FINAL REPORT

INVESTIGATION OF WATER AND STEAM AS THE WORKING FLUIDS IN A FLUID AMPLIFIER SPEED GOVERNOR FOR TURBINE GENERATOR SETS

Prepared for: Department of the Navy
Office of Naval Research
Power Branch Code 429

Contract No: Nonr-4001 (00) FBM Amendment #2

Prepared by: General Electric Company
Research and Development Center
Schenectady, N. Y.

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1.0 INTRODUCTION

This report gives the results of work done under amendment #2 to Contract No. NONR-4001(00) FBM, Study of the Feasibility of Applying Fluid Controls to Turbine Generator Sets. The contract was supported by the Department of Navy, Office of Naval Research, Power Branch Code 429.

The work done on Amendment #2 was a follow-on to the basic feasibility studies completed in the first phase of the Contract. The study selected the most promising circuit approach for a turbine generator speed control and selected condensate water and steam as potential candidates for the working fluid.

The two major tasks completed on Amendment #2 were: The mechanization of the recommended speed control loop using water as working fluid and an evaluation of the potential of steam as a working fluid.

2.0 SUMMARY AND CONCLUSIONS

The following specific tasks were accomplished on Amendment #2 to the contract:

1. A turbine-generator speed governor using fluid amplifiers with water as the operating fluid was mechanized and evaluated. The control loop consists of a reed governor for speed sensing and discrimination, a closed loop steam valve actuator stage and a fluid amplifier reset-plus-proportional gain circuit. The control loop was demonstrated and tested using a scale air turbine and generator.

2. Several basic concepts for a no-moving-parts integrator for the reset circuit were conceived, built and evaluated. The concepts were based on a time constant multiplication circuit using a fluid operational amplifier. The concepts were shown to be fundamentally sound, however, the specific requirements for a speed loop controlling a turbine in the 5 - 15 sec. time constant range could not be met due to signal/noise limitation in the fluid circuits.

The reset circuit for the loop which was mechanized uses an entrapped air volume to give an integration time constant. Functionally, this approach works satisfactorily, it does add a water-to-air interface which is made through a flexible diaphragm.

3. A no-moving-parts derivative circuit was conceived, built and evaluated. The derivative circuit functions satisfactorily and is a required functional block in control loops where an open loop actuator stage is used for integration.

4. A closed loop actuator stage utilizing position feedback was built and tested. Sufficient loop gain was incorporated to handle the severe steam valve reaction forces encountered on the multiple poppet type steam valves in common usage.

5. A materials and design study on fluid amplifiers for steam operation was completed. Thirteen fluid amplifiers of several different configurations and materials were fabricated and checked out for the steam life test.

6. Wet and dry steam life test fixtures were designed and constructed. The wet steam fixture supplies constant quality steam to the test elements with steam inlet conditions ranging from saturated to $300^{o}F$ superheated steam at 380 - 420 psig.

7. A life test with 2080 hours of operating time was completed on 13 fluid amplifiers. Three amplifiers were operated with dry steam and ten on wet steam of 6% quality. Results of the test when used in conjunction with the extensive erosion data and "know-how" accumulated by the General Electric Large Steam Turbine Dept. can be used to predict a useful life of the fluid amplifier as a function of the materials used and the steam conditions.

8. An analytical study on the dynamics of a steam actuator for the turbine steam valve was completed. Results of this study were conceptual circuit designs for a fluid amplifier actuator drive stage.

Conclusions

1. The steady state and dynamic requirements of MIL-G-21410 can be met with a water operated loop consisting of the reed governor, a

closed loop actuator stage and a fluid amplifier reset stage. Bandwidths adequate for crossing the overall speed loop as high as 20 rad/sec are possible. Steam valve reaction forces of typical poppet type steam valves can be readily handled by the closed loop actuator approach.

2. The basic loop configuration and component designs mechanized for water are directly applicable to other incompressible fluids such as lubrication or hydraulic oils. Individual amplifiers and the reeds for the reed discriminator would have to be redesigned to accommodate the high temperature-dependent viscosity of the oils.

3. Mechanization of a "no-moving-parts" reset loop based on time constant multiplication is beyond the present "state-of-the-art". Self generated noise in the fluid amplifiers is the primary limitation, improvements on the order of 10:1 are required. Subsequent Company sponsored work indicates that such improvement is realizable.

A reset circuit based on using a water to pneumatic interface is much more tolerant of noise, has the required flexibility and range. It is the recommended short range approach where incompressibl fluids are used.

4. Amplifiers using an incompressible fluid are susceptible to shock and vibration induced noise. Acceleration of fluids in lines and interconnecting passages between amplifiers results in a dynamic head across the input to the amplifier. This is particularly critical in the low level stages where saturation may result. Therefore, the use of a push-pull system is mandatory;

dual lines carrying signal information must be routed along a common path and the projected line length along the control port axis must be kept to an absolute minimum. Parallel input stages, interconnected so that true error signals add and vibration induced signals cancel, may be required.

The susceptibility of air or steam operated amplifiers to induced vibration is reduced by several orders of magnitude as compared to water. The use of air or steam is attractive for the low level stages of amplification.

5. Results of the steam life test show that erosion of fluid amplifiers operating on steam is not a serious problem. Useful life of 5 - 10 years can be predicted for amplifiers fabricated of chrome steel and operated under adverse steam conditions. The overall performance of both the chrome steel and carbon steel amplifiers improved as a result of the life test. This indicates that factors, other than gross changes in geometry due to erosion, have a strong influence on performance. The conclusions that can be inferred from the life test results were suggested and substantiated on a current General Electric funded program on basic noise reduction in fluid amplifiers and are considered proprietary. The conclusions and recommendations have been sent to the contracting officer under separate cover.

6. Depositions of carry-over from the boiler feed water and the steam lines can be a significant factor not only in the fluid amplifiers but also in power supply orifices and other circuit components. Definite deposition patterns were detected on the amplifiers subjected

to dry steam. No adverse effects on amplifier performance were attributable to deposition, the amounts accumulated were small and the amplifiers would be expected to operate satisfactory for thousands of hours under the specific test conditions. However, the rate of deposition has more uncontrollable factors than rate of erosion. It cannot be as readily predicted and may vary widely between installations.

7. The analytical study of the dynamics of a fluid amplifier steam operated actuator for a turbine steam valve produced conceptual designs for both a balanced low-reaction steam valve and for the more commonly used high reaction poppet valve. From the standpoint of dynamics an actuator stage for either valve can be mechanized with a total of five fluid amplifier stages. The actuator area required is large (20 in^2) compared to a water operated actuator (8 in^2).

The practical problems associated with building a "non binding" actuator were beyond the intended scope of the study. However, past experience on steam, particularly dry steam actuators has been unfavorable. For this reason a steam operated actuator is not recommended for a general purpose speed control loop. A steam actuator could be considered for specific wet steam applications.

8. A hybrid system using two working fluids is attractive for future applications. This system can capitalize on the inherent advantages of either fluid without the attendant disadvantages. A compressible fluid such as steam would be used for the speed sensor, discriminator and for the reset plus proportional gain function. This would eliminate the shock and vibration problem associated with the incompressible fluids. No-moving-parts reset circuits can be implemented with

simple fluid circuits, well within the present "state of the art". The ease of speed signal generation and transmission allows more flexibility of installation of the speed sensor on the prime mover and the basic accuracy of the speed reference can be enhanced by the use of higher Q reeds.

The fluid amplifier closed loop actuator stage would use an incompressible fluid such as oil or water. The implementation of high performance actuator loops capable of handling high reaction steam valves is relatively straight forward. The fluid amplifier power levels are sufficiently high at the actuator to be relatively immune to vibration and shock and also the problems associated with high viscosity fluids would be minimized at this level.

3.0 MECHANIZATION OF SPEED CONTROL WITH WATER

This phase of the contract involved a system analysis to select the basic system configuration, derivation of design goals for the functional blocks, design and fabrication of the required hardware, test and evaluation of components and assemblies required to derive the required functions and the test and evaluation of the overall speed control loop using a scale air turbine and generator to simulate the full scale prime mover and load.

3.1 System Analysis and Design Goals

System design goals were established using the requirements of MIL-G-21410 applied to a DL-5 type turbine-generator and steam valve combination. The DL-5 is a 600 kw set using a relatively high reaction multiple poppet steam valve. The turbine-generator time constant is approximately 12 seconds.

The pertinent requirements of MIL-G-21410 are:

1. Maximum steady state speed regulation of $\pm$ 0.2%.
2. Maximum speed deviation of 2% on 100% jump load change with recovery to 0.5% in less than 1.5 seconds.

Consideration of these two requirements dictate an isochronous control. The minimum gain of the reset loop is established by the steady state requirement. The transient requirements establish the reset time constant and the minimum acceptable system bandwidth.

Block diagrams of two basic system approaches which will meet these requirements are shown in Figures 3.1.1 and 3.1.2. The two approaches differ primarily in the method of obtaining the integration required for isochronous control.

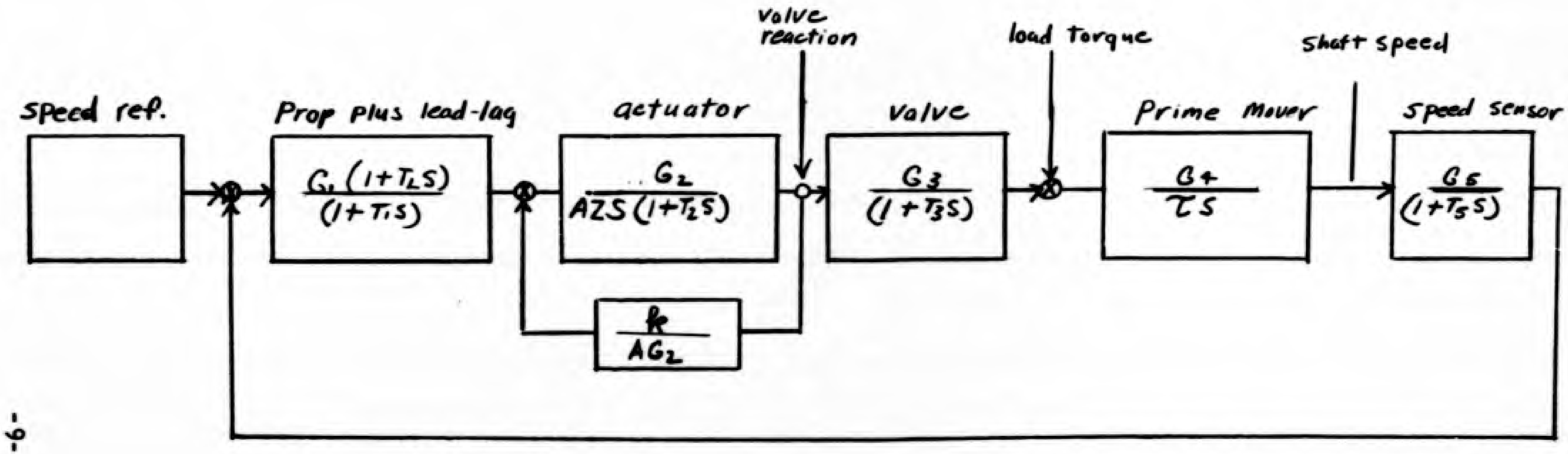


Figure 3. 1. 1 Block Diagram Opcn Loop Actuator System

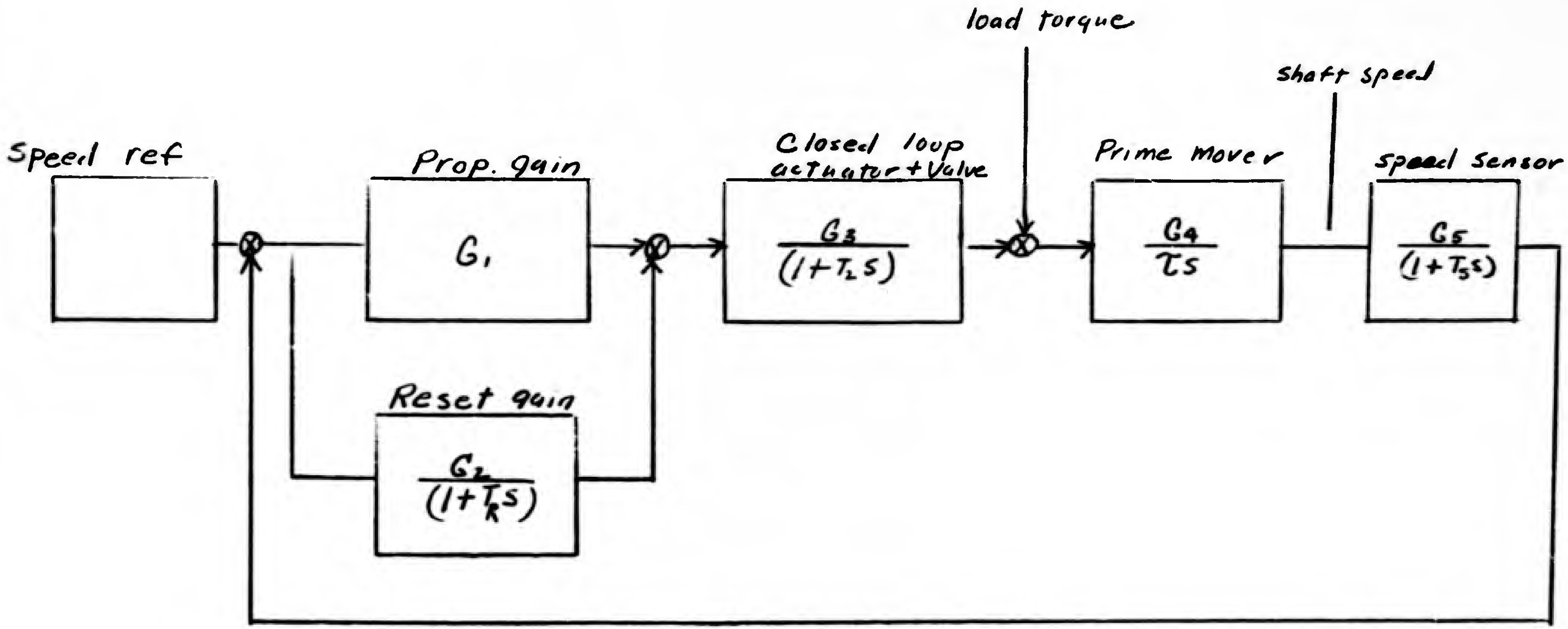


Figure 3. 1. 2 Block Diagram Closed Loop Actuator System

The loop of Figure 3. 1. 1 integrates in the actuator for the main steam valve. The actuator stage is open loop, i. e. the actuator assumes a velocity in response to a speed error signal. This loop will give negligible steady state droop if steam valve reactions and other extraneous forces on the actuator can be neglected. The overall loop requires a derivative circuit for dynamic stability.

The system shown in Figure 3. 1. 2 utilizes a closed loop actuator stage. The actuator assumes a position proportional to speed error. The integration required for isochronous control is obtained with a low bandwidth, high gain fluid amplifier in parallel with a low gain proportional amplifier. The overall performance of this system can be made independent of the actuator characteristics.

