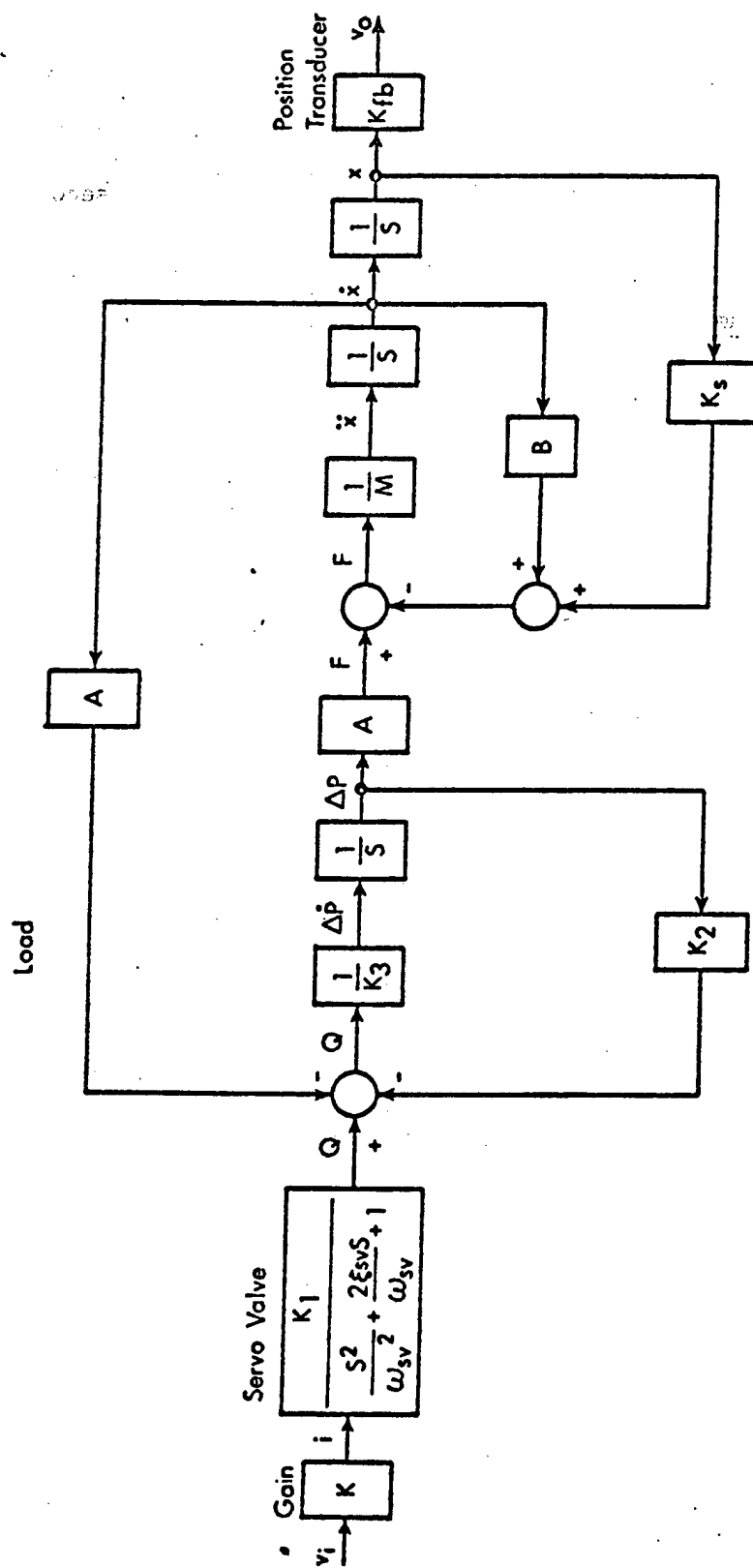


FIGURE 6 - Block Diagram of Servovalve and Load

$M$  = Load mass, lb/inch/sec<sup>2</sup>

$K_s$  = Load spring, lb/inch

$K_2$  = Pressure gain, cis/psi (often referred to as  $C_1$ , with

$K_2 = C_1 + C_2$ ,  $C_2$  being a laminar leakage coefficient)

The arrangement of the transfer function provides a single equation for use with either a pure mass load or for a load with a parallel spring. The low frequency behavior,  $\omega$  approaching zero, is that of an integrator or a gain of  $A/K_2K_s$ . For most systems  $A^2$ , is much greater than  $K_sK_3$  and at high frequency the constant term in the denominator is negligible compared to the other terms. Then at high frequencies the denominator is the time constant form of a second order system with a free integration whose natural frequency and damping ratio are given by:

$$\omega_L = \sqrt{\frac{A}{K_3M}}, \quad \xi_L = \frac{K_2}{2A} \sqrt{\frac{M}{K_3}}$$

The constants  $A$ ,  $K_3$ ,  $M$  and  $K_s$  are different for each system and are discussed separately. The constants  $K_1$  and  $K_2$  are taken from a graph of the servo valve pressure, flow and current characteristics, Figure 7.

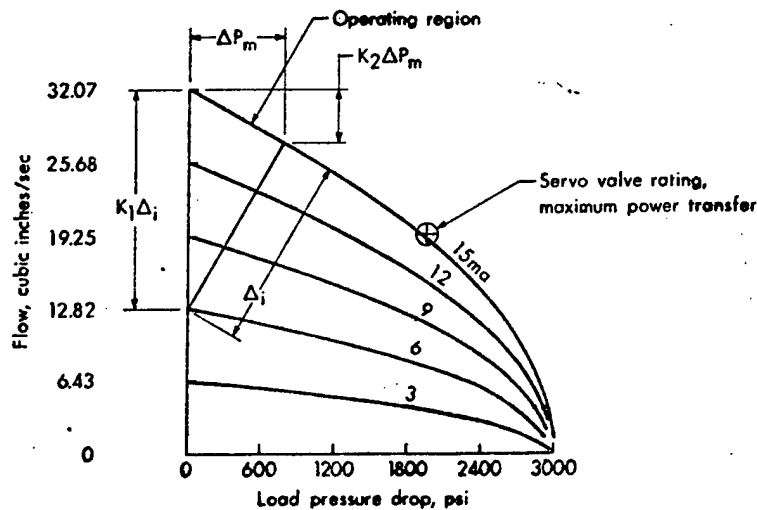


FIGURE 7 - Characteristics of 5 gpm Servo valve with 3000 psi Supply Pressure

$$K_1 = \left| \frac{\partial Q_m}{\partial i} \right|_{P_m = \text{const}} = \frac{K_1 \Delta i}{\Delta i}$$

$$K_2 = \left| \frac{\partial Q_m}{\partial P_m} \right|_{i = \text{const}} = \frac{K_2 \Delta P_m}{\Delta P_m}$$

Both systems are lightly loaded, operating in the linear region at the left of the graph. The flow gain,  $K_1$ , is nearly constant in this region at 2.13 cis/ma. The pressure gain,  $K_2$ , is not constant, varying from nearly zero at the origin to  $5.85 \times 10^{-3}$  cis/psi at the top of the graph, marked "Operating Region". Hansen (1961) gives a typical order of magnitude value of  $10^{-3}$  cis/psi. Therefore a median value of  $3 \times 10^{-3}$  is used.