Figures 3. 1. 3 and 3. 1. 4 show a straight line approximation of the attenuation vs. frequency plots for the two systems. By choice of parameters the transient and steady state errors resulting from jump load changes or demand speed changes can be made identical. Based on these criteria the loop of Figure 3. 1. 1 would be the logical choice. It can be implemented with one-half the fluid amplifiers and eliminates the fluid summing junctions required for the closed loop actuator system shown in Figure 3. 1. 2. Referring to the detailed analysis presented in Appendix A, it is apparent that this loop is limited to turbine generator sets utilizing a balanced low reaction steam valve. This is a severe restriction for a steam turbine generator control system. The configuration selected for the feasibility model is the more complex control system of Figure 3. 1. 2. This system can handle the commonly used high reaction poppet steam valves as well as the low reaction type. The mechanization of the loop is based on the frequency modulation techniques recommended as a result of the basic feasibility studies completed on Phase I of the

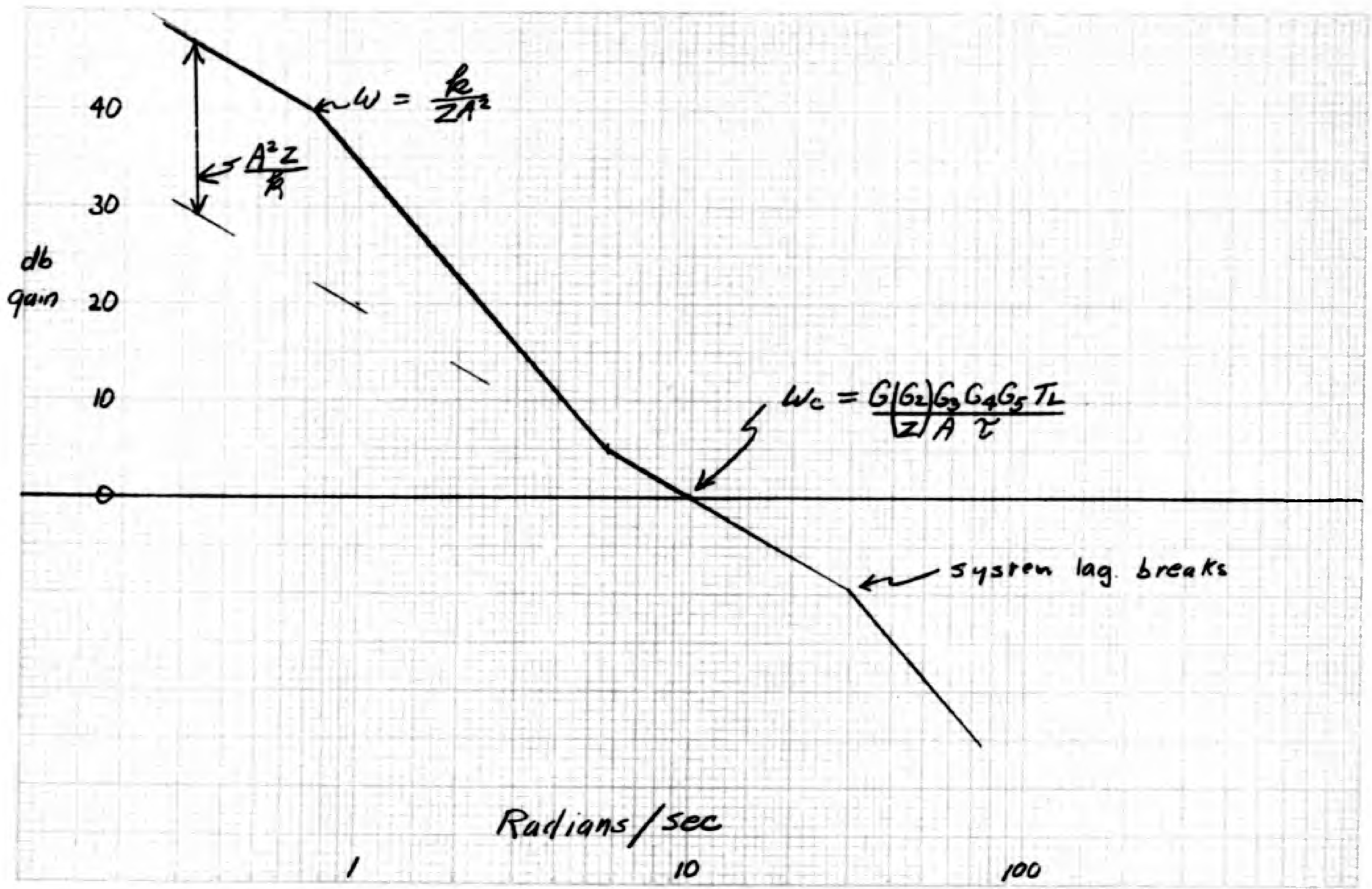


Figure 3. 1. 3 Attenuation Plot for Open Loop Actuator System

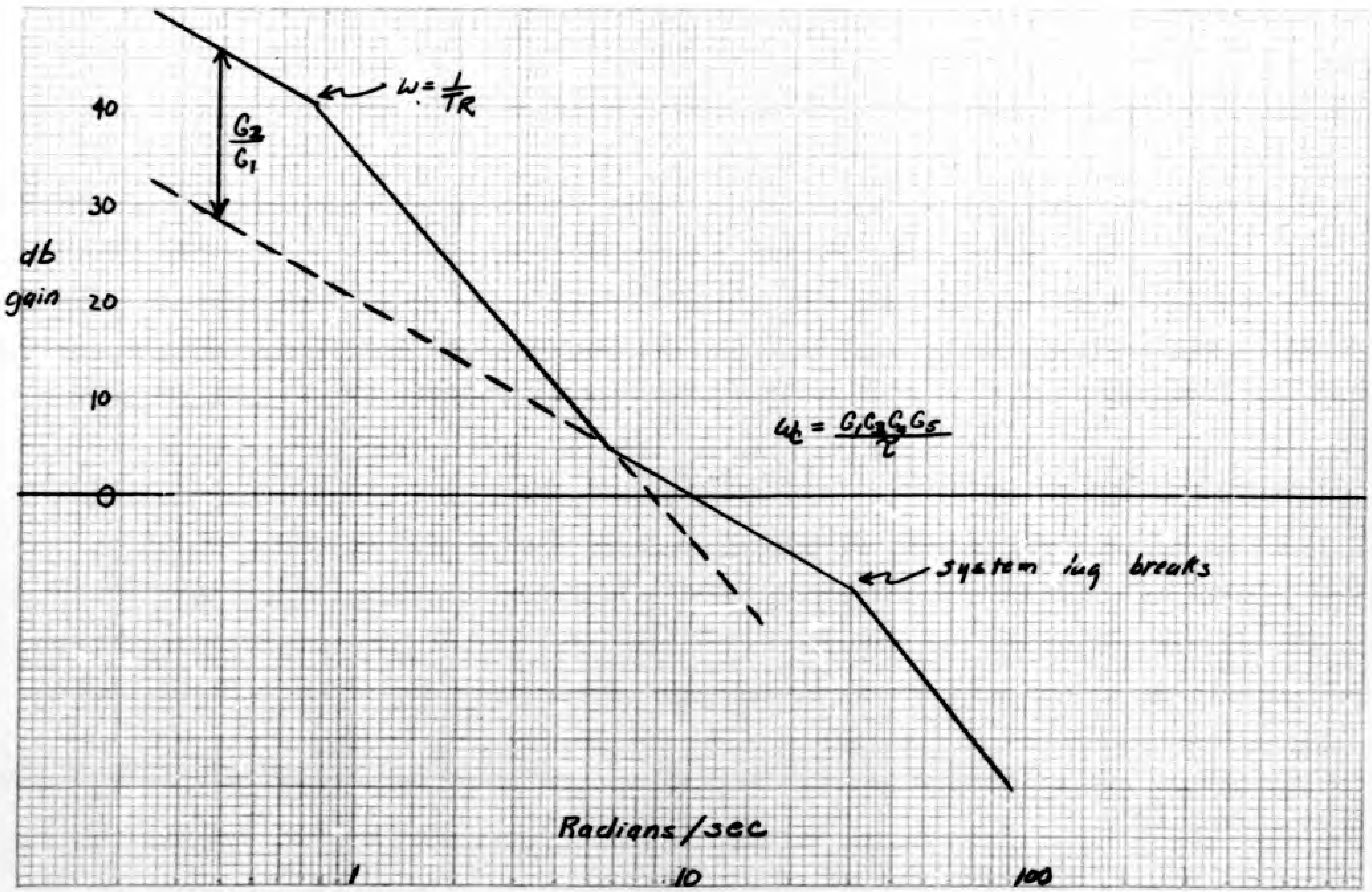


Figure 3. 1. 4 Attenuation Plot for Closed Loop Actuator System

current contract. The complete system consists of a sinusoidal chopper located on the prime mover shaft, a reed discriminator which compares actual speed with the desired set speed and derives an analog speed error signal, a proportional plus reset fluid amplifier gain block and a closed loop actuator stage.

Appendix A discusses the derivation of the design goals for the overall system and the functional blocks comprising the system. The following goals were established.

1. Overall system crossover at 10 rad/sec with a minimum open loop phase margin of 25°.
2. Reed discriminator and speed sensor - contribute less than 5° phase lag at 10 rad/sec.
3. Proportional gain path - less than 2° phase shift at 10 rad/sec.
4. Ratio of reset to proportional gain of at least 10:1 with a minimum reset time constant of 1 sec.
5. Closed loop actuator - less than 8° phase shift at 10 rad/sec.

3.2 Speed Sensor

Two possible configurations for the sinusoidal chopper are shown in Figure 3.2.1. The principle of operation of the two configurations is identical, the major differences are in the ease of manufacturing and tolerance to end play in the turbine shaft.

In the A configuration the impedance of the two downstream nozzles is varied as a function of shaft rotation by varying the clearance between the end of the nozzle and a lobe located on the circumference of the turbine shaft. This configuration is insensitive to end play on the turbine shaft and is the most likely candidate for installation on the full scale turbine generator set.

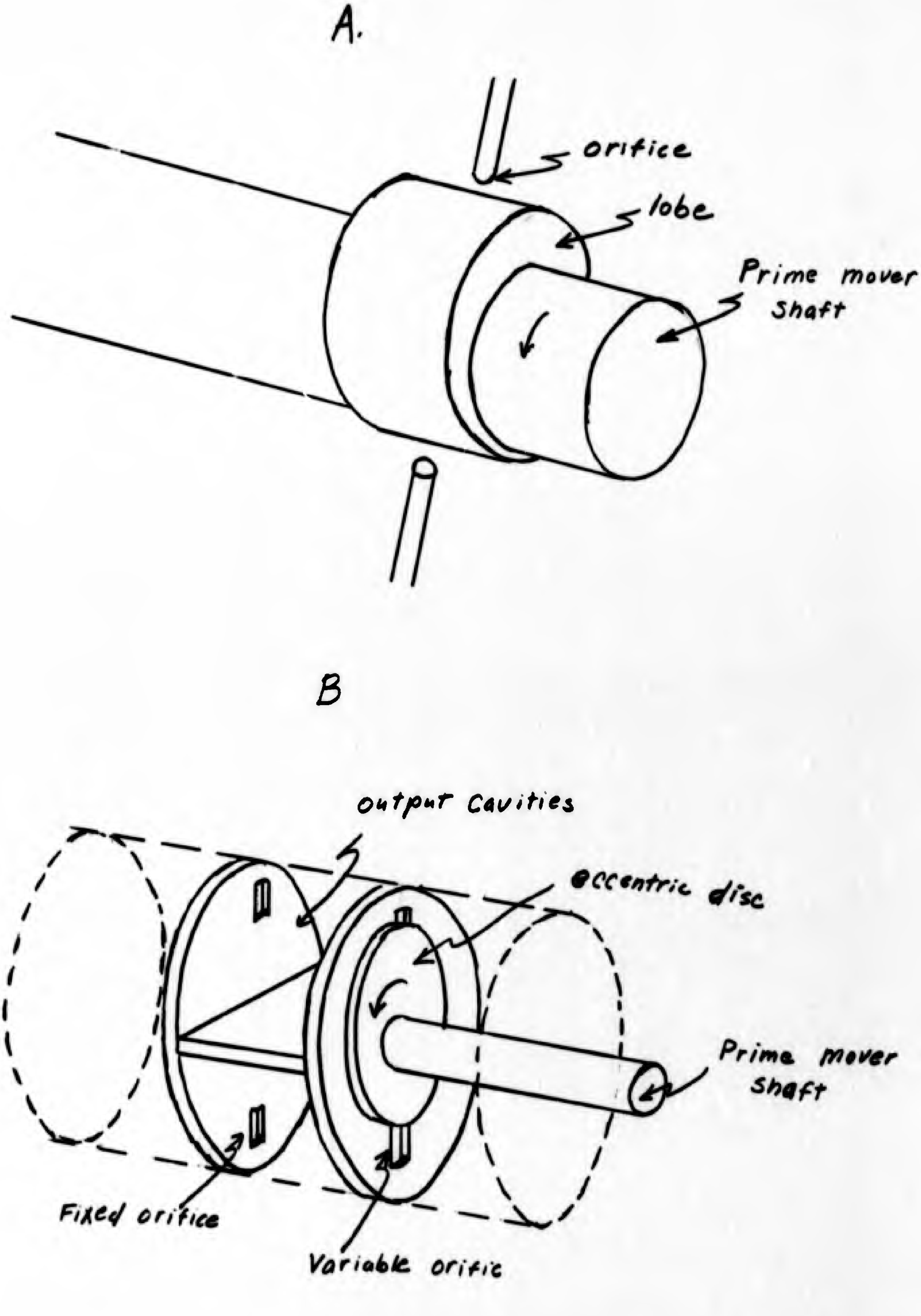


Figure 3. 2. 1 Sinusoidal Choppers for Speed Sensing

Configuration B varies the area of two rectangular orifices by rotating an eccentric disc in front of the orifice exit. The primary advantage of this configuration is the ease of generating good sine waves without requiring close dimensional tolerances. The maximum pressure modulation is a function of end play in the prime mover shaft. The variation in amplitude with time and operating conditions can be minimized at the expense of gain by designing for large initial clearances.

Both configurations can be used to generate frequencies which are some multiple of shaft speed. A two lobe cam with the orifices or nozzles located on axis 90° apart will generate a push-pull second multiple of shaft speed. This feature permits greater flexibility in the design of the reed discriminator.

Configuration B was selected for mechanization of the feasibility control loop because of its greater flexibility and ease of construction. The partially disassembled speed sensor is shown in Figure 3.2.2. The nominal area of both upstream and downstream orifices is variable so that the impedance of the sensor can be matched to the input impedance of the reed discriminator. The push pull output is taken from cavities formed by two plates containing the rectangular orifices and the outer case. The dimensions of the cavities were selected to minimize inductance in the signal path.

The measured frequency response of the sensor is shown in Figure 3.2.3. The frequency response is relatively flat over the frequency range checked. The response is sufficiently high to satisfy any anticipated application.

Figure 3.2.4 is a measured plot of the differential output pressure vs. shaft rotation. The output is a good sine wave; this is a prerequisite when