1. Pressure Control Valve. The load mass, a linear motion air valve plug, is .16 lbs/in/sec<sup>2</sup>. The actuator is a 2½ inch bore, 1 inch double ended rod cylinder with an area of 4.12 square inches. The stroke is 2.5 inches. With the piston centered, the volume under compression on one side is 5.15 cubic inches. The constant resisting force, whence  $K_s = 0$ , is caused by differential air pressure acting over a small unbalanced area on the virtually balanced double acting air control valve plug and is 800 pounds. The load pressure drop is  $800/4.12 = 194$  psi. The flow at 194 psi and 15 ma, Figure 7, is 31 cis for a maximum velocity of  $31/4.12 = 7.52$  in/sec. The pressure drop across the valve is  $3000 - 194 = 2806$  psi.

The compressibility coefficient,  $K_3$ , is  $3 \times 10^{-5}$ . The natural frequency,  $\omega_L$ , is 1880 radians per second. Multiplying the servo valve and load transfer functions together would produce a characteristic equation whose roots would show a frequency of 880 radians/second. In other words, the servo valve cannot control a mass system at that high a frequency. The performance must be downgraded so that



attenuation occurs before  $\omega_{sv}$ . Then the high frequency load characteristics are reduced to almost nothing and the load equation can be simplified to:

$$\frac{x}{Q}(s) = \frac{1}{AS}$$

Crossover should occur at about  $1/3 \omega_{sv}$  (Broome, 1963) or 300 radians/second. With no poles or zeros to the left of  $\omega_c$ , then  $\omega_c = K_v$ , the velocity coefficient. The gain will be adjusted to achieve full velocity. With a feedback coefficient,  $K_{fb}$ , of six volts/inch, the amplifier gain is:

$$K = \frac{(K_v)(i_{max})}{(K_{fb})(Vel)} = \frac{(300)(15)}{(6)(7.5)} = 100 \text{ ma/volt}$$

This is a well behaved second order system with a free integration. It's relative stability can be readily shown to be adequate. Slowing the system down to match the servo valve is only a degradation of performance in the sense of following an oscillatory input. It will be shown in Section III-C that the crossover frequency will be further reduced but that a high velocity is desirable and is retained.

2. Subsonic Choke Valve. The load mass is a rotary disk in a butterfly style valve which is driven by a crank, and linear hydraulic cylinder. Operating characteristics of the valve are shown in Figure 8, Curves A through G.

Curve A shows the rated flow through the valve. Curve B shows the Mach number at various disk angles. Past the angle for Mach 1 the flow cannot be choked in the valve and the valve flow will be subsonic. Tunnel Mach number will then depend on other conditions.

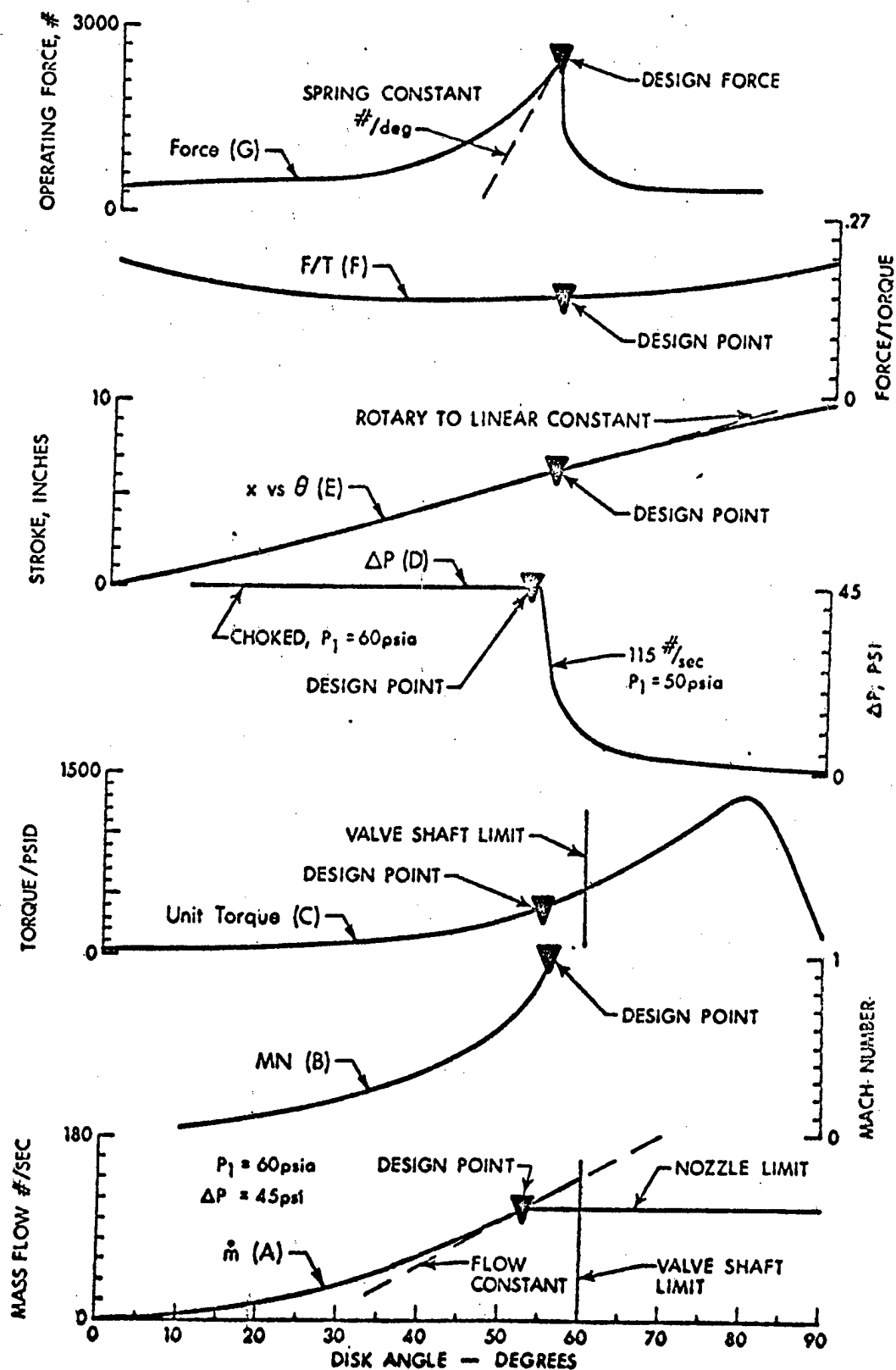


FIGURE 8 - Characteristics of the Butterfly Valve Throttling the Tunnel Airflow

Curve C shows the typical unit torque characteristics (Beard, 1969) of butterfly valves. The high torque peak at eighty degrees will not be experienced by the valve because the differential pressure drops as shown on Curve D when the valve unchokes.

Curve E shows the stroke versus valve disk angle. The feedback potentiometer is rotary, which introduces a slight nonlinearity in the analysis. The coefficient is selected at the design point.

Curve F shows the unit force per unit torque which when multiplied by the values of Curves C and D provides the operating force curve shown on Curve G. The piston stroke, crank length and initial angle were chosen for minimum F/T and maximum linearity of F/T in the Mach number range of .3 to .9. The operating forces to move the disk shown on Curve G provide a variable mechanical spring constant. Since the forces are caused by airflow, and the valve will be operated with and without flow, the spring constant varies from zero to a maximum at the design point.

Performance required of the control system, other than stability, is mainly accuracy. As Mach 1 is approached, the curve of Mach number versus disk angle gets very steep as shown on Curve B. The valve is normally pre-set before a run with possibly minor adjustments made during a run. Hence operating speed is not a problem.

(a) Design analysis - The valve disk is approximated by a steel six inch diameter hub, sixteen inch long plus a one inch thick, fifteen inch diameter disk, having a combined polar moment of inertia,  $J$ , of 3.20 inches lb sec<sup>2</sup>. The linear

motion diagram of Figure 6, and the resulting load transfer function, apply to this system if the rotating components are referred to the linear. The rotating mass can be referred to the linear piston operation by:

$$J\ddot{\theta} = Mr\ddot{x}$$

where  $r$  = radius =  $\frac{1}{F/T}$  from Figure 8, Curve F

$F/T = .14$  at design point and

$$\ddot{x} = .1250$$

since  $x = .1250$  at design point.

$$\text{Then } M = 3.58 \text{ lb/in/sec}^2$$

The hydraulic cylinder has a  $2\frac{1}{2}$  inch bore, a 1 inch rod and is pivoted at one end. The average area,  $A$ , is 4.51 square inches. Using the average area and the design load resisting force, 2800 lb, the load pressure drop is 621 psi. Then at maximum current, from Figure 7 the flow,  $Q$ , is 28.2 cis and the velocity,  $Q/A$ , is 6.25 in/sec. This is an approximation with an unbalanced piston area. The velocity can be calculated from a balance of forces. The valve is rated by Moog at 5 gpm at 1000 psid valve drop. The drop through each side of the valve is then

$$\Delta P = 500 \sqrt{\frac{19.25 \text{ cis}}{AV}}$$

where  $A$  is area,  $\text{in}^2$  and  $V$  is velocity, in/sec. Using 3000 psi supply,  $A_B$  for the blind end area and  $A_R$  for the rod end area, a balance of forces gives:

$$V = \frac{19.25}{A_B} \sqrt{\frac{3000A_B - \text{LOAD}}{500 A_B \left[ 1 + \left( \frac{A_R}{A_B} \right)^2 \right]}} = 6.37 \text{ in/sec}$$

The velocities found by the two different methods differ very little because the ratio of rod area to blind area is .84 or nearly unity. An average piston area will be used and the areas assumed equal to avoid complicating the analysis.

At mid stroke the volume of oil under compression on one side is 22.51 cubic inches, neglecting the rigid connecting tubing. The compressibility coefficient,  $K_3$ , is then  $1.33 \times 10^{-4}$ . The natural frequency,  $\omega_L$ , is 207 radians per second. Using the same rule of thumb by Broome, crossover occurring at about  $1/3 \omega_L$ ,  $\omega_c$  should be about 70 radians per second. With an attenuation of 18 DB per octave after 207 radians per second, the attenuation is so great at 880 radians per second that the servo valve characteristics can be ignored. The load dynamics dominate the performance.

The damping ratio,  $\omega_L$ , is .054. Using a small tube to bypass fluid from the piston inlet to exhaust can improve the damping ratio. A flow of 2 gpm, or 7.69 cis at 1000 psid can be tolerated. Since the load pressure drop is 621 psid, the maximum quiescent flow, under load, is 1.24 gpm. A 10 gpm pump is available for supplying leakage flow plus a small accumulator for supplying peak demand of 17 gpm for less than 1.5 seconds.

At 7.69 cis per 1000 psid, the crossport leakage coefficient is  $7.69 \times 10^{-3}$  cis/psi. This improves  $K_2$  from  $3 \times 10^{-3}$  to  $10.69 \times 10^{-3}$ . The damping ratio increases from 0.54 to .192.

The feedback coefficient,  $K_{fb}$ , is 1.33 volts/inch at the operating point. the gain to operate at a  $K_v = \omega_c = 70$ , with no poles or zeros to the left of  $\omega_c$ , and at the calculated velocity of 6.25 inches/second is:

$$K = \frac{(K_v)(1_{\max})}{(K_{fb})(V_{\max})} = 126 \text{ ma/volt}$$

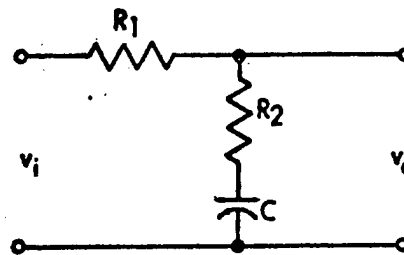
The steady state gain with a spring constant of 1800 lb/in is:

$$\frac{v_o}{v_i} = \frac{\frac{KK_1K_{fb}}{A}}{\frac{K_2K_s}{A^2}} = 137$$

To provide separate adjustment of high and low frequency gain, a lag compensator is added and the gain is increased by 75%. The amplifier gain becomes 220 ma/volt, the loop gain approximately 140 and the steady state gain with 1800 lb/in spring constant of 146. The compensator provides a gain margin at  $\omega_L$  of 2. Additional gain margin is provided by the neglected and somewhat nebulous load damping. The transfer function for the lag compensator is:

$$\frac{v_o}{v_i} = \frac{R_2 CS + 1}{(R_1 + R_2)CS + 1}$$

Where  $C = 5\mu\text{f}$   
 $R_1 = 236\text{K}\Omega$   
 $R_2 = 50\text{K}\Omega$



An open loop frequency response plot of the system is shown in Figure 9 with spring constants of 0, 500 and 2000 lb/in. Also shown, labeled "before",

is the high frequency portion of the system prior to the addition of the lag filter, gain increase and crossport leakage. The transfer function plotted is:

$$\frac{v_o}{v_i}(s) = K \frac{(\tau_1 s + 1)}{(\tau_2 s + 1)} \quad (2.13)$$

$$\frac{1/4.51}{(2.34 \times 10^{-5} s^3 + .176 K_2 s^2 + (1 + 6.54 \times 10^{-6} K_s) s + .049 K_2 K_s)} \quad (1.33)$$

where the values used are:

	INITIAL	FINAL SYSTEM
K	126	220
$\tau_1$	0	125
$\tau_2$	0	1.43
$K_2$	$3 \times 10^{-3}$	$10.79 \times 10^{-3}$
$K_s$	0	0, 500, 2000

The crossport leakage decreases the maximum velocity under load from 6.25 in/sec to 5.19 in/sec. The leakage also dissipates .73 horsepower under load. The heat dissipated results in a gradual oil temperature buildup which is not a problem for a blowdown tunnel.

Accuracy is difficult to address from theory. First it must be assumed that the two pivots have minimal clearances and that the typically bothersome keyway has no clearance, possibly a light drive fit on the sides. The possibility of lost motion between the positioning cylinder and the measuring point can introduce other control problems besides accuracy. With no lost motion, there are several ways to attempt to assess accuracy.