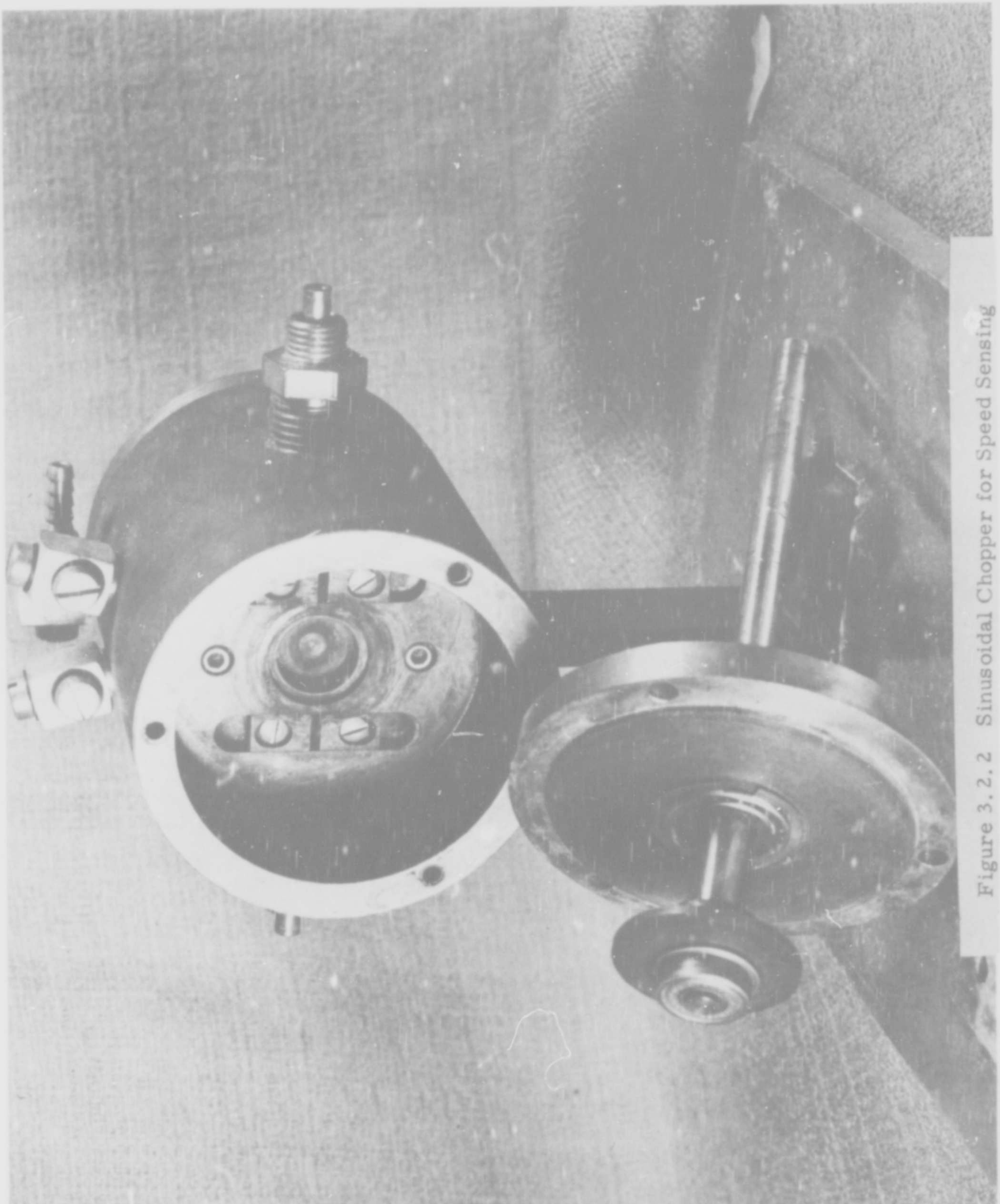


Figure 3.2.2 Sinusoidal Chopper for Speed Sensing

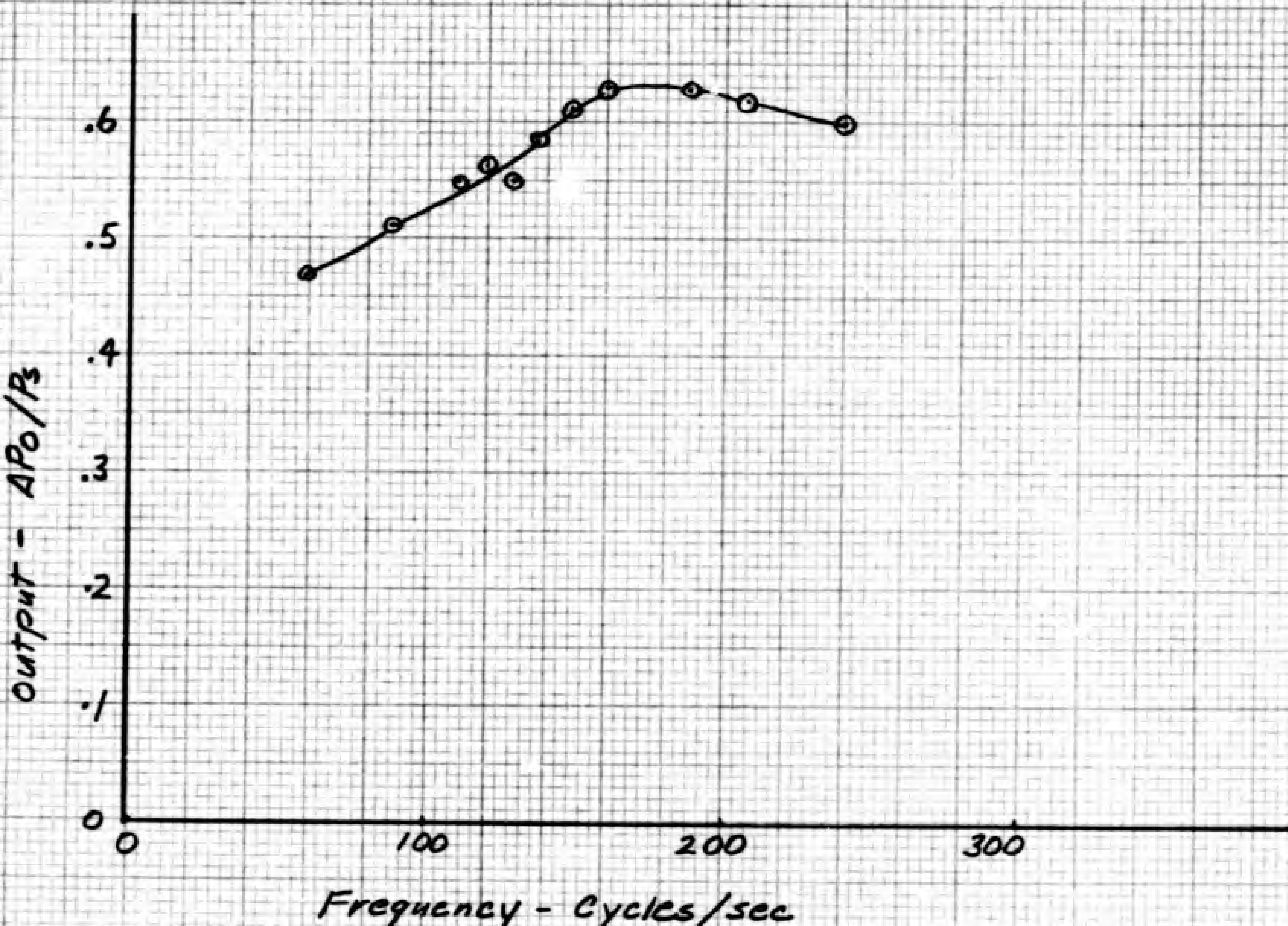


Figure 3.3.3 Sinusoidal Chopper Frequency Response

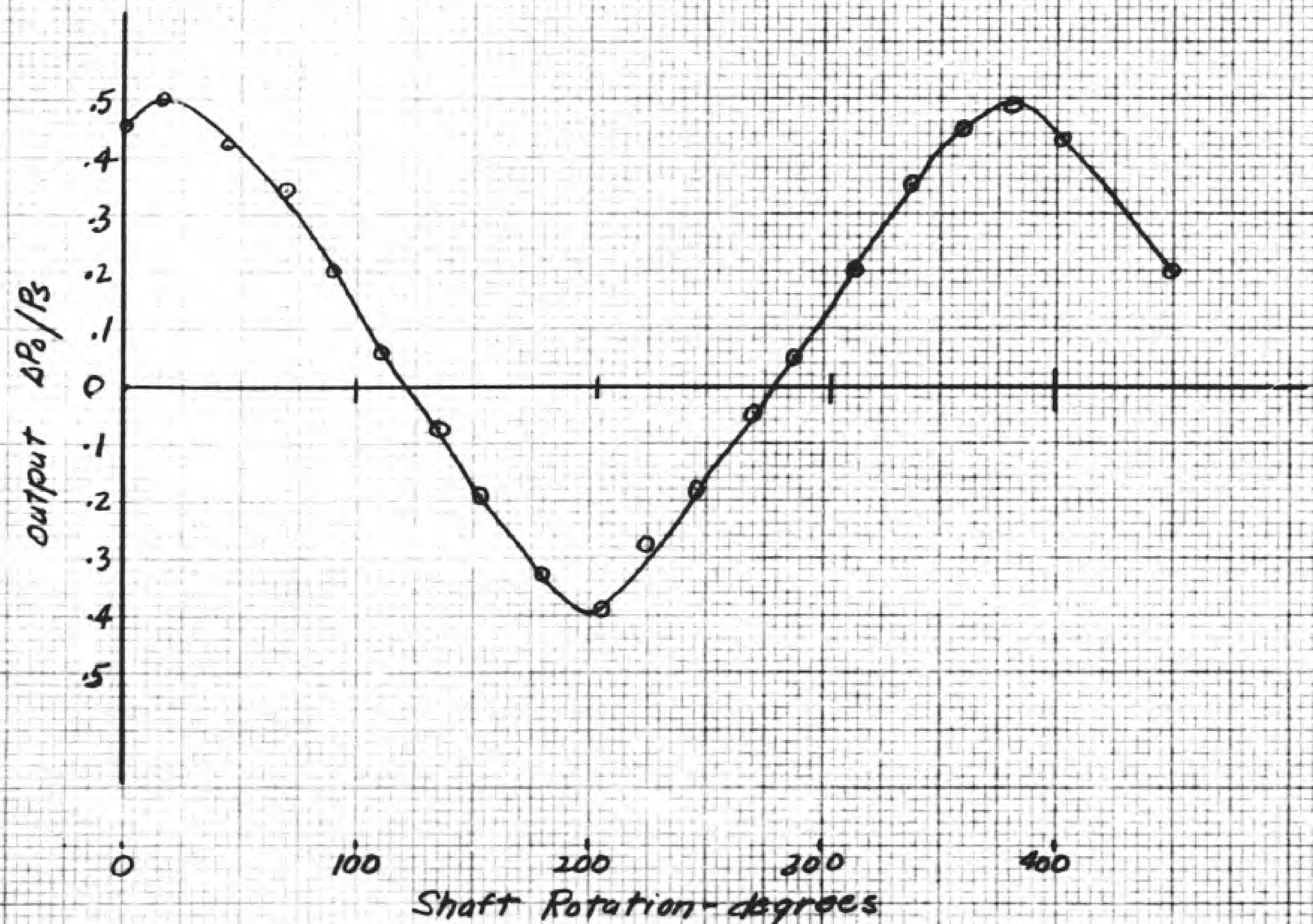


Figure 3. 2. 4 Sinusoidal Chopp.r – Steady State Characteristics

used in conjunction with a reed discriminator. The presence of higher harmonics in the output can excite the reeds of the discriminator and cause a "lock-in" at a speed which is a sub-multiple of the desired set speed. The maximum output is 50% of the power supply. This is sufficiently high to use the speed sensor output for direct excitation of the reed discriminator in the feasibility model. For full scale applications where the speed sensor must be remotely located from the reed discriminator, it may be desirable to operate the speed sensor at a relatively low level and use the output to excite a driver amplifier located directly at the reeds. The efficiency of the sensor and discriminator combination will be improved, the gains and bandwidths will be unaffected.

3.3 Reed Discriminator

From the standpoint of the overall system the most critical characteristic of the reed discriminator is the bandwidth. A phase shift of 5° lag at 10 rad/sec was allocated to the discriminator. This corresponds to a bandwidth of 115 rad/sec.

The bandwidth of a reed for a signal which originates due to a shaft modulation frequency is a function of the reed quality factor (Q) and the operating point (k) on the reed characteristic output vs. excitation curve. Q is defined as the ratio of the peak energy stored in the reed to the average power dissipated per radian. The operating point (k) is conveniently expressed in terms of Q, the reed resonant frequency (W_R) and the nominal operating frequency (W_c).

$$k = \frac{Q\,(\pm W_c \mp W_R)}{W_R}$$

Past experience on reeds and tuned circuits indicates that an operating point of $k = 1$ is about optimum. Higher values of k result in larger bandwidths at the expense of gain. Values less than 1 result in higher gains (maximum occurs at k = 0.4) however, the bandwidth drops off quite rapidly.

The bandwidth for a reed or any resonant type circuit operating at k = 1 is:

$$BW = \frac{1.3\ W_R}{2\ Q}$$

The maximum Q that can be used is then determined by the system bandwidth requirements. Considering a 12,000 rpm machine with a speed sensor giving a one-to-one conversion between excitation frequency and shaft speed the max. Q for this application becomes:

$$Q_{max} = \frac{1.3\ W_R}{2\ (BW)} = 7.1 \text{ rad/sec}$$

The design of a reed to give a certain damping factor is largely empirical. For reeds immersed in water the reed damping will be a function of velocity squared if the Reynolds number at the reed tip is larger than 1000 and a direct function of velocity if less than 10. The reeds for this application will operate in the range of 100 - 1000, hence reed damping will be non-linear. The Q and bandwidth will be a function of the amplitude of oscillation or driving force. The reed must be designed so that the required Q is compatable with the amplitude of oscillation required for the reed pick offs.

Test results on a preliminary reed design are shown in Figure 3.3.1. The Q decreases with increasing amplitudes at a rate which indicates that the reed damping is almost a function of velocity squared. This type of data was extrapolated to yield the final reed design shown in Figure 3.3.2.

The conversion of reed amplitude of oscillation to a pressure is accomplished by a small through hole at the tip of the reed. A supply nozzle

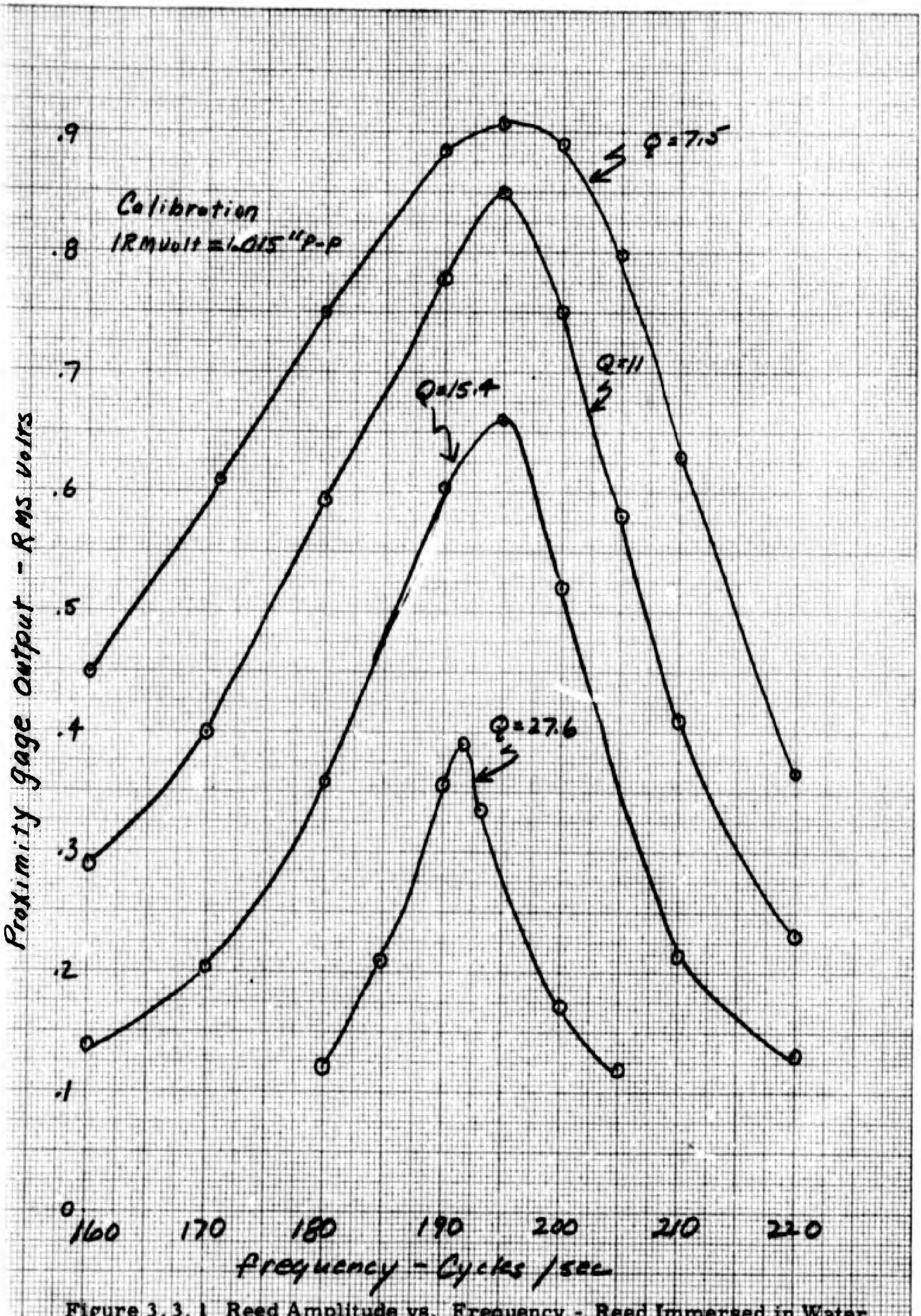
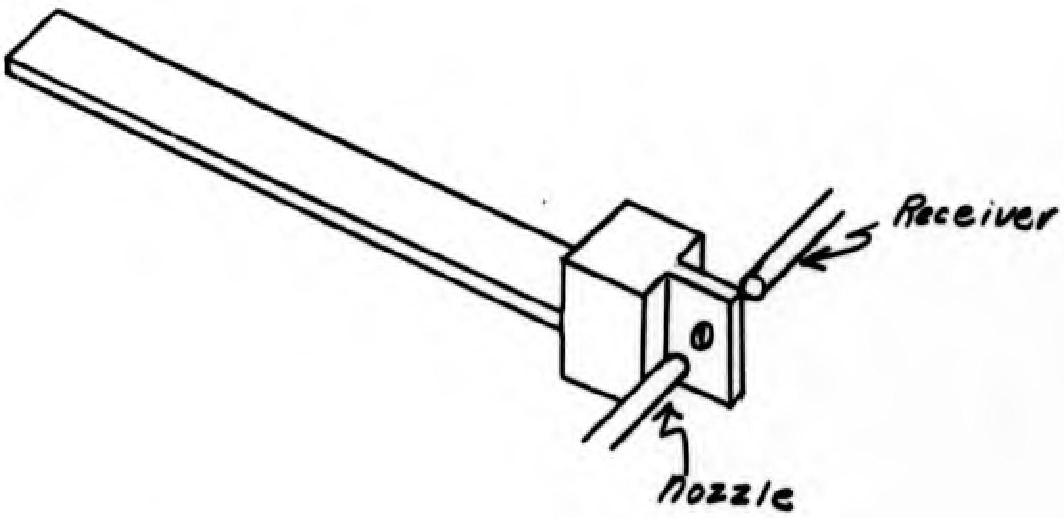


Figure 3.3.1 Reed Amplitude vs. Frequency - Reed Immersed in Water

160 x 220 29/64 In. Div.



Reed dimensions:
1.5" x .125" x .03"
.16" x .16" x .16" mass

Figure 3.2.2 Discriminator Reed Design.

and receiver are aligned symmetrically with the axis of the hole. Maximum pressure recovery in the receiver is obtained when the reed is not vibrating. As the frequency of the reed driving force approaches reed resonance the reed amplitude increases, causing interception of the flow from the supply nozzle to the receiver. The receiver recovery pressure is a minimum at resonance.

Test results on the reed discriminator excited directly from the speed sensor is shown in Figure 3.3.3. The blocked load gain is 1 psi for a 1% change in speed. This drops to 0.5 psi/% when the output is loaded with the input impedance of a fluid amplifier.

Figure 3.3.4 shows the reed discriminator assembly. This model was designed primarily for flexibility. The desired set speed is obtained by adjusting the length of reed clamped in the base fixture. With the length established, the nozzle and receiver are aligned with the hole at the end of the reed. This is most conveniently done by energizing the nozzle and maximizing the pressure recovery in the receiver. The location of the reed driving nozzles is not critical and requires no adjustment. Experience with the design has shown that initial adjustments and alignment is relatively simple. Once aligned there should be no need for further adjustment.

3.4 Actuator Drive Stage

The critical characteristics of the actuator loop are the bandwidth and the loop gain required to cope with steam valve reaction forces.

The phase lag allocated to the actuator stage is 8° at 10 rad/sec. This corresponds to a bandwidth of 70 rad/sec.