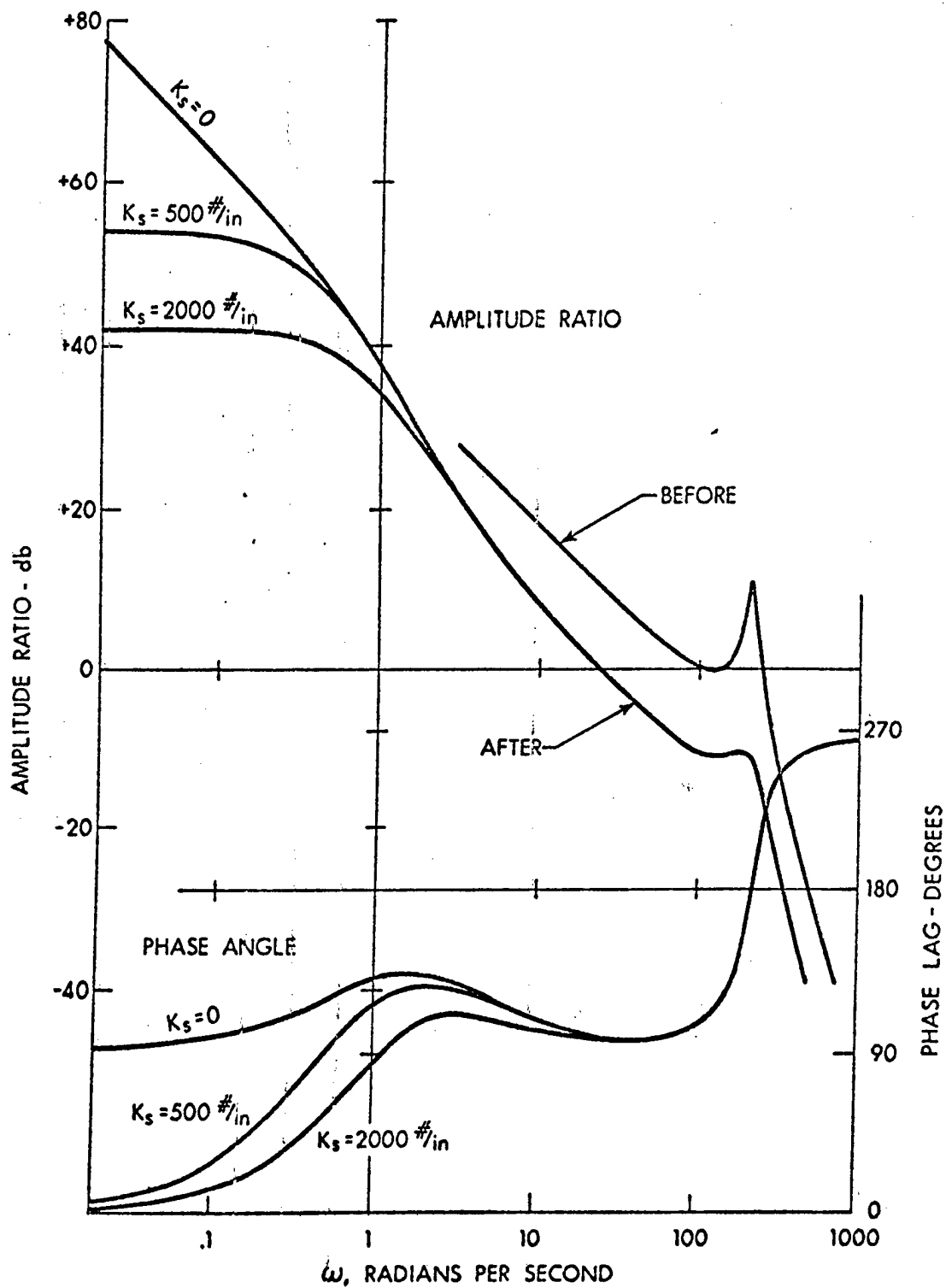


FIGURE 9 - Butterfly Choke Valve Frequency Response



The steady state characteristics are that of a proportional control system when tunnel air is flowing. Proportional control has an error of the step command divided by one plus the loop gain. If the step is 6.3 inches then the error is .033 inches. However it is probable that only the last half inch is slow enough to be considered a first order response. In any event, the .033 inches error produces 7.7 ma, half scale, from the amplifier, which is certainly going to move the load.

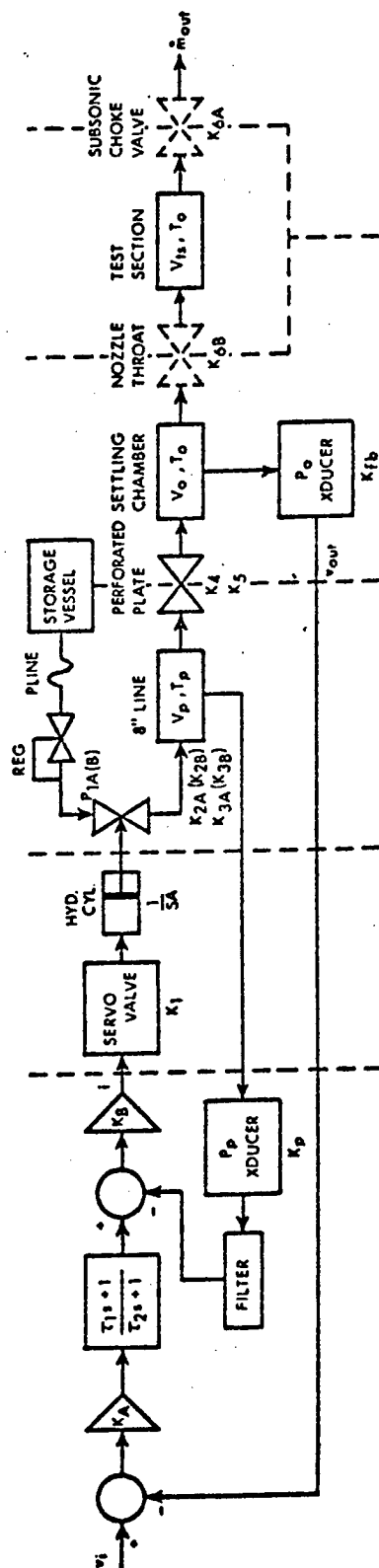
If the disk is assumed to be exactly positioned and then the load is applied, the piston will have to move .02 inch to correct for oil compression. The proportional droop is then .0001 inch.

If it requires a 3% of maximum signal to move the servo valve (an assumption of Broome), then the error which can be sustained is .002 inch with the amplifier and feedback gains in use. This represents one minute of angle on the disk. It is difficult to read a precision inclinometer closer than two minutes of angle. Therefore the accuracy is probably as good as can be reasonably put to use.

### C. PRESSURE CONTROL SYSTEM

#### 1. Description and Coefficients

(a) Description of system - This is a dual volume system which is modulated by the  $P_o$  control valve. Position control of the  $P_o$  valve for check-out purposes was discussed under Section II-B. A schematic diagram and a block diagram of the system is shown in Figure 10 and 11, respectively. The constants used in the figures are discussed later and given in Table I.



**FIGURE 10 - Pressure Control System-Hardware Schematic**

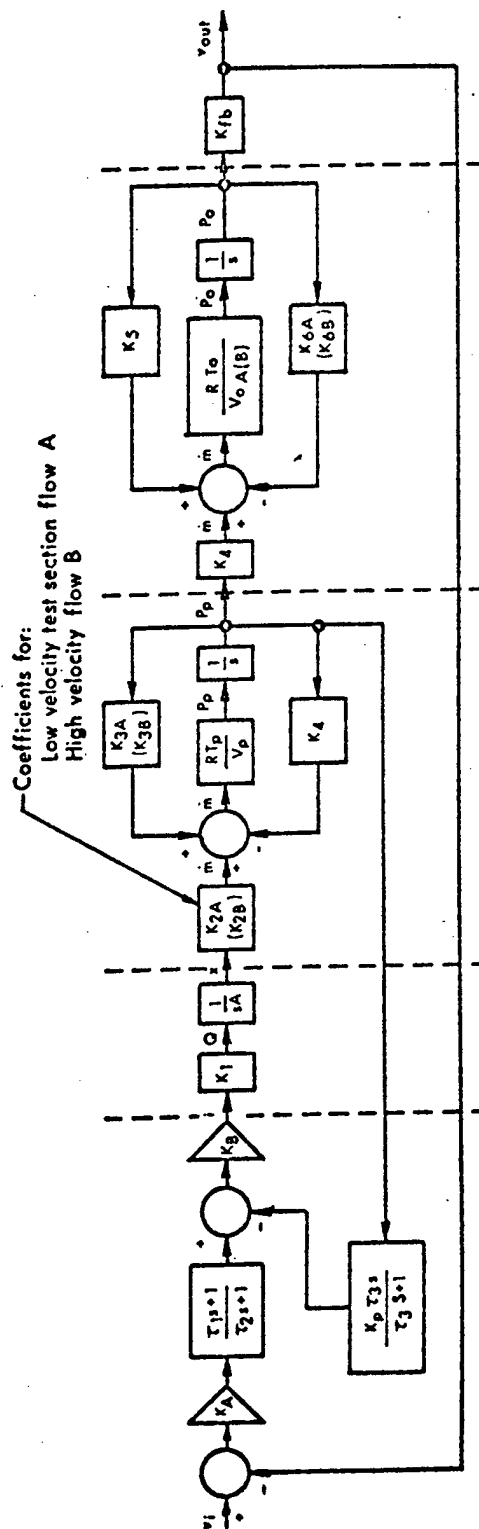


FIGURE 11 - Pressure Control System-Control Block Diagram

It is assumed that the flow rate is sufficiently high to cause choked flow at the perforated plate and at either the nozzle throat or the subsonic choke valve. The choice of the latter choke point depends on Mach number and not pressure. This assumption is valid when stagnation pressure,  $P_o$ , is above 30 psia since the exhaust is to atmosphere. The pressure upstream of the perforated plate,  $P_p$ , must be greater than 60 psia since the exhaust is to  $P_o$ . The justification for this assumption is twofold. Firstly, the choked flow at  $P_p$  of 60 psia is only 17 lb/sec or 15% of maximum. This is a starting condition of little interest. Secondly, a negative feedback coefficient which is zero based on the assumption, is non-zero when the assumption is invalid. Since negative feedback has a stabilizing effect, the system stability increases when the assumption is invalid.

A perturbation analysis is required due to nonlinearities. At Mach 1 or above the system is inherently more stable due to the higher mass flows which produce a greater negative feedback. However at this time, with 60 psia controlled pressure, the control valve requires a higher velocity to stay within the linear model because the valve is not choked and there is a downstream pressure effect. At Mach .3, the lowest Mach number of interest, the reverse is true. Inherent stability is less but the valve velocity requirements are less and the large power application which can be demanded by phase lead type of compensation can be realized.

(b) System coefficients and characteristics - The constants and coefficients which apply to Figure 11 are tabulated in Table I. Some of the coefficients are partial derivatives of equations numbered in Section III-B, with respect to the variable listed in the table.

TABLE I  
CONSTANTS AND COEFFICIENTS FOR PRESSURE CONTROL

QUANTITY	VALUE	APPLICABLE FOR MACH NUMBER OF	SOURCE AND COMMENTS
P <sub>1a</sub>	695 psi	.3	Constant
P <sub>1b</sub>	669 psi	1	Constant
R	12 x 53.3	.3, 1	Constant
T <sub>p</sub>	520°R	.3, 1	Constant
T <sub>o</sub>	520°R	.3, 1	Constant
V <sub>p</sub>	3317 cu in	.3, 1	Constant
V <sub>0A</sub>	215,800 cu in	.3	Constant
V <sub>0B</sub>	227,260 cu in	1	Constant
K <sub>2A</sub>	96.3 lbs/sec/in	.3	Equation 1, Var X, Sonic Valve Flow
K <sub>2B</sub>	86.2 lbs/sec/in	1	Equation 2, Var X, Subsonic Valve Flow
K <sub>3A</sub>	0	.3	Equation 1, Var P <sub>p</sub> , Sonic Valve Flow
K <sub>3B</sub>	-.151 lbs/sec/psi	1	Equation 2, Var P <sub>p</sub> , Subsonic Valve Flow
K <sub>4</sub>	.287 lbs/sec/psi	.3, 1	Equation 3, Var P <sub>p</sub> , Sonic Perforated Plate Flow
K <sub>5</sub>	0	.3, 1	Equation 3, Var P <sub>o</sub> , Sonic Perforated Plate Flow
K <sub>6A</sub>	.933 lbs/sec/psi	.3	Equation 4, Var P <sub>o</sub> , Sonic Choke Valve Flow = Subsonic Nozzle Flow
K <sub>6B</sub>	1.927 lbs/sec/psi	1	Equation 4, Var P <sub>o</sub> , Sonic Nozzle Flow

The system of Figure 11 contains two first order systems in cascade. Applying block diagram algebra to the first system, for variable  $x$  to variable  $P_p$ , and using the A subscripts, results in a transfer function of the form:

$$\frac{P}{X}(s) = \frac{\frac{K_{2A}}{K_4 - K_{3A}}}{\frac{V_p}{RT_p} \frac{s}{(K_4 - K_{3A})} + 1} = \frac{K}{\frac{s}{\omega} + 1}$$

where  $K$  = steady state gain

$\omega$  = break frequency

Using this form of the transfer function, and the values given in Table I, the characteristics have been computed and are given in Table II.

TABLE II  
PRESSURE SYSTEM CHARACTERISTICS

NAME	MACH NUMBER	GAIN	BREAK FREQUENCY
Flow into $V_p$	.3	335.5 psi/in	28.78
Flow into $V_o$	.3	.308 psi/psi	1.438
Flow into $V_p$	1	196.8 psi/in	43.92
Flow into $V_o$	1	.149 psi/psi	2.820

(c) Valve positioning coefficients - It was shown in Section II-B, that the load dynamics of driving the valve plug mass may be simplified to  $1/SA$ , where  $A$  is 4.12 square inches. Similarly, based on the break frequencies given above, the output signal will be greatly attenuated before the servo valve natural frequency,  $\omega_{sv}$ , is reached. Therefore the servo valve may be approximated by a pure gain of 2.13 cis/ma.

(d) Performance requirements - A high degree of accuracy is required so that tunnel conditions are repeatable. This can help in troubleshooting data problems using raw data. Reduced data is typically non-dimensionalized.

The maximum steady state airflow of the tunnel is more than fifty times the rate of which the storage cylinders can be recharged. This means that once an appreciable mass flow has started the tunnel should be on condition in a minimum time. In addition, valve opening and closing should be smooth and should require a couple seconds to avoid water hammer in the supply lines. This requirement is translated to mean three to four seconds from start to operating condition and 2.5 second shutdown. Time at operating condition may be as little as ten seconds if model attitude and tunnel walls are not changed, or it may be much longer.

Operation should be simple. It is preferred, but not required, that no controller adjustments are required in changing from one Mach number to another.