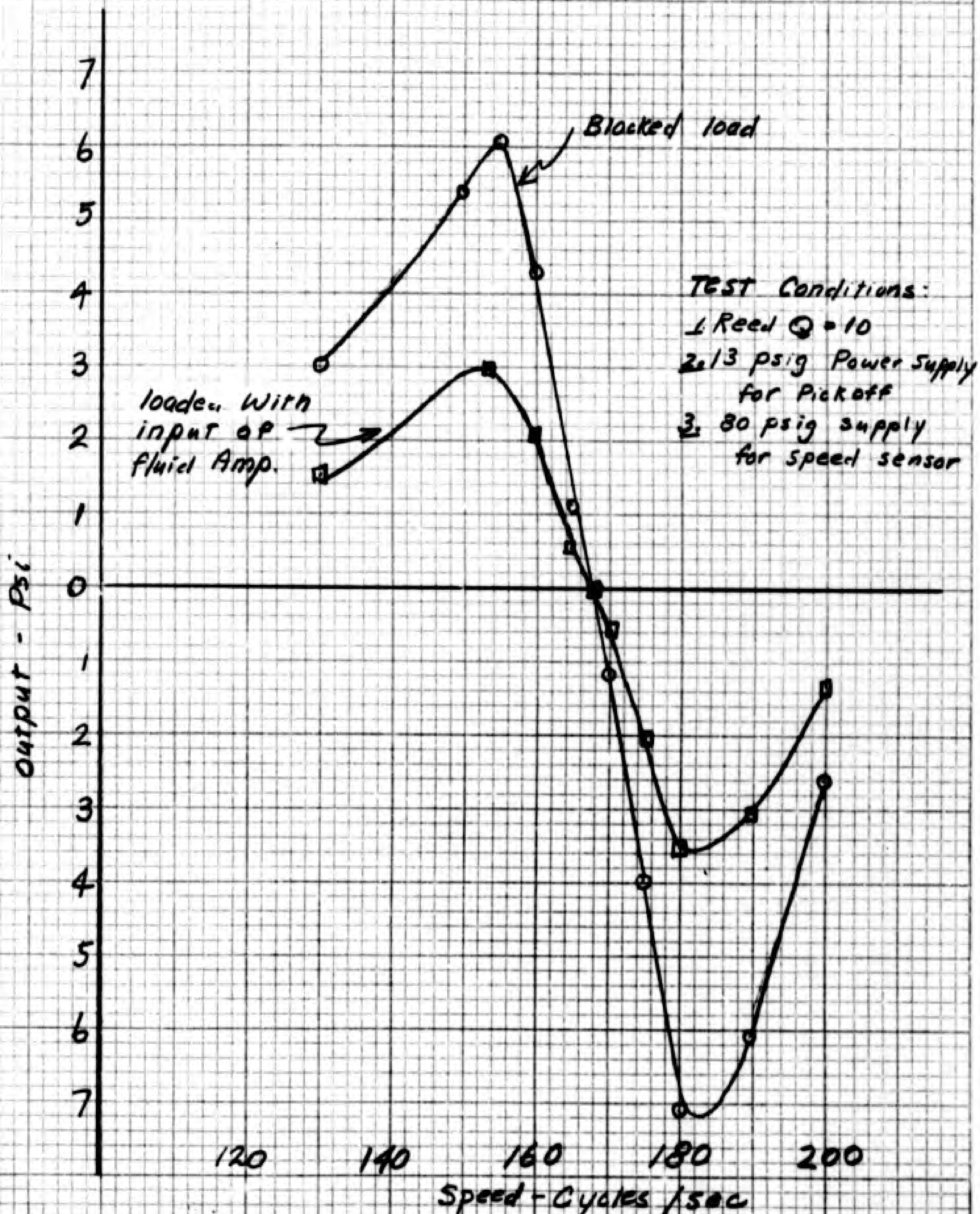


Figure 3.3.3 Output Reed Discriminator Vs. Speed.

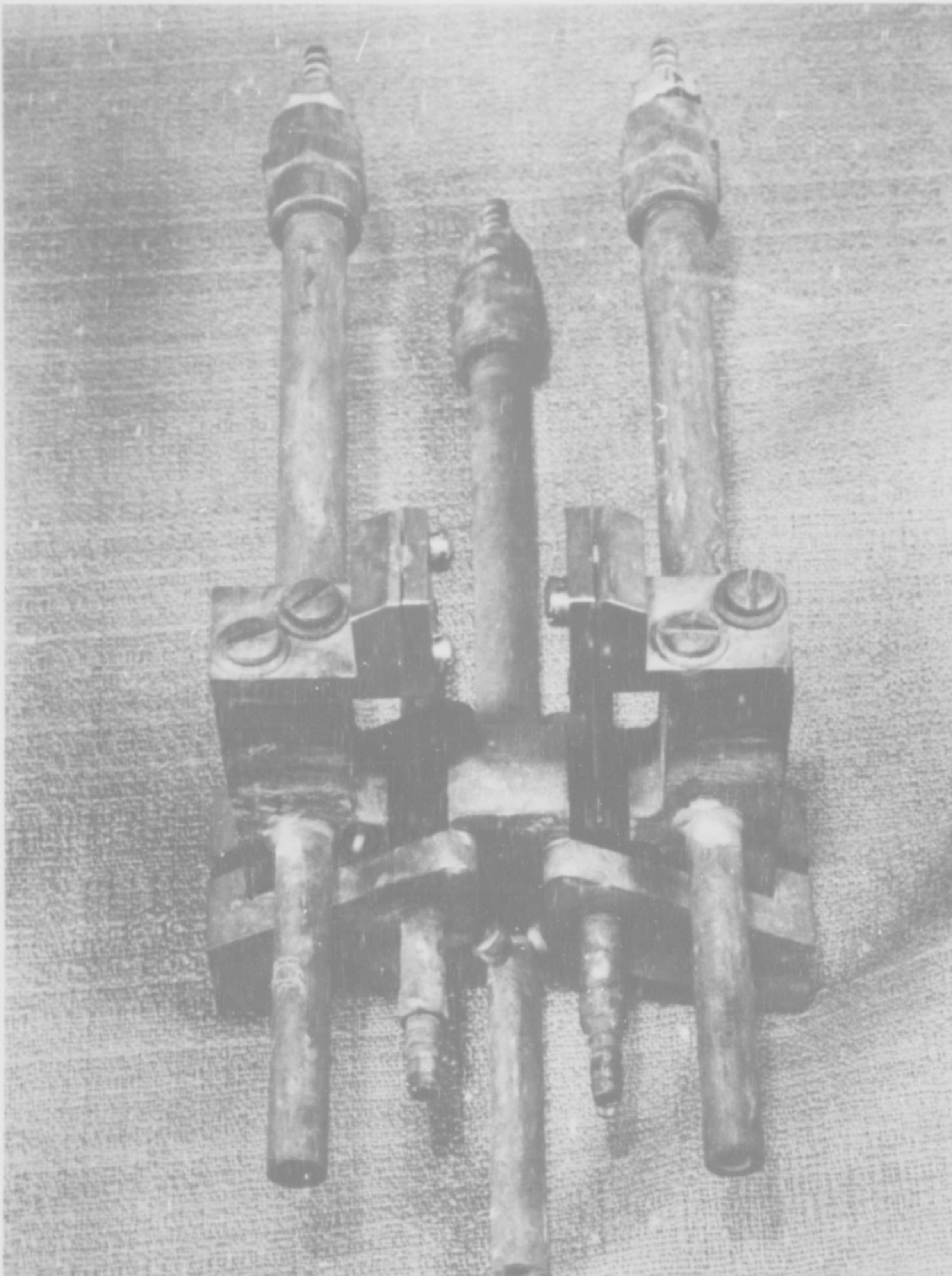


Figure 3.3.4 Reed Discriminator

The loop gain requirements (see Appendix A) are that:

$$G_2 KA \geq 4 k$$

G_2 K is the open loop gain of the actuator, A is the area of the actuator and k is the steam reaction gradient acting on the actuator. Typically k is on the order of 1000 lb/in, hence

$$G_2 KA = 4000 \text{ lb/in}$$

Selection of the actuator area is somewhat arbitrary. The maximum force capability is based on the steam valve cracking forces and the valve spring gradient and preload. A force capability of 1000 lb. is adequate for the steam valve used on the DL-5 class of turbine generator set.

A 200 psig water supply is assumed for the output amplifier. The maximum pressure recovery across a blocked actuator will be 60% of the available supply, or 120 psi. This yields an actuator area of 8.3 in^2. The required fluid amplifier and feedback gain then becomes:

$$G_2 K = \frac{4 k}{A} = 480 \text{ lb/in}^2\text{/in.}$$

The open loop position gain is:

$$G_p = \frac{G_2 K}{Z_o AS}$$

and the crossover frequency becomes:

$$W_c = \frac{3 G_2 K}{4 Z_o A}$$

Z_o is the output impedance of the amplifier driving the actuator.

All the parameters have been established with the exception of Z_o. The maximum Z_c is set by the 2% maximum speed deviation permitted for

100% jump loads. This requirement sets a velocity capability of at least 2 in/sec on the actuator for a 10 sec. turbine (see Appendix A).

The required Z_o can now be determined by:

$$Z_o = \frac{\Delta P}{A V}$$

Where ΔP is the pressure developed across the actuator at the velocity (V) and A is the actuator area.

From measurements taken on a fluid amplifier driving a double ended actuator it was determined that the maximum pressure across the actuator is 75% of the maximum load pressure recovery of the fluid amplifier. Z_o can now be determined as 4.85 lb. sec/in^5.

The minimum actuator cross-over frequency required for steam valve reaction forces is then:

$$W_c = \frac{3 G_2 K}{4 Z_o A} = 9 \text{ rad/sec.}$$

This is a considerably lower bandwidth than the 70 rad/sec specified as a design goal. Overall system damping characteristics rather than steam valve reaction forces will then set the minimum bandwidth, and the open loop gain of the actuator stage. The open loop gain required for 70 rad/sec cross over is then:

$$G_2 K = W_c Z_o A = 4200 \text{ lb/in}^2\text{/in.}$$

Figure 3.4.1 is a block diagram of the actuator stage. The

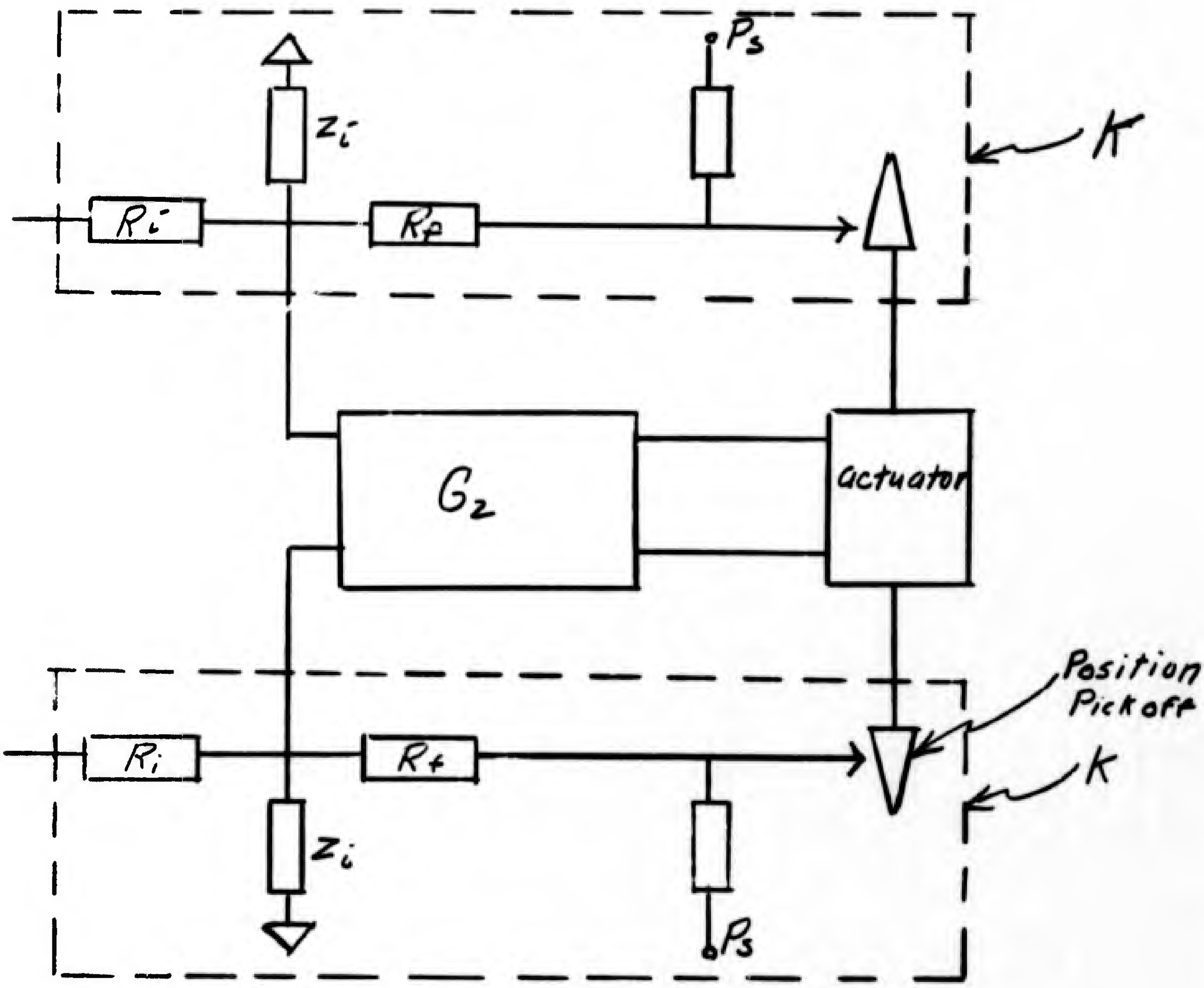


Figure 3. 4. 1 Block Diagram Actuator Stage

expression for K used in the open loop gain can be derived directly from Figure 3.4.1.

$$K = \frac{P_g Z_i R_i}{\left(R_f Z_i + R_i Z_i + R_f R_i\right)}$$

P_g is the gradient (lb/in^2/in) at the input of the feedback resistor. It is good design practice to make the input and feedback resistors equal. These resistors are orifices and are non-linear, however, the ratio of output to input can be made reasonably linear if equal resistors are used in a push-pull circuit. It is also desirable to make the loop gain independent of the input impedance of the first stage. This can be achieved by making $Z_i \gg R_i$ or R_f. K can then be expressed as:

$$K \approx \frac{P_g}{2}$$

P_g will be a direct function of the power supply across the fixed and variable orifices used as the position sensor. Push-pull gradients of 60 lbs/in^2/in. can be obtained with a 30 psig power supply. The required fluid amplifier gain then becomes:

$$G_2 K = G_2 \left(\frac{P_g}{2}\right) = 4200$$

$$G_2 = 140$$

The size of the output amplifiers for this block of gain can be determined from the maximum output impedance calculated for the actuator. An empirical equation for the output impedance is:

$$Z_o = \frac{0.66}{A_N} \sqrt{\frac{P_S \rho}{2}}$$

A_N is the nozzle area of the amplifier, P_S is the power supply and ρ is the mass density of the working fluid.

For the specified output impedance of 4.85 lbs. sec/in^5

$$A_N = 0.013 \text{ in}^2$$

The output stage nozzle flow is 7 gpm at 200 psig or the equivalent of 0.825 hp.

The fluid amplifier gain block will consist of 4 cascaded amplifiers. The three amplifiers at the input end will have a nozzle area of 0.0016 in^2.

The response of the fluid amplifiers must be sufficiently high to insure less than 50^o phase lag at the open loop crossover of 70 rad/sec. Measured response data is compared to calculated values in Figures 3.4.2 and 3.4.3. There is reasonably good correlation between the calculated and measured. This data can then be extrapolated to larger elements operating at other power supply levels. Extrapolating to the 0.013 in^2 output stage operating at 200 psig yields a predicted phase lag of 5^o at 70 rad/sec. The total phase shift of the cascaded elements should be less than 20^o at 70 rad/sec which is well within the system design goals.

Selection of the 200 psig power supply for the output stage was somewhat arbitrary. The factors which must be considered when making the selection are actuator bandwidth requirements, actuator area and the loop gain requirements of the actuator. The maximum actuator bandwidth is roughly proportional to $P_S^{5/4}$, and actuator area and loop gain requirements are inversely proportional. All of these factors favor a high power supply pressure. The product of flow and pressure drop across the nozzle of the output stage is independent of the supply pressure. For this application

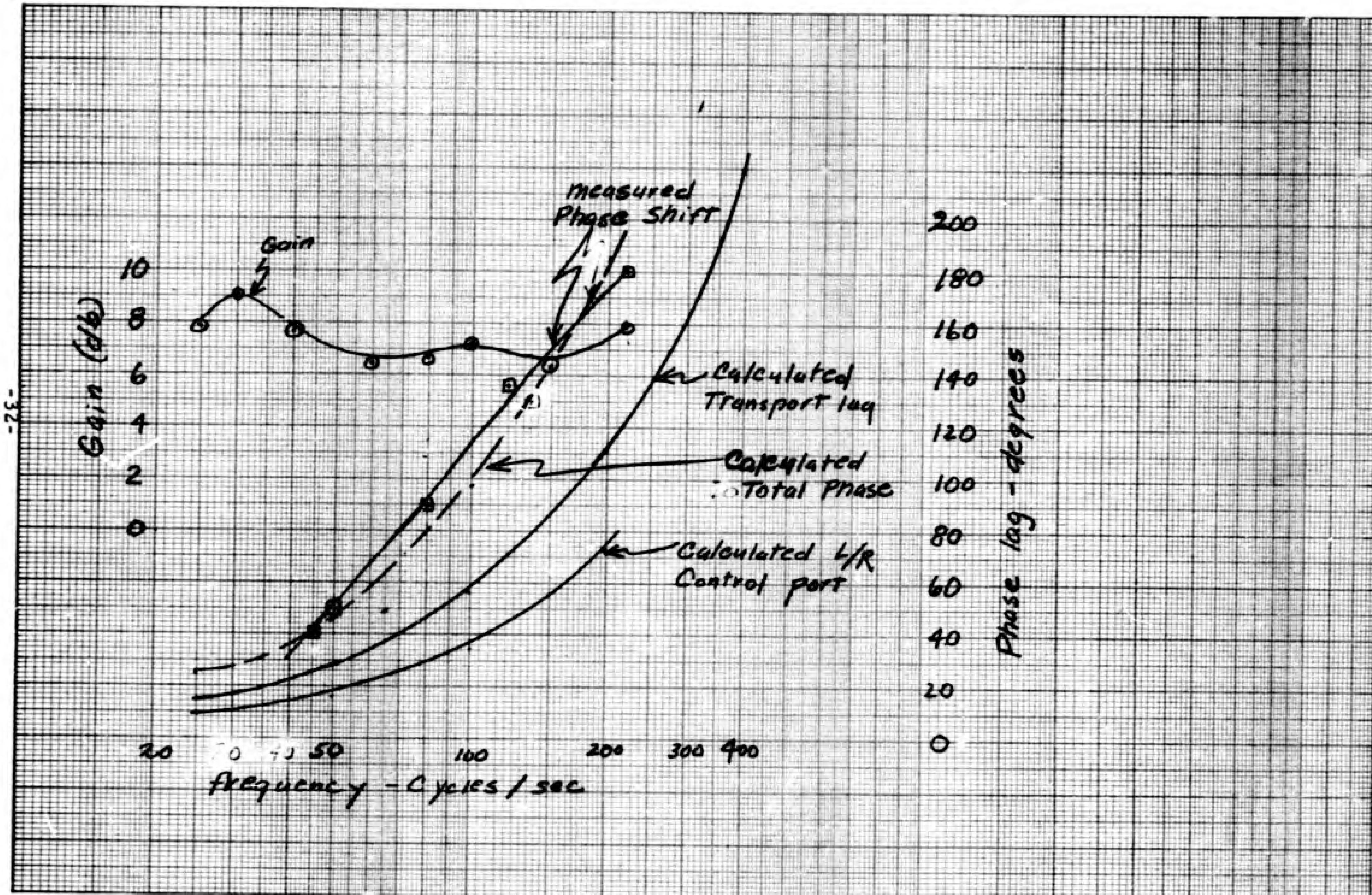


Figure 3. 4. 2 Response of Water Fluid Amplifier - 0. 04 x 0. 04 in. Nozzle - 5 psig Supply.

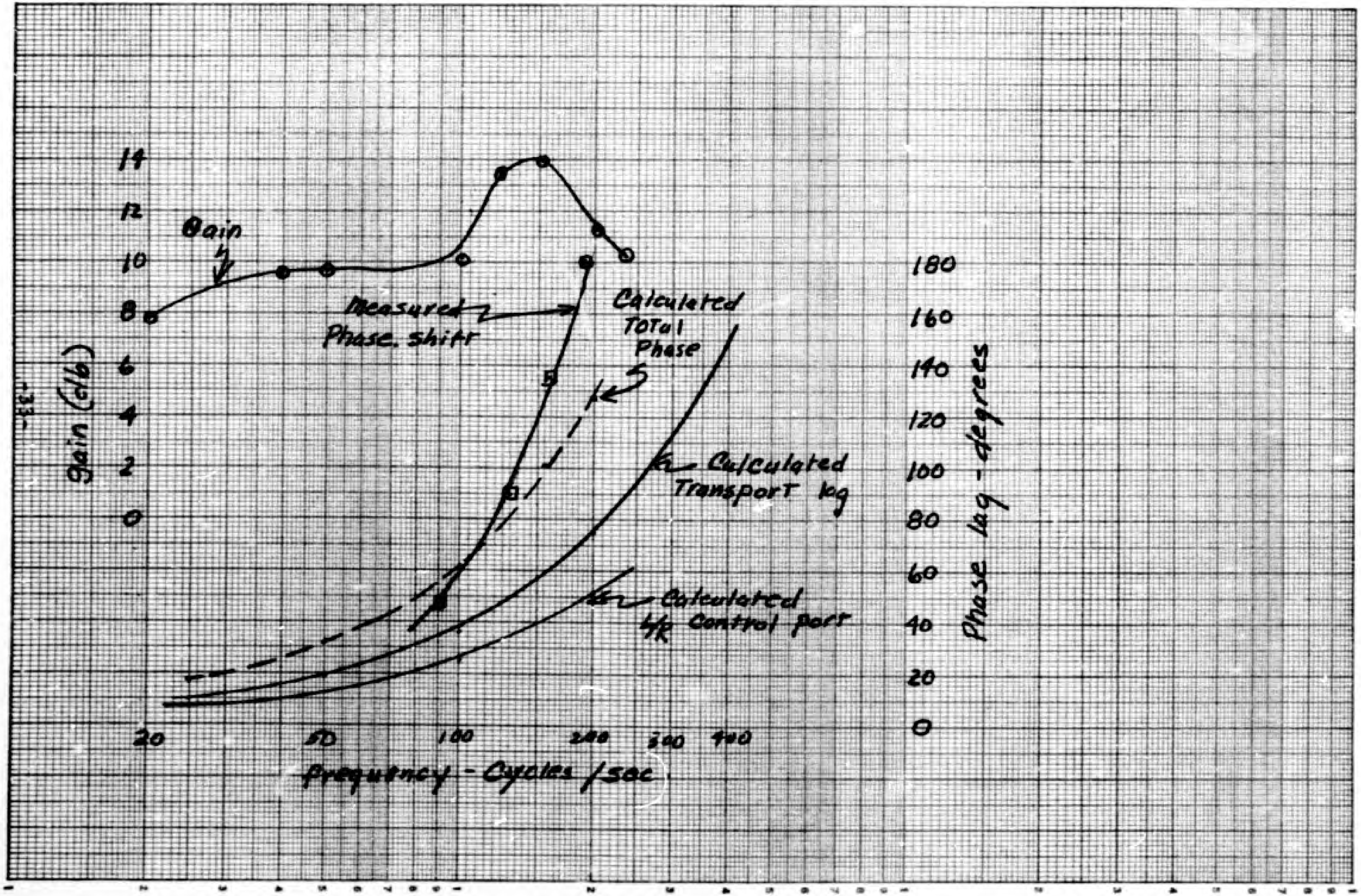


Figure 3. 4. 3 Response of Water Fluid Amplifier - 0. 04 x 0. 04 in. Nozzle - 10 psig Supply.

the 70 rad/sec bandwidth requirement sets a minimum supply pressure of 70 psig.

The scale model actuator for the feasibility model uses a 0.0016 amplifier as the actuator driver. A 25 psig power supply is required to give the same bandwidth as the full scale amplifier operating at 200 psig. The actuator area and stroke were selected to give stroke times identical to the full scale actuator.

A circuit diagram of the scale model actuator is shown in Figure 3.4.4. Three cascaded stages of 0.0016 in^2 elements are used. The circuit board is a planor configuration fabricated in photo sensitive glass by a photo etching process. The primary design consideration was to minimize the inductance of interconnecting passages to obtain maximum bandwidth. The measured gain of the three stages is 350. The desired open loop gain of 4200 is obtained by adjusting the supply pressure across the position pick off orifices.

A measured closed loop attenuation vs. frequency plot is shown in Figure 3.4.5. The bandwidth is 63 rad/sec and is highly damped. The bandwidth could be extended by a factor of at least 2:1 and still have acceptable damping.

Figure 3.4.6 is a sketch of the actuator assembly showing the displacement pick offs and the air metering valve for the scale air turbine. The actuator is a double ended actuator. The displacement pick off consists of a slot in the cylinder wall, a land on the actuator shaft moves along the slot varying the discharge orifices from cavities A and B. Flow into the cavities is controlled by two fixed orifices. A push-pull pressure signal

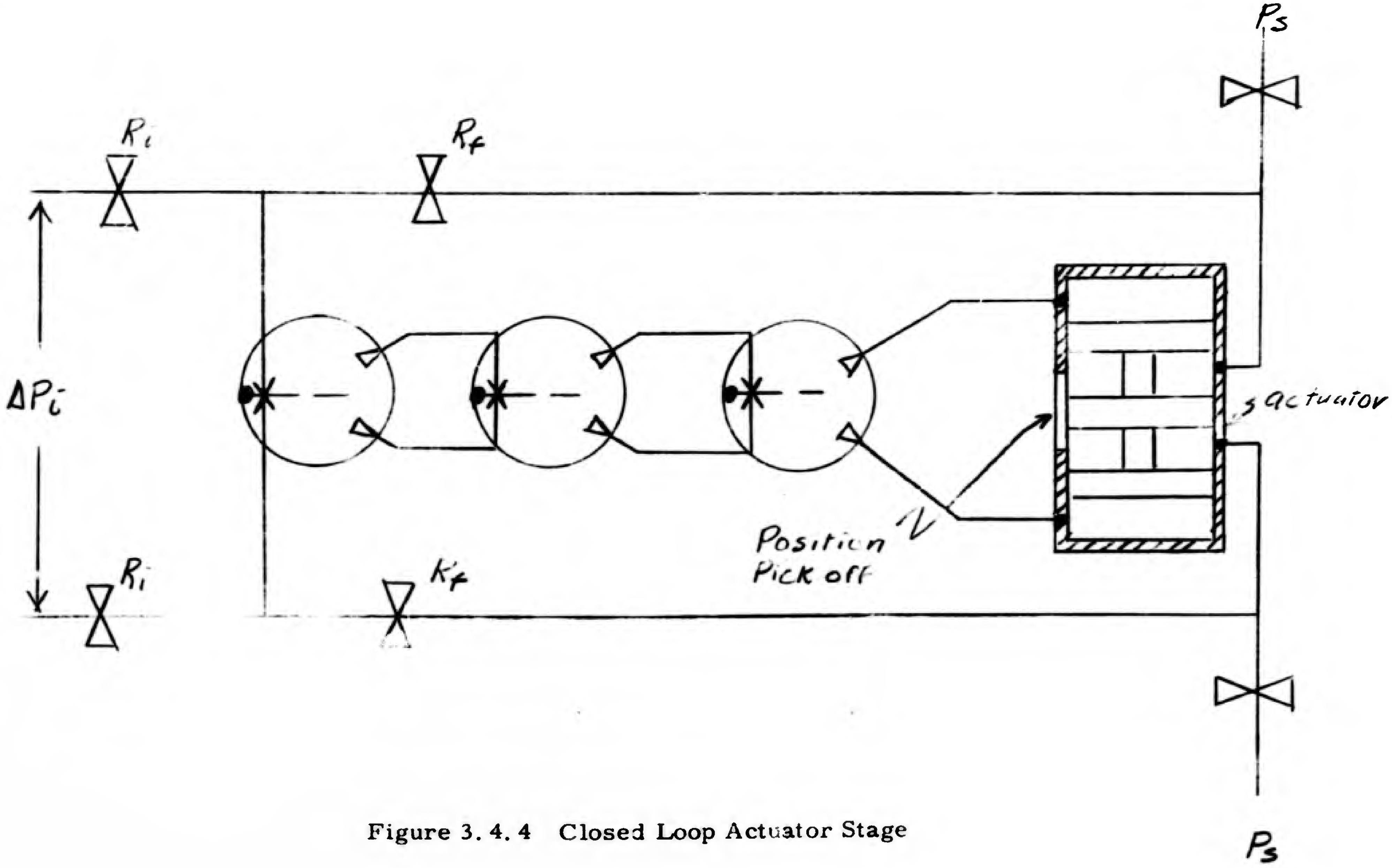


Figure 3. 4. 4 Closed Loop Actuator Stage

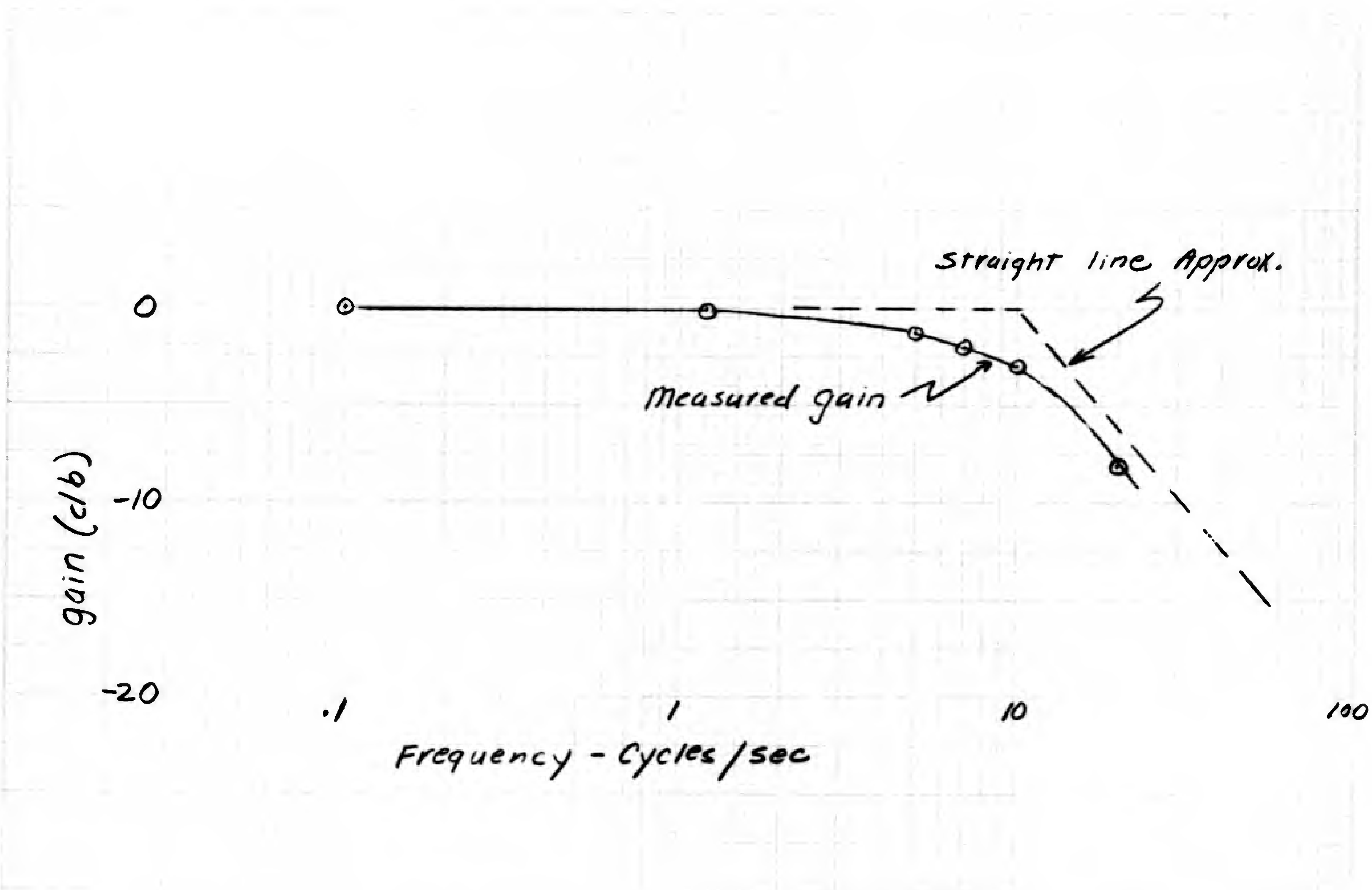


Figure 3. 4. 5 Gain of Actuator Stage. Ratio of Displacement to Input Pressure.