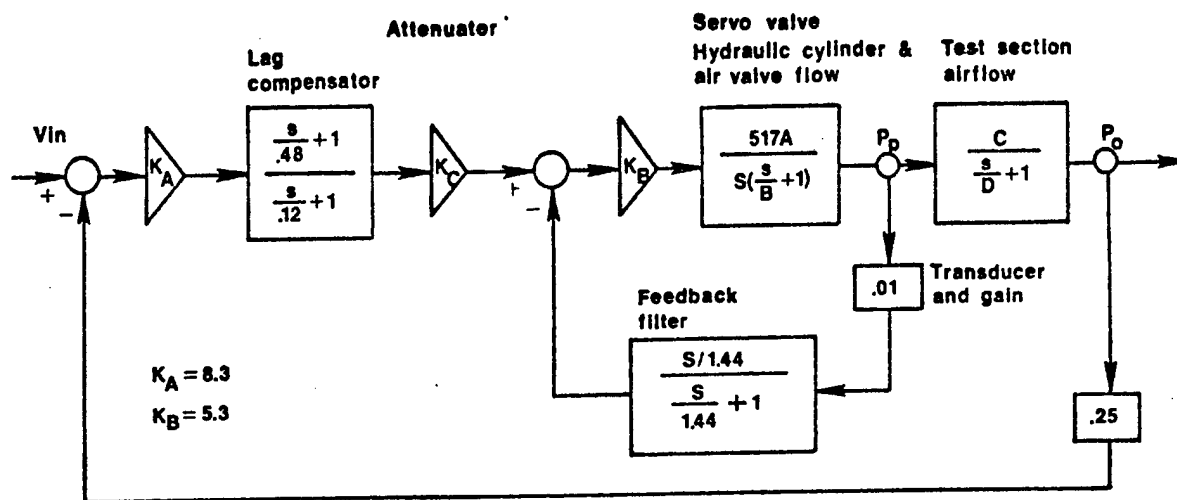
2. Frequency Domain Design. The system is partially compensated by feeding the intermediate pressure,  $P_p$ , back through a high pass filter in a minor loop. This is a phase lead type of compensator which has several advantages over phase lead compensation in the forward path. It provides a fixed zero but a pole which increases in frequency as gain is increased. It provides a means of independently setting the gain margin, using the error amplifier gain, and the phase margin, using the minor loop feedback gain. It also provides a means of dampening oscillation in  $P_p$ .

Lag compensation is also used to permit a gain increase and improve the steady state accuracy. Theoretically this should not be required because the free integration provided by the hydraulic cylinder provides zero steady state error. In a real system however, the presence of unknown forces such as stiction can modify the integral effect at steady state to that of a gain. The accuracy requirement and the servo-valve performance determine the gain. The strain gage pressure transducer, used for feedback, is accurate to about .25% at 60 psia. The servo system will not degrade the performance if it has an accuracy of .15% of 60 psia, or .0225 volts at .25 volts per psi. If the servo valve requires 3% of the 15 ma full scale to move, then .0225 volts into the error amplifier should provide .45 ma out. The required gain is 20 ma/volt. A gain of 44 will be used at Mach 1.

The lag compensator is not intended to provide an accurate ramp following capability because the large capacitor required would make the system sluggish in settling down to a steady state value. A ramp input will be used, however an input prefilter will be used to round the corners of the ramp and provide smoother operation and decreased settling time at the operating pressure.

A condensed block diagram of the system is shown in Figure 12. Additional system gains and pole locations for Mach .1 operation are also shown. The minor loop gains and feedback filter were selected through the use of a Nyquist diagram plotting program. Values for the Mach 1 case were adjusted until the minimum values of both gain margin and phase margin as suggested by Doebelin (1962) were achieved. These are 2.5 and 30 degrees respectively. The forward path gains,  $K_A$  and  $K_B$  on Figure 12, were then adjusted until adequate response was achieved.

(See "Time Domain Simulation"). Nyquist plot of the open loop frequency response of the system shown in Figure 12, with the attenuator set at 1, is shown in Figure 13A.



$MN$	$A$	$B$	$C$	$D$	$K_C$
.1	346.2	28.78	.883	.501	.525
.3	335.5	28.78	.308	1.438	.619
1	196.8	43.92	.149	2.820	1

FIGURE 12 - PRESSURE CONTROL SYSTEM-CONDENSED BLOCK DIAGRAM

The attenuator of Figure 12 is a potentiometer which is geared to the choker valve shaft. The potentiometer has special taps at the approximate angles for Mach .1, .3 and 1.0 operation of the choker. Shunting resistors will be used



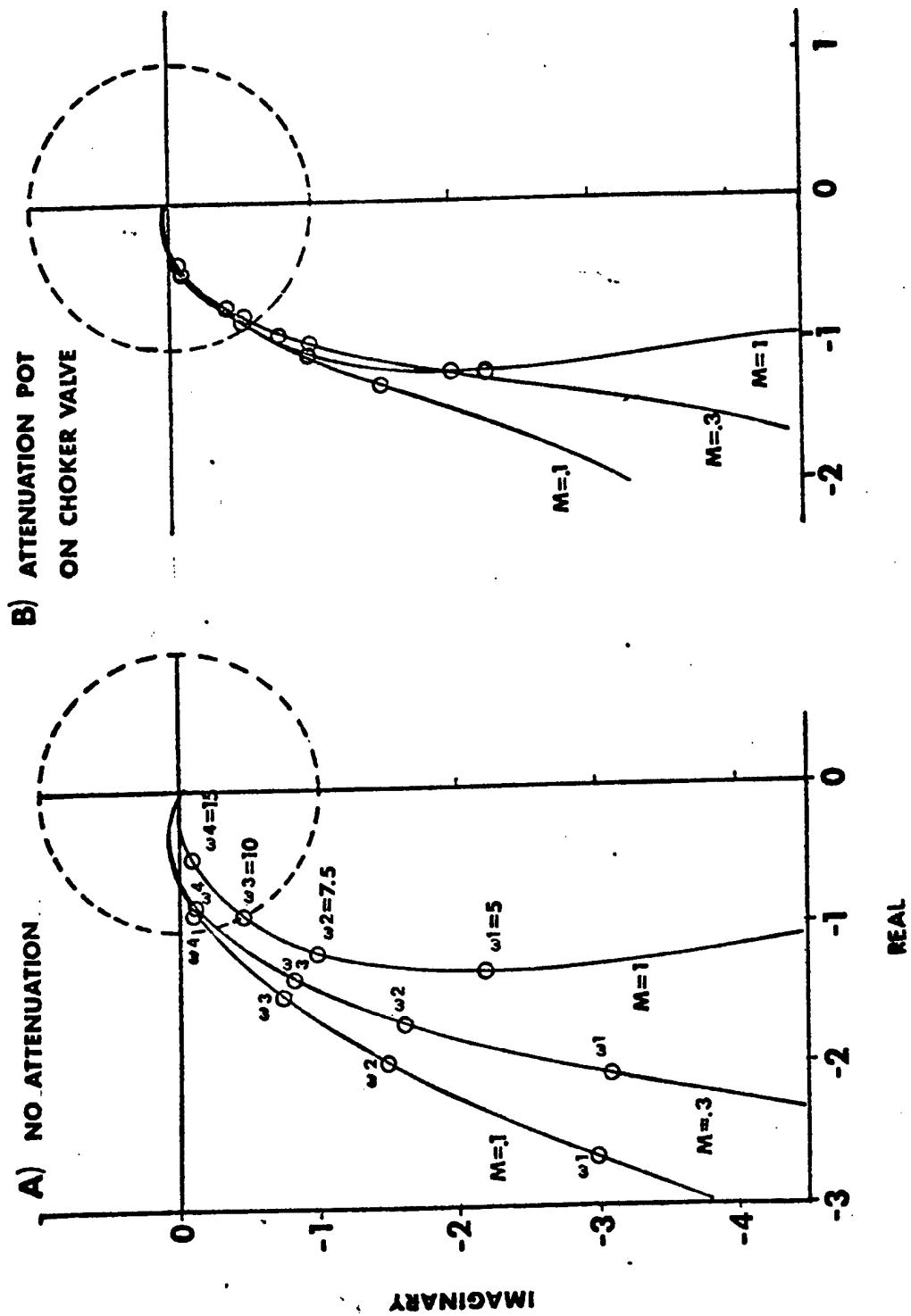


FIGURE 13 - Pressure Control System Frequency Response - Nyquist Plot

across these taps to provide the attenuation shown in Figure 12 for MN .1 and .3. This corrects the gain and phase margins for low Mach number operation as shown in Figure 13B. The gain is then automatically adjusted and no operator adjustment is required.

An open loop Bode plot, corresponding to Figure 13B, is shown in Figure 14. The lag compensator pole to zero frequency ratio was chosen to provide the minimum gain. The pole was selected at a fairly high frequency which would still give the desired effect and would keep the capacitor and resistors reasonably sized.

3. Time Domain Simulation. The system was simulated using an IBM 1130 Continuous Systems Modeling Program which runs on the 9" x 9" Facility Hewlett-Packard mini-computer.

For the simulation, the gain blocks  $K_2$  and  $K_3$  shown in Figure 11 were replaced with a subroutine which simulated the actual  $P_o$  control valve operation. The subroutine considered upstream line losses and calculated flow for either the sonic or subsonic case, depending on the pressure drop available. A block diagram of the subroutine is shown in Figure 15.

A ramp rate of 22.5 psi/sec was used. This is a two second ramp from ambient to maximum pressure of 45 psig. Since simulation runs were started at 15 psig only 1.33 seconds was required for the ramp.

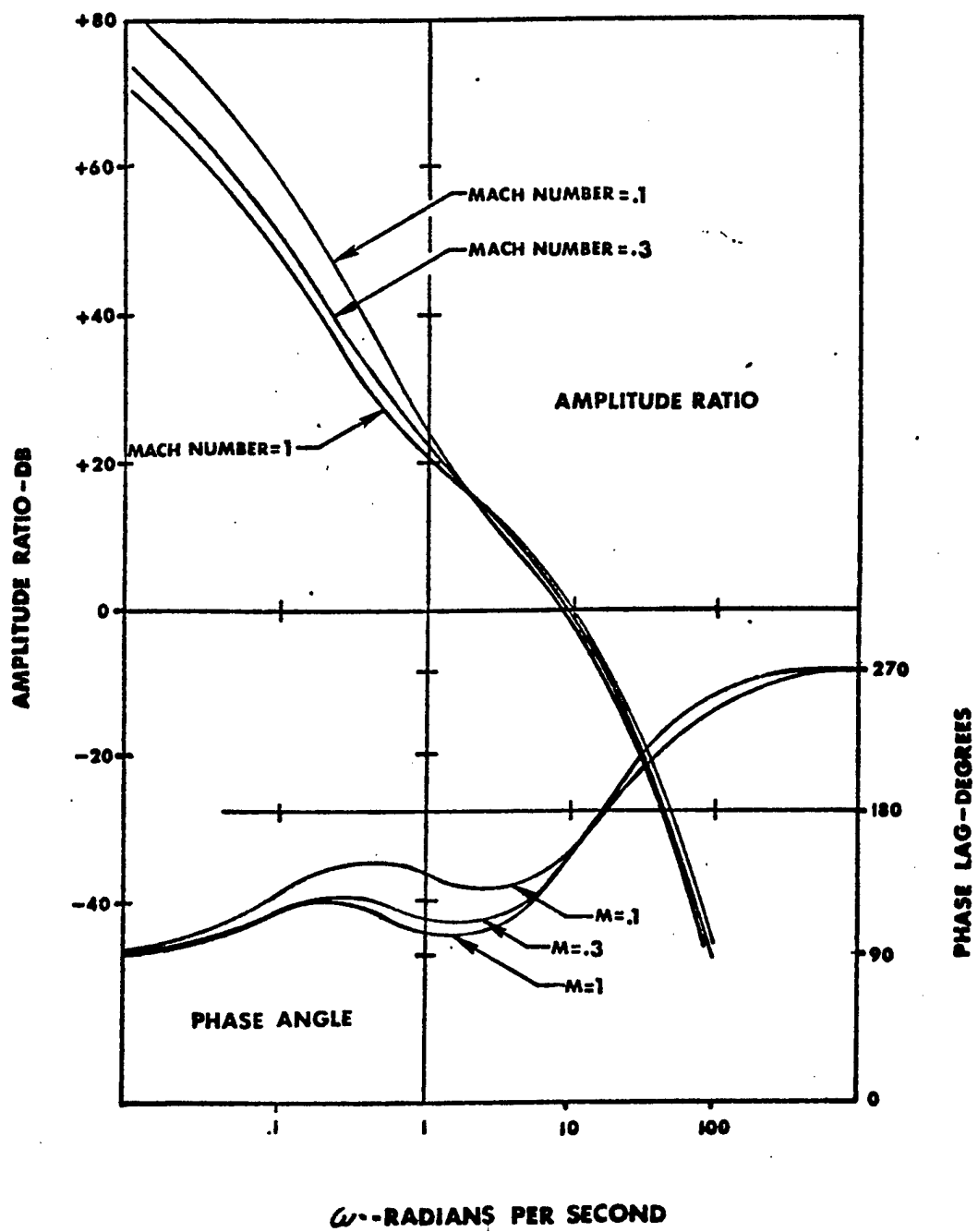
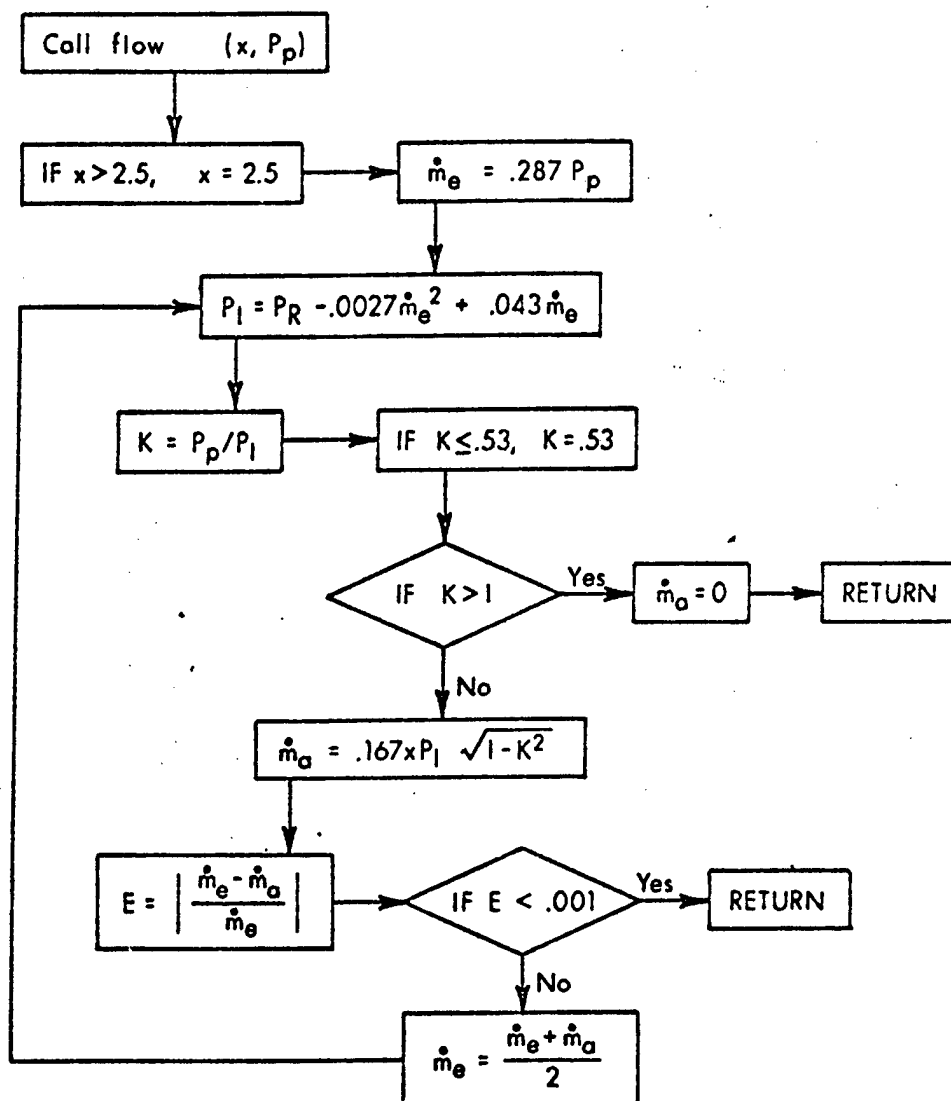