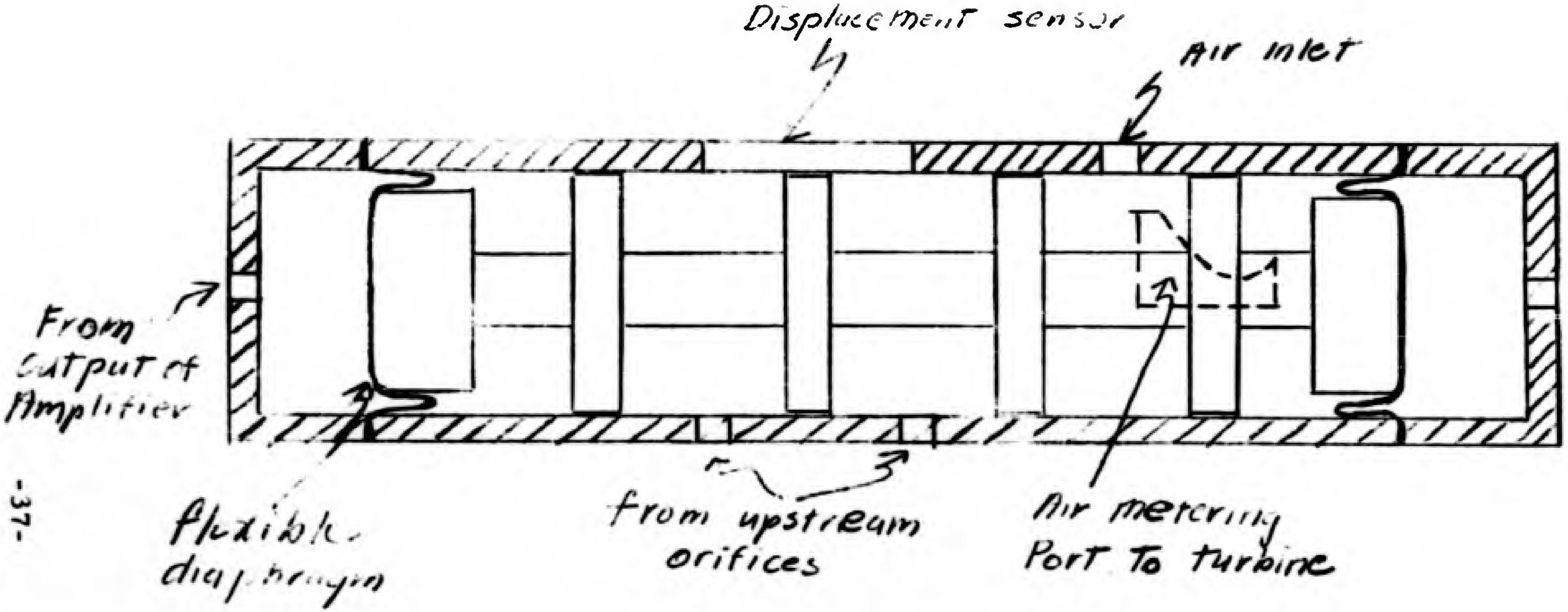


Figure 3.4.6 Cutaway Sketch of Actuator Assembly

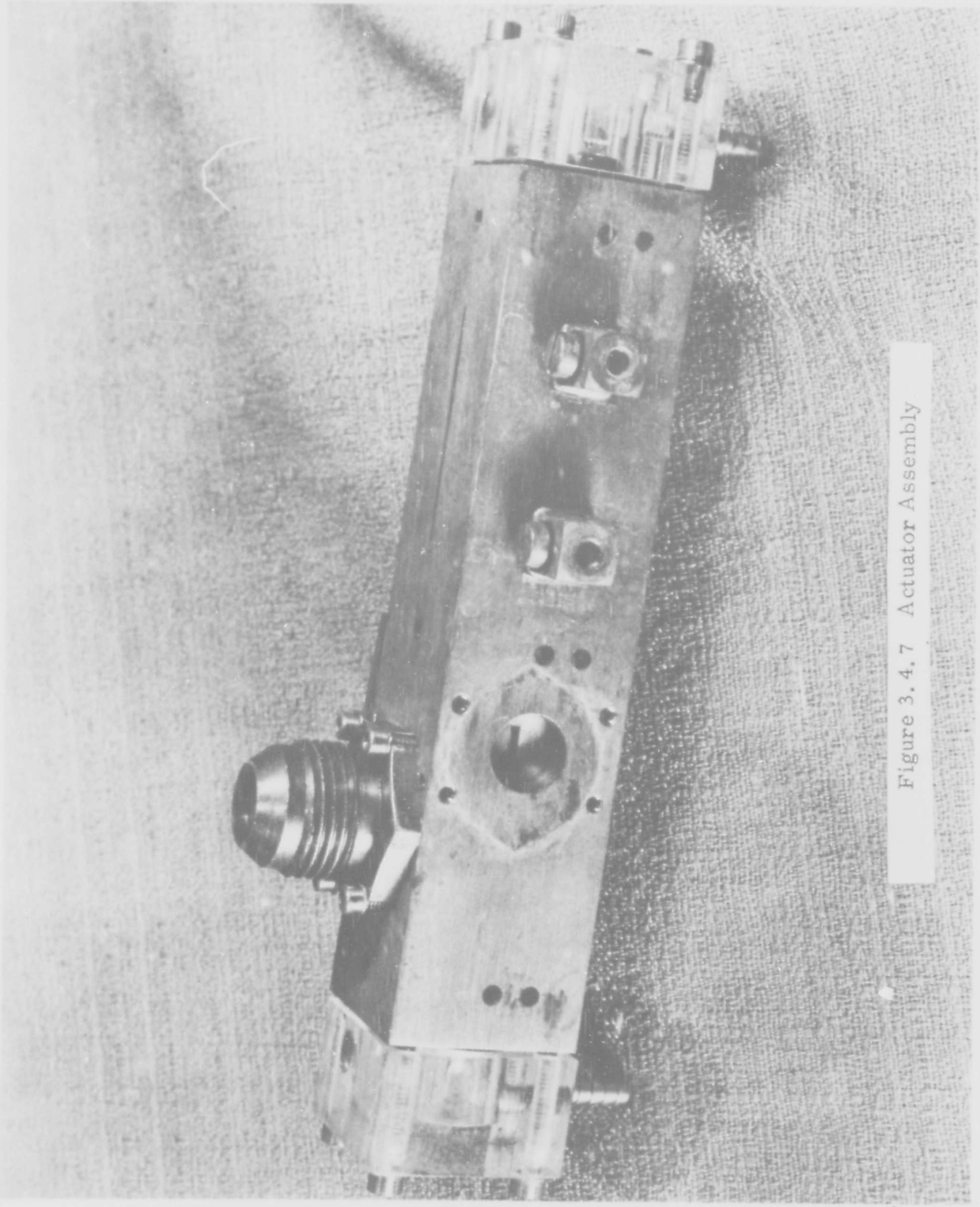


Figure 3.4.7 Actuator Assembly

proportional to displacement of the actuator is developed between cavities A and B. This signal is fed back through the feedback orifices and summed with the speed error signal.

Figure 3.4.7 is a photo of the assembled actuator.

The profile of the air metering valve was calculated to give a linear torque vs. displacement characteristic to the air turbine when supplied with a 50 psig air supply.

3.5 Reset Circuit

The pertinent characteristics of the reset loop are the static gain and the reset time constant. The static gain requirements are set by the steady state regulation limits of MIL-G-21410. Overall system damping requirements determine the minimum value of the reset time constant. The specific requirements for a DL-5 class of turbine-generator are a steady state gain at least 10 times the proportional gain path and a minimum reset time constant of 1 sec.

The simplest method of obtaining the reset action is with an open loop actuator stage. This approach is limited to balanced low reaction steam valves hence was not considered adequate.

The general method selected uses a multi-stage high gain fluid amplifier with a low frequency lag break. This amplifier is used in parallel with the proportional gain stage. The proportional gain path sets the system crossover, the high gain amplifier controls the steady state droop characteristics.

The primary problem in the reset circuit is obtaining the required low frequency lag break without resorting to moving parts. With compressible fluids the low frequency break can be readily achieved by using a capacitor (volume) at the output of an amplifier stage.

The two approaches considered for obtaining the low frequency break in a compressible fluid are:

1. Utilizing the inductance of a long tube. A lag break occurs at a frequency determined by the total circuit resistance (R_t) divided by the inductance (L).
2. Utilizing the capacitive effects of a compressible fluid volume. The interface between the compressible and incompressible fluid is made through a soft diaphragm. Other variations of this approach are a spring loaded actuator or a soft bellows.

3.5.1 Reset Time Constant with Inductance

The reset time (T_R) is defined as the reciprocal of the lag break. The time constant of a fluid inductor is $\frac{L}{(R_i + R + R_o)}$

R_i and R_o are the input and output impedances of the device used to excite the inductor and to pick off a signal. R is the resistance of the tube.

$$\frac{L}{R} > T_R$$

For laminar flow conditions

$$\frac{L}{R} = \frac{\rho D^2}{32\mu}$$

where ρ is mass density, D is the diameter of the tube and μ is viscosity.

To obtain a 1 sec. time constant:

$$D > \sqrt{\frac{32\mu}{\rho}} = 0.22'' \text{ for water at } 70^{\circ}\,F$$

This is the absolute minimum tube diameter and will approach a 1 sec. time constant only if $(R_o + R_i) \ll R$.

In a practical system using fluid amplifiers to excite the tube, R_o and R_i cannot be made negligible compared to the tube resistance. The L/R time constant of the tube must be made considerably larger than the desired reset time. Figure 3.5.1.1 is a plot of tube length vs. tube diameter to obtain a 1 sec. time constant. Length is minimized when the time constant of the tube is made equal to twice the desired reset time constant or where $R = R_o + R_i$. The minimum length of 1660 feet is obtained with a 0.31" diameter tube. The excessively long line lengths required dictate an approach which utilizes the L/R time constant in some other manner than in a straight forward series network.

One approach which multiplies the physical L/R time constant is an operational amplifier using an inductance in the feedback. Two configurations for obtaining a time constant multiplication are shown in Figures 3.5.1.2 and 3.5.1.3. The circuit of Figure 3.5.1.2 uses a high gain amplifier, the output is fed back and summed with the input by $R_f + R_i$. An inductance shunts the summing point to ground, the resistance R is the inherent resistance of the tube. Qualitatively the circuit functions as follows (Refer to Figure 3.5.1.4). At low frequency the open loop gain is less than unity and the closed loop gain is controlled by the forward gain of the amplifier. At high frequencies the loop gain is much greater than unity and the closed loop gain is controlled by the feedback. The transition is at the open loop crossover. The result is a lag time constant in the

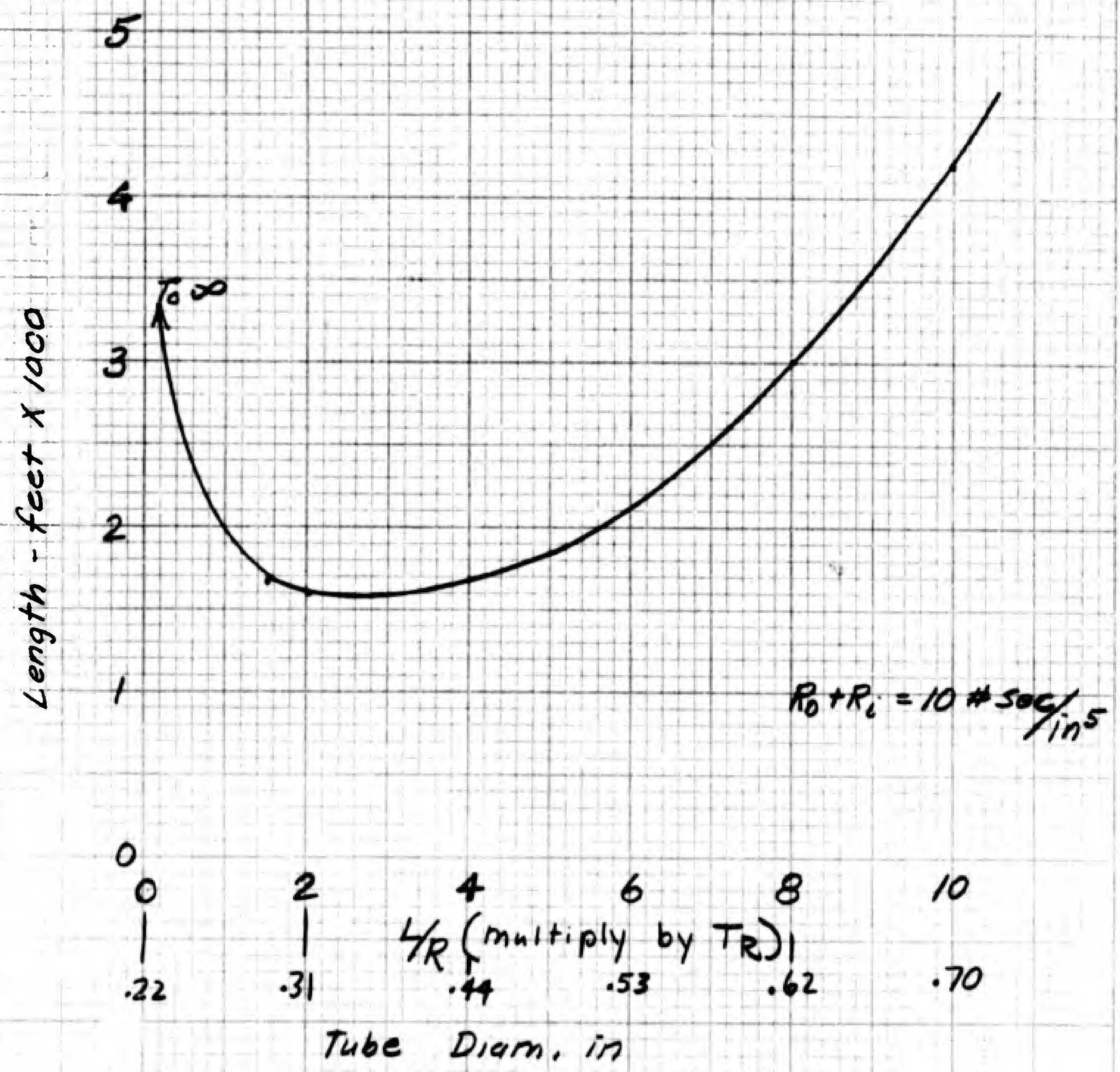


Figure 3.5.1.1 Length of Tube Required for a 1 sec Time Constant

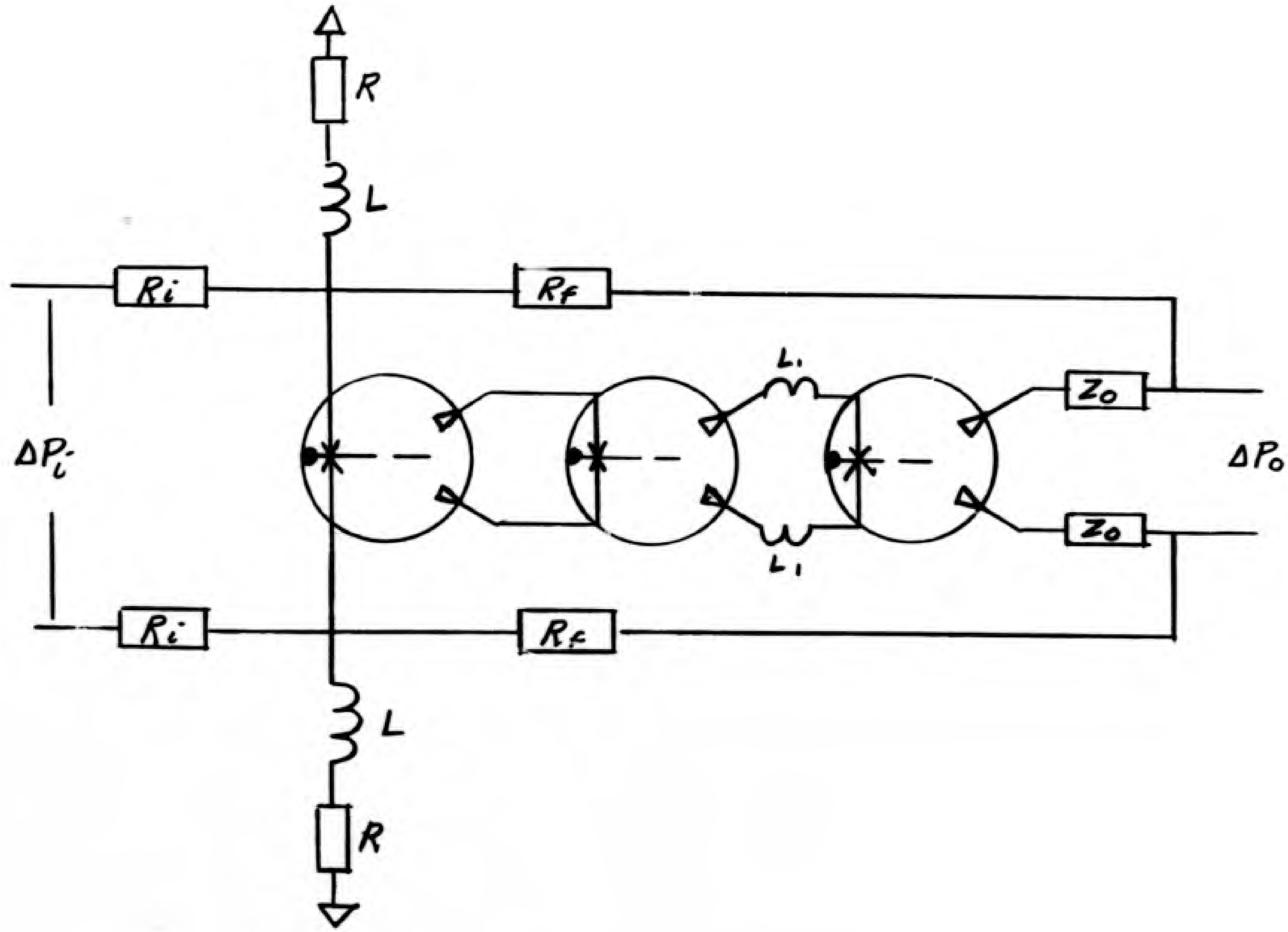


Figure 3.5.1.2 Time Constant Multiplication Circuit.

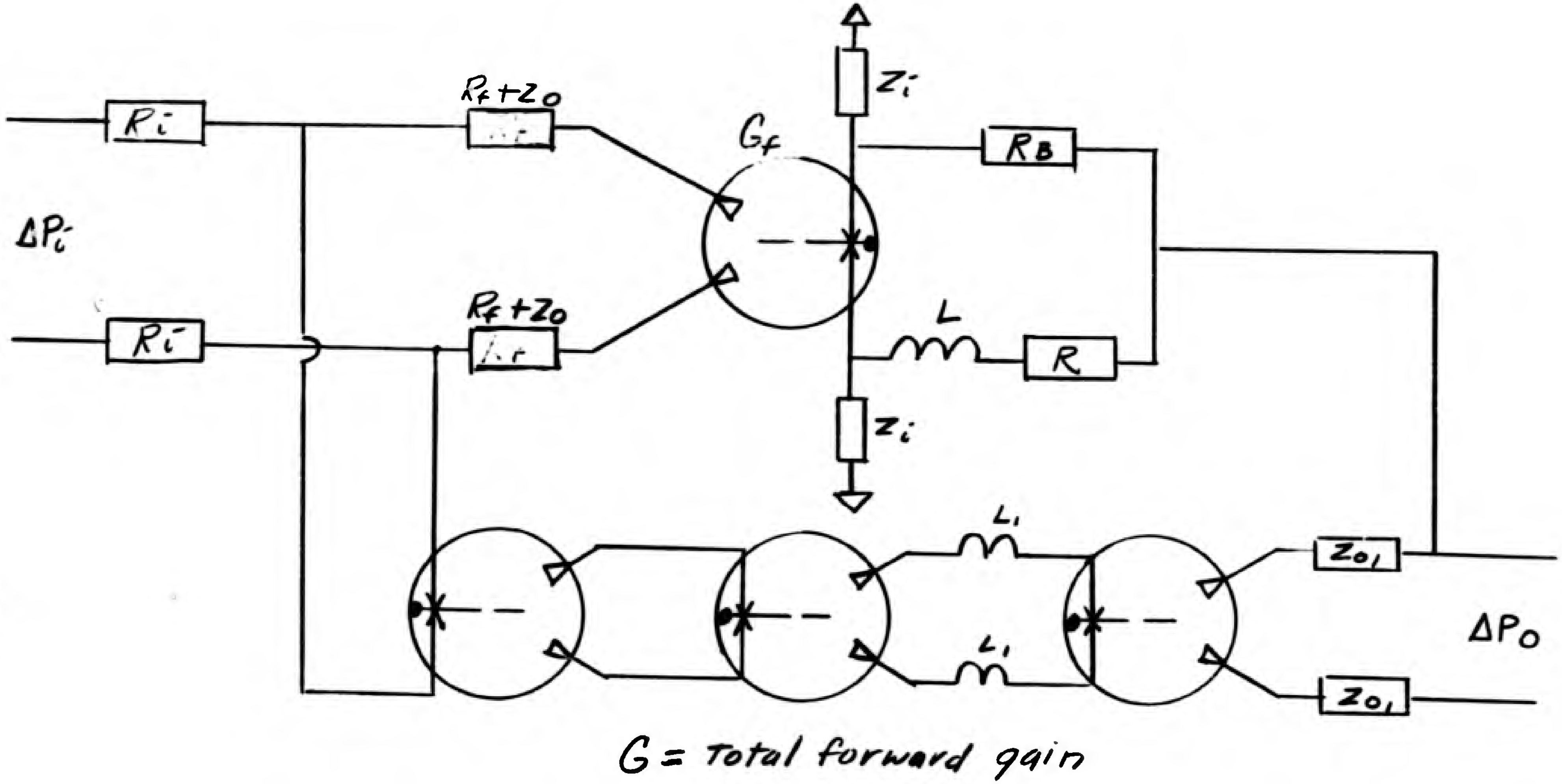


Figure 3. 5. 1. 3 Time Constant Multiplication Circuit.

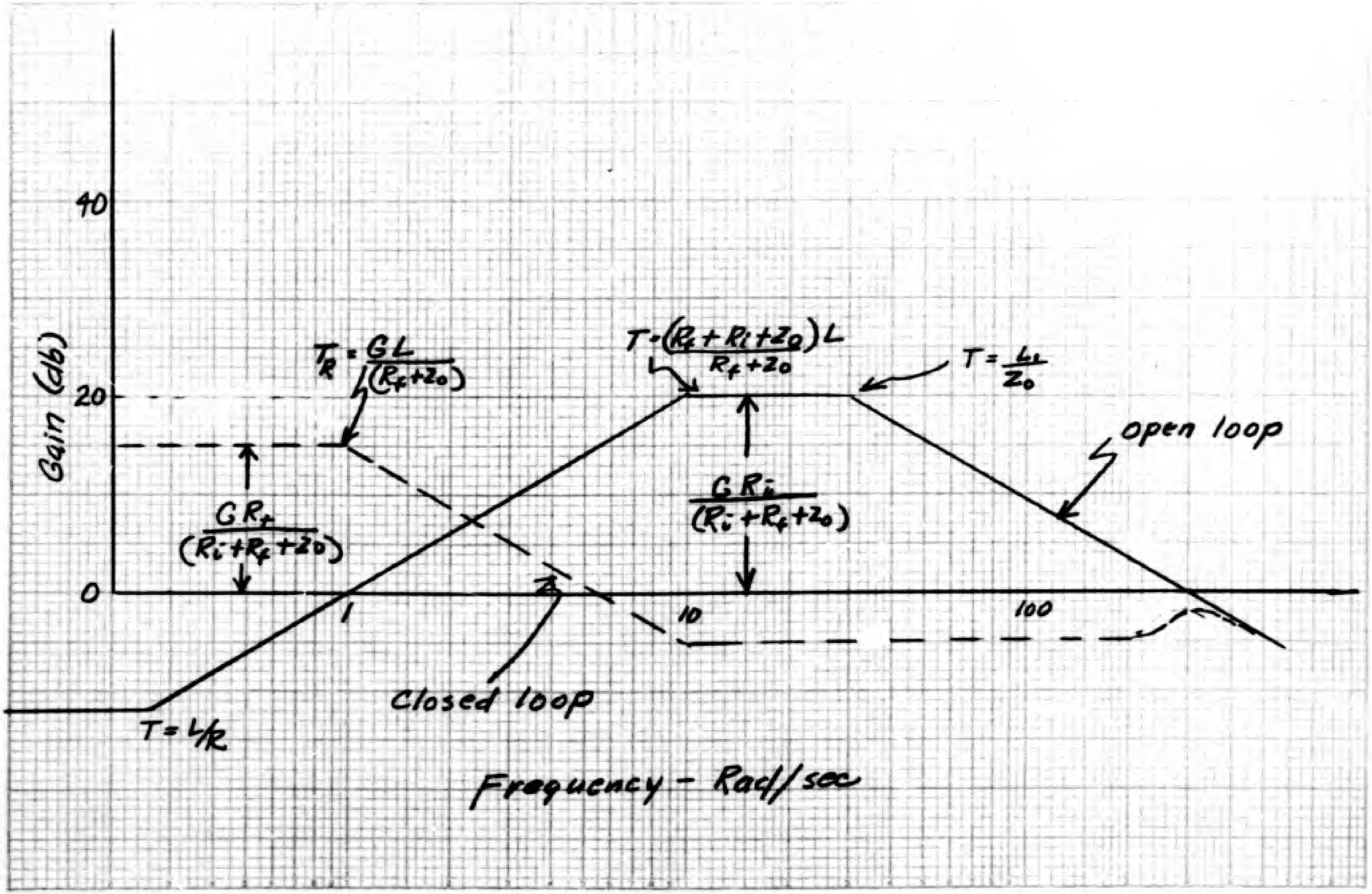


Figure 3.5.1.4 Attenuation Vs. Frequency Plot for Time Constant Multiplier

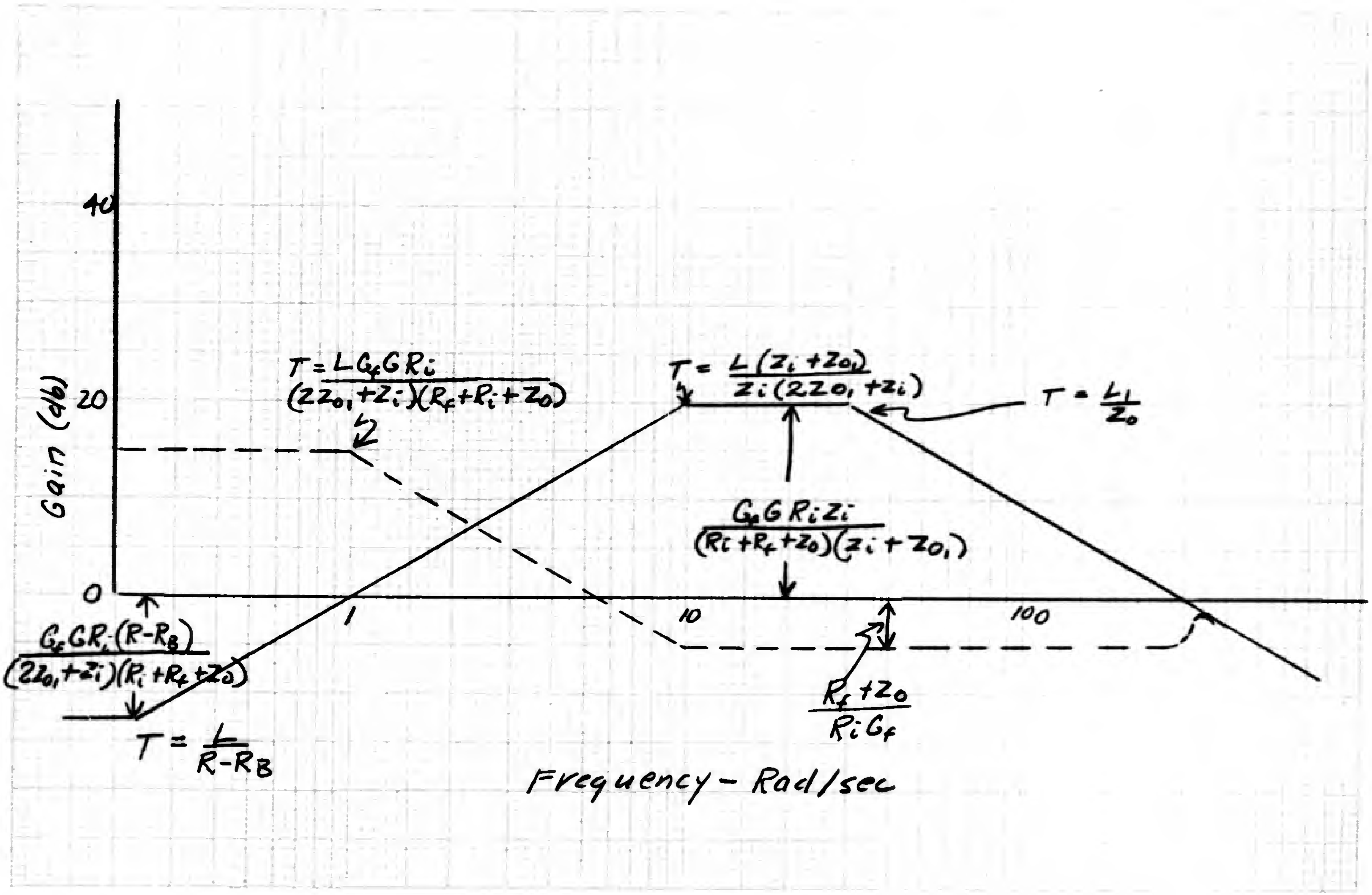


Figure 3.5.1.5 Attenuation Vs. Frequency Plot for Time Constant Multiplier

closed loop characteristic, the lag time constant occurs at the open loop crossover. This lag occurs at $G\left(\frac{L}{R_I + R_O}\right)$ where G is the gain of the fluid amplifiers. Comparing this to the first method, $(R_I + R_O)$ will have approximately the same numerical value as $(R_i + R_o)$, hence the time constant is multiplied by a factor of G or the tube length decreased by the same factor. The L/R time constant of the tube used in the feedback must still be greater than the desired reset time, the minimum tube diameter is then on the order of 0.3 inches.

The multiplication factor that can be obtained is limited by the response time of the fluid amplifier. The open loop high frequency crossover must be made with adequate phase margin. For a given reset time constant the high frequency crossover will be directly proportional to G. For the specific application the reset circuit will operate with supply pressures ranging from 2 - 20 psig on the amplifiers. From measured response data (Figures 3.4.2 and 3.4.3) the maximum loop crossover for the cascaded amplifiers will be on the order of 200 rad/sec. and the maximum multiplication factor will be 200. This would reduce the 1660 foot tube length required for the first method to 8.3 feet of 0.3" diameter tubine.

The circuit shown in Figure 3.5.1.3 utilizes the inductive time constant in a different manner. The inductance is used in conjunction with an amplifier to form a derivative circuit which is then used in the feedback path of the operational amplifier. The basic advantage of this approach is the elimination of the restriction on minimum tube diameter.

Referring to Figure 3.5.1.5 the time constant determined by $\frac{L}{(R - R_B)}$ must be greater than the desired reset time. The resistance of the tube (R) can be cancelled by an equal resistor (RB) in the opposite control port of the derivative amplifier. In the derivation of the circuit equations equal control port impedances were assumed. In practice the unbalance in the total resistance including the control ports will determine the time constant. The only restriction on tube diameter is that R should be small compared to the control port impedance. This minimizes the need for close matching of R & R_B. In a typical application an R of 1 lb. sec/in^5 is acceptable, this corresponds to 5 feet of 1/8" ID tubing with water at 70°F as the fluid. Assuming the same loop gain as in the previous approach a 1 sec. time constant can be obtained with a 1.5' length of 1/8" tubing.

This approach was sufficiently promising to warrant hardware construction and evaluation. The key component is the derivative circuit shown in Figure 3.5.1.6. The basic requirements can be deduced from the transfer function. To obtain a reset time of 1 sec. the low frequency break in the derivative circuit should be at least 2 sec.

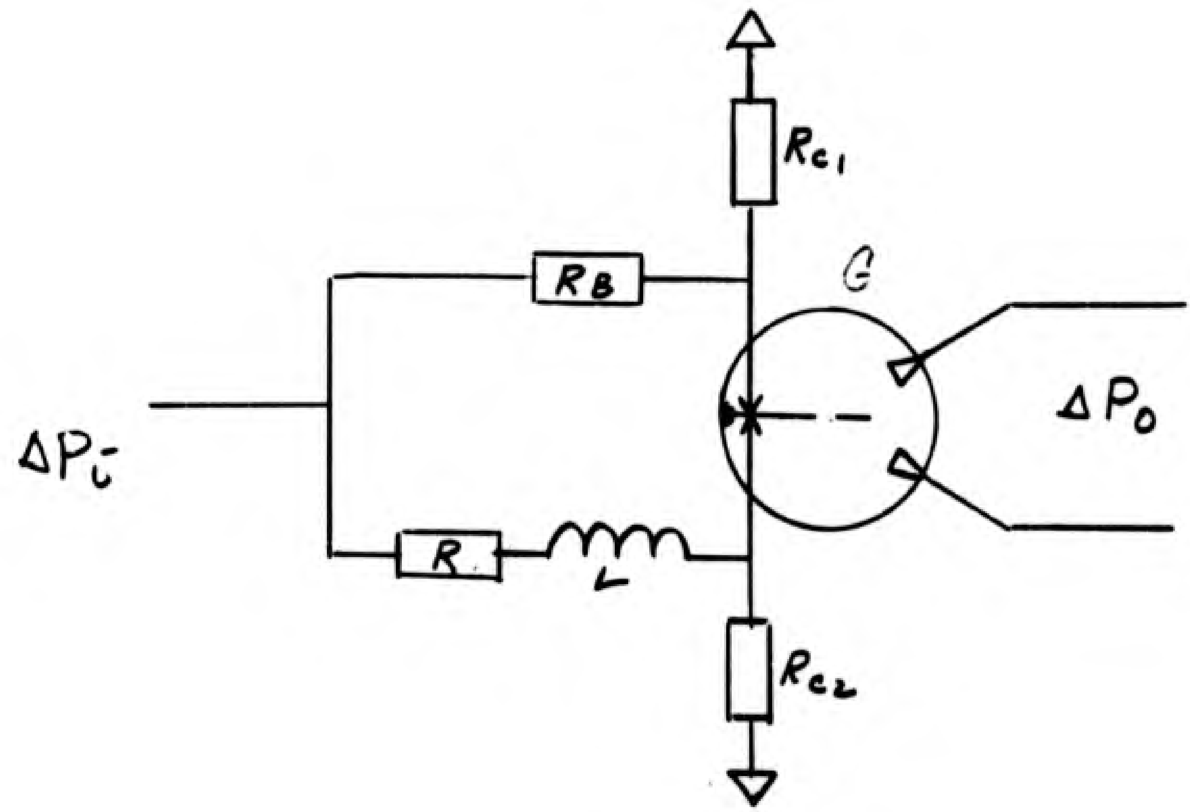
$$\frac{L\,R_{c1}}{R_{c1}R - R_{c2}RB} \geq 2 \text{ sec.}$$

A 30 inch length of 1/8" tubing was selected for the inductance. This yields an inductance of 0.22 lb. sec^2/in^5 and a resistance of 0.6 lb. sec/in^5.

The degree of balancing required on R & R_B can be deduced by assuming balanced control port impedances.

$$\left|R - R_B\right| \leq \frac{L}{2} \leq 0.11 \text{ lb. sec/in}^5 \quad \text{or approx. } 18\%$$

The balance requirements on the control ports can be similarly deduced. For a typical control port impedance of 10 lb. sec/in^5 they must be balanced to 1.83 lb. sec/in^5. In general, the requirement for control port balance can be relaxed by using a larger inductance whereas the balance requirements on the tube resistance is independent of inductance, providing the ID of the tube is maintained constant.



Transfer Function

$$\frac{P_o}{P_i} = \frac{G\,(R_{c1}R - R_{c2}R_B)\left(1 + \dfrac{S\,L\,R_{c1}}{(R_{c1}R - R_{c2}R_B)}\right)}{(R_B + R_{c1})\,(R + R_{c2})\left(1 + \dfrac{S\,L}{R_{c2} + R}\right)}$$

For ideal amplifier where $R_{c1} = R_{c2}$ & $R = R_B$

$$\frac{P_o}{P_i} = \frac{G\,S\,L\,R_c}{(R + R_c)^2\,\left(1 + \dfrac{SL}{R_c + R}\right)}$$

Figure 3.5.1.6 Derivative Circuit for Water Amplifier

The gain requirements for a 1 sec. time constant (See Fig. 3.5.1.5) are determined by

$$\frac{LG_f G R_i}{2 Z_{o1} + Z_i} = 1$$

Using typical values of R_i, Z_o, and Z_i

$$G_f G = 230$$

A total of 4 fluid amplifiers will be required. Three are cascaded in the forward gain loop and one is used in the feedback.

From the standpoint of closed loop stability the gain of concern is

$$\frac{G_f G R_i Z_i}{(R_i + R_f + Z_o)(Z_i + Z_{o1})} \approx 46$$

Open loop crossover will then be in the order of 50 rad/sec which is within the response capabilities of the fluid amplifiers.

Figure 3.5.1.7 shows test results on the derivative circuit designed for this application. Two low frequency gain levels are shown. The lower plot represents the minimum that could be obtained by balancing of external resistors. The higher gain curve represents the minimum that can be expected without periodic adjustment of the external resistor. The minimum obtained comes very close to meeting the design goal of 2 sec. on the derivative circuit. Because of low signal to noise ratio on the circuit the low frequency break was extrapolated from DC measurements and the lowest excitation frequency giving reliable data.