FIGURE 14 - Pressure Control System Frequency Response - Bode Plot



Where:

- $\dot{m}_e$  = Estimated Flow
- $\dot{m}_a$  = Actual Flow
- $P_1$  =  $f(\dot{m}_a)$  due to line loss
- $P_p/P_1$  = Pressure Ratio, .53 = Sonic
- $E$  = Flow estimation error
- $P_R$  = Regulated Pressure = 700 psia

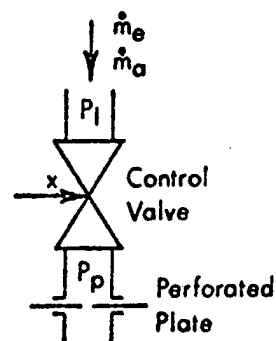


FIGURE 15 - Block Diagram of Control Valve Subroutine

The simulation runs showed that the state variables were never saturated and that the linear model was a good approximation. The maximum valve velocity and position of 7.5 in/sec and 2.5 inches respectively were never reached.

Several runs were made with different input prefilter time constants at Mach 1. The shortest time to 0.5% settling, 2.20 seconds, was obtained with a prefilter time constant of .3 seconds. The additional time required for starting at 0 psig instead of 15 psig is .67 seconds for a total time of less than 3 seconds. Since saturation did not occur, the ramp rate can be increased to 30 psi/sec or more. The total time to stabilized conditions is less than 2.5 seconds and the air lost in the process is less than 70 pounds mass.

The velocity constant,  $K_v$ , recorded was 17.7 which is a little less than the theoretical value for the linear system. Four of the simulation runs for Mach 1 are shown in Figure 16. Values recorded for the runs, including variables subject to saturating, are shown in Table III.

TABLE III  
DATA RECORDED IN  $P_0$  SIMULATION RUNS

ITEM	VALUE			
RUN	A	B	C	D
Pre-filter Time Constant	.01	.25	.3	.4
Maximum Control Valve Speed	-2.27	+1.09	-2.81	+.90
Maximum Valve Position	1.61	1.44	1.40	1.32
Maximum Flow	129.8	122.6	120.9	117.0
Maximum $P_0$	61.79	60.33	60.29	60.23
0.5% Settling Time	3.55	3.10	2.20	2.55

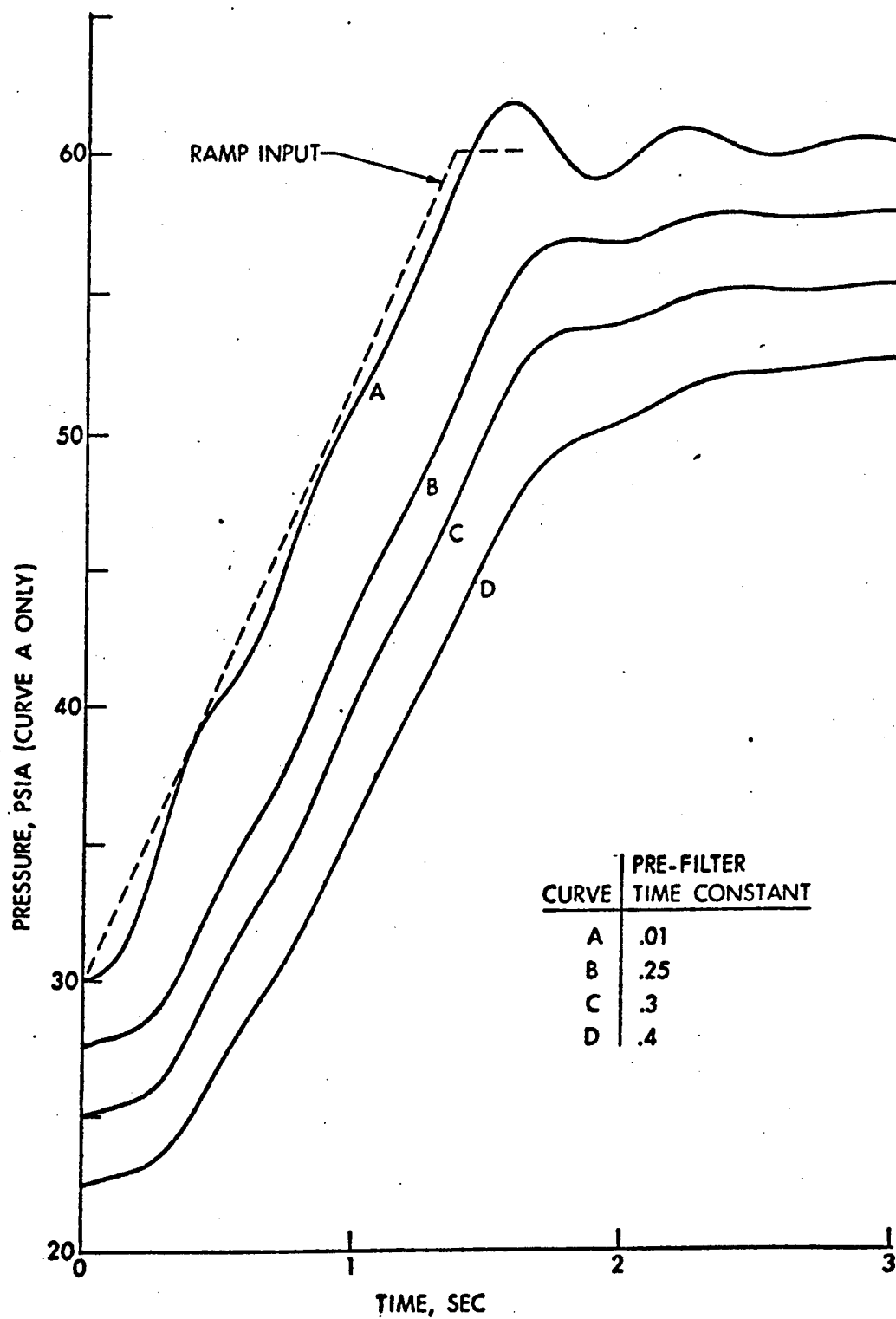


FIGURE 16 -  $P_o$  Simulation Runs-Response to a Ramp to Operating Pressure

## VI. CONCLUSIONS

The existence of a design source document on the wind tunnel should prove valuable in the future for the familiarization of personnel for maintenance, operation and any future modifications of the tunnel. Detailed design information is given which includes, for instance, the trends in airflow control systems stability associated with changing tunnel Mach number. The piping and control systems are shown to be adequate from the standpoints of safety, control accuracy and facility running time.

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