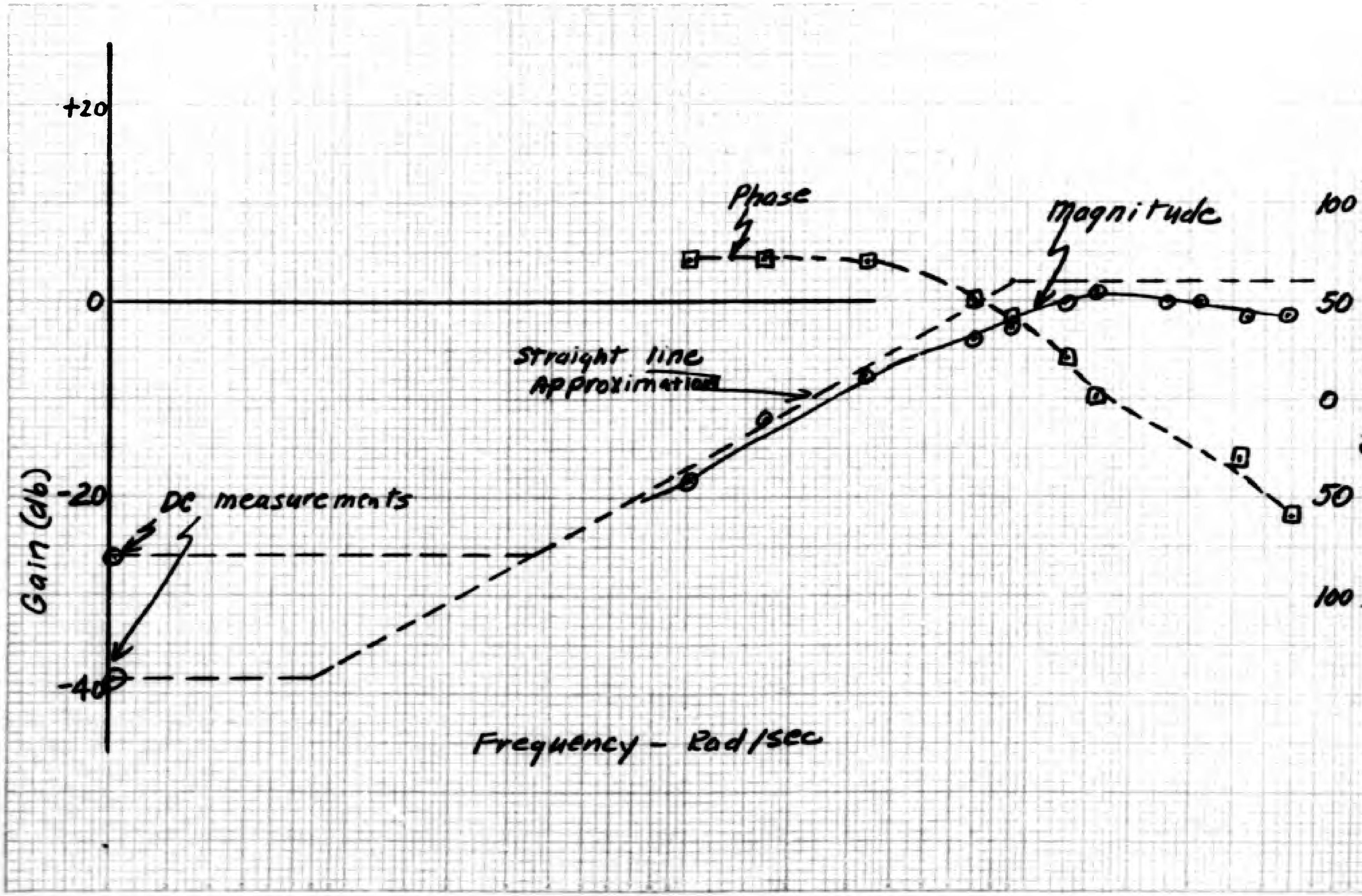


Figure 3. 5. 1. 7 Measured Response on Derivative Circuit

This derivative circuit was combined with a three-stage operational amplifier to form the reset circuit. Test results on the overall reset circuit are shown in Figure 3.5.1.8. The reset time constant is 1/4 sec. which misses the design goal by a factor of four. There is a peak in the response curve at 170 rad/sec. This corresponds to the open loop cross-over frequency and indicates that the overall response of the amplifiers is marginal at that frequency.

The limitations to the maximum time constant that could be obtained were low signal to noise ratio on the fluid amplifiers and the degree of balance obtained on the derivative circuit. Noise is critical in this application because of the broad bandpass required in the open loop. The noise generated in the fluid amplifiers is amplified by each succeeding stage. The output noise is then fed back with very little attenuation to the input stage of the amplifier. The loop gain must be limited to a value which will prevent saturation of the amplifiers.

A Company sponsored noise program identified the primary sources of noise in the fluid amplifiers and recommended design changes to reduce the noise. Reduction of noise by a factor of 3:1 was demonstrated, however, the results were too late to incorporate in working hardware.

Although the design goals were not met with this approach, the basic feasibility of the concept was demonstrated. The 1/4 sec. time constant was obtained with a 30 inch length of 1/8" I. D. tubing as contrasted to the 420 feet of 0.3" I. D. tubing which would be required without time constant multiplication. With more basic work on noise reduction and establishment of the critical parameters of the derivative circuit this approach still shows promise for meeting the design goals. It is the only known approach

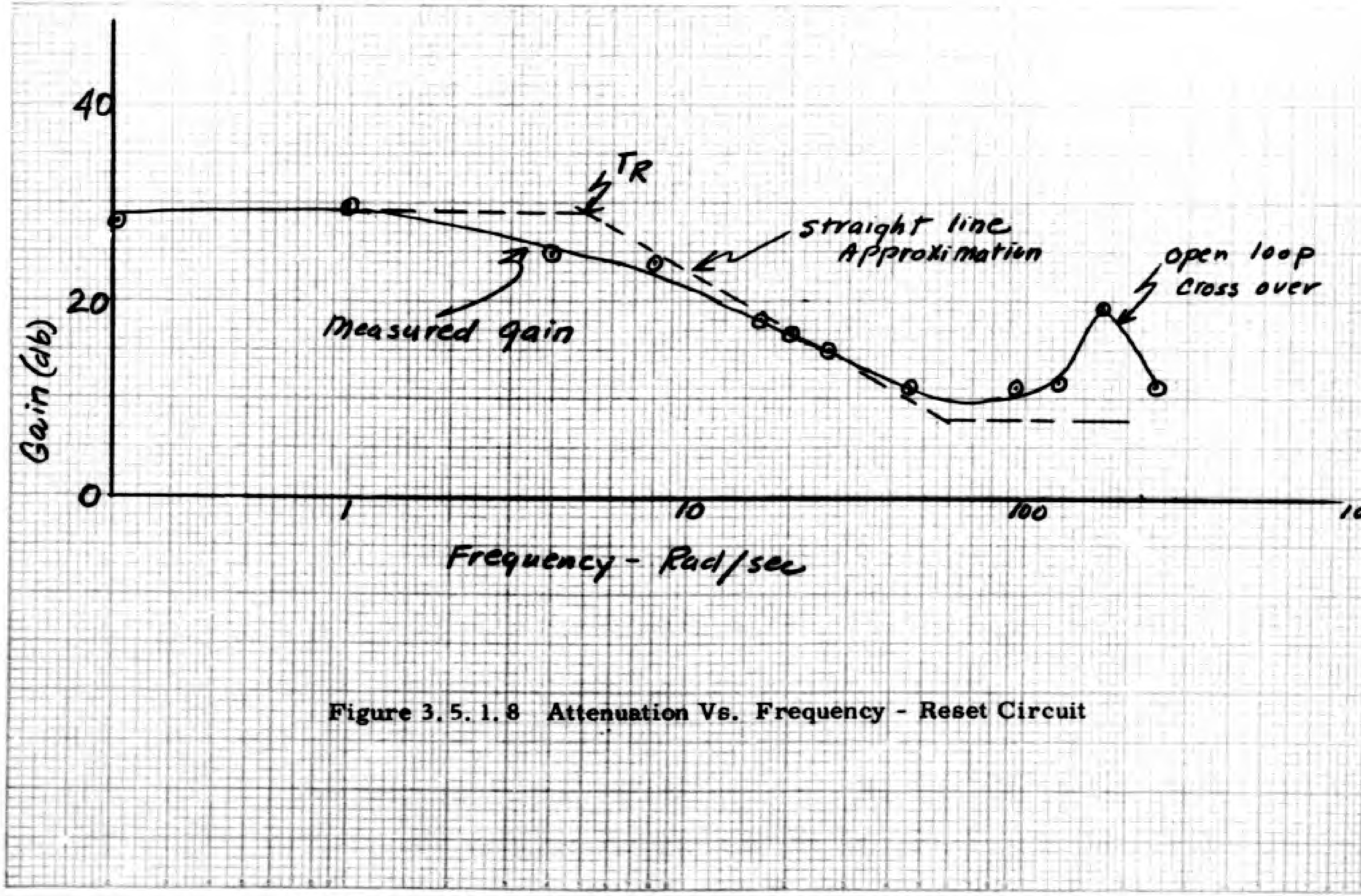


Figure 3.5.1.8 Attenuation Vs. Frequency - Reset Circuit

for a no-moving parts reset loop which has possibilities of functioning on fluids of relatively high viscosity.

3.5.2 Reset Loop Using a Compressible Fluid Volume

The circuit schematic for this approach is shown in Figure 3.5.2.1. The reset time constant is

$$T_R = \frac{Z_o R_s C}{(Z_o + R_s)}$$

Where C is the equivalent capacity.

For a 1 sec. time constant and using representative values for $(Z_o + R_s)$ of 10 lbs. sec/in^5.

$$C = \frac{Z_o + R_s}{Z_o R_s} (T_R) = 0.2 \text{ in}^3/\text{lbs}/\text{in}^2$$

Assuming a cylindrical configuration for the capacitor:

$$C = \frac{\pi D^2}{4} \left(\frac{L}{P_o}\right)$$

Where D is the diameter of cylinder, L is the length and P_o is the initial pressure of gas.

Using a typical fluid amplifier output impedance and a 1 1/2" diameter cylinder, the required length is 2.5 inches for a 1 sec. time constant. This capacitor is non-linear, however, if one is used at each of the two outputs of a push-pull amplifier the sum remains relatively constant over a wide range. A linear capacitor can be obtained by replacing the pneumatic spring with a mechanical spring.

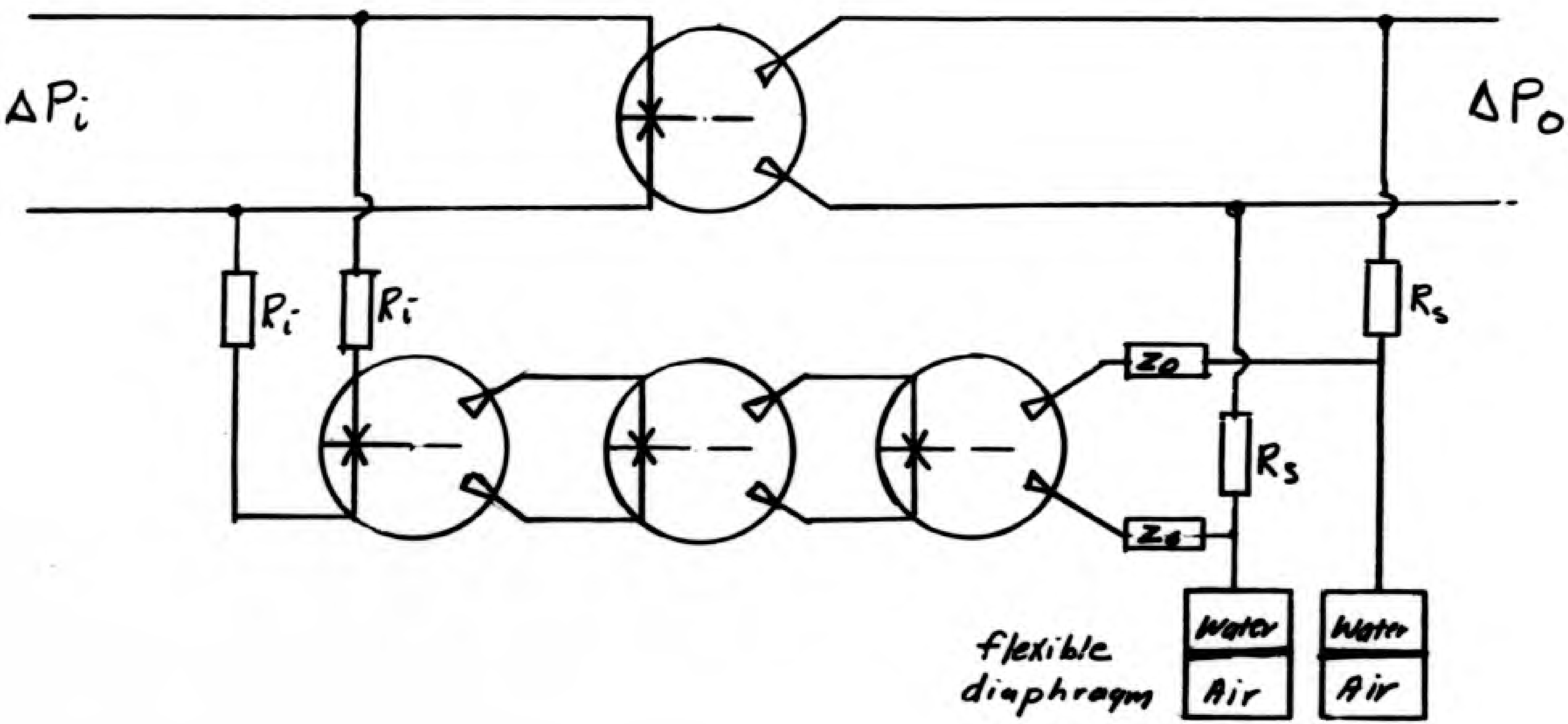


Figure 3.5.2.1 Reset Circuit Using a Compressible Fluid Volume.

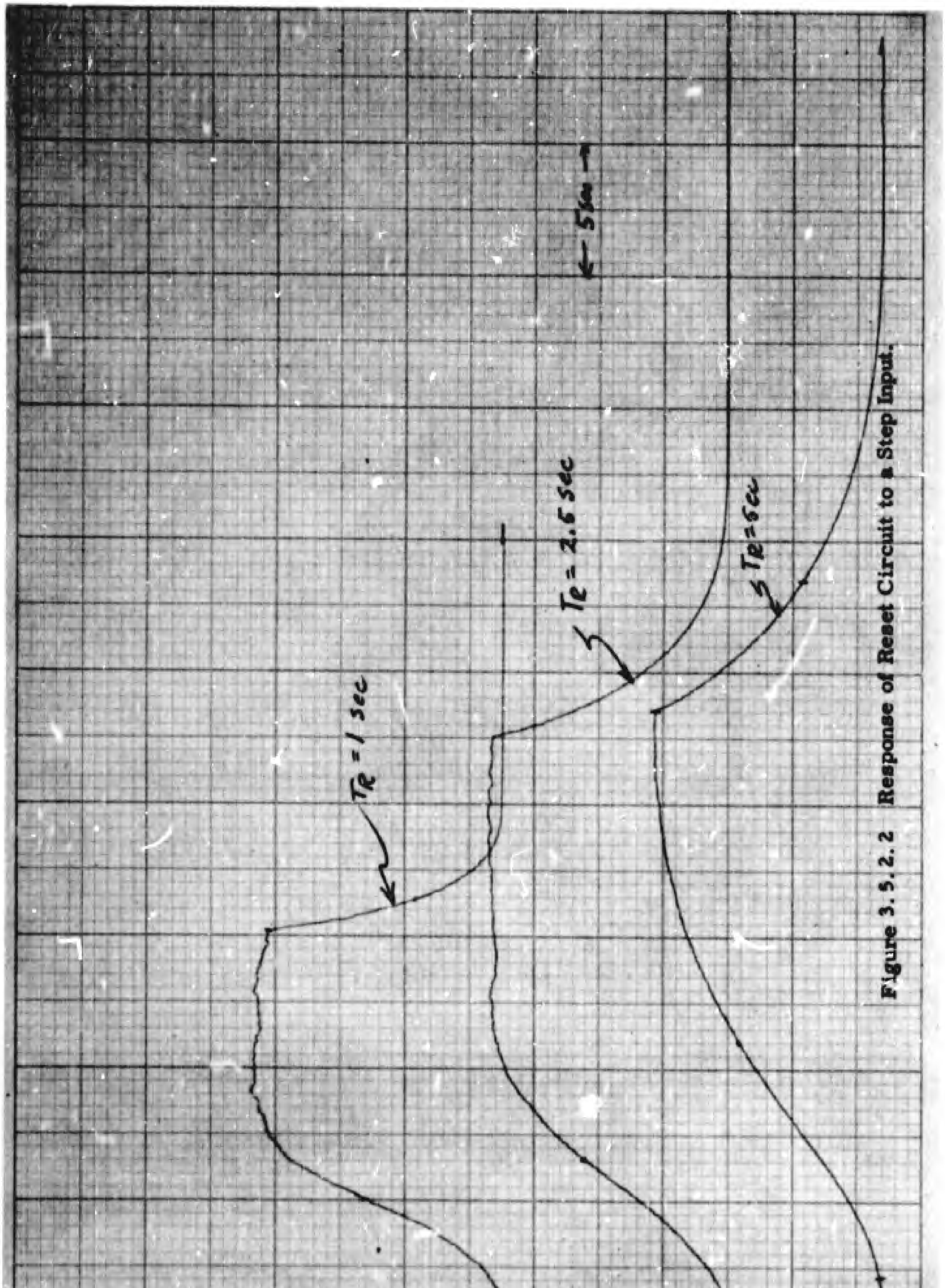


Figure 3.5.2.2 Response of Reset Circuit to a Step Input.

The capacitor for the feasibility model uses a direct water to air interface to facilitate changing the reset time over broad limits. This has worked out very well for test and evaluation of the overall loop.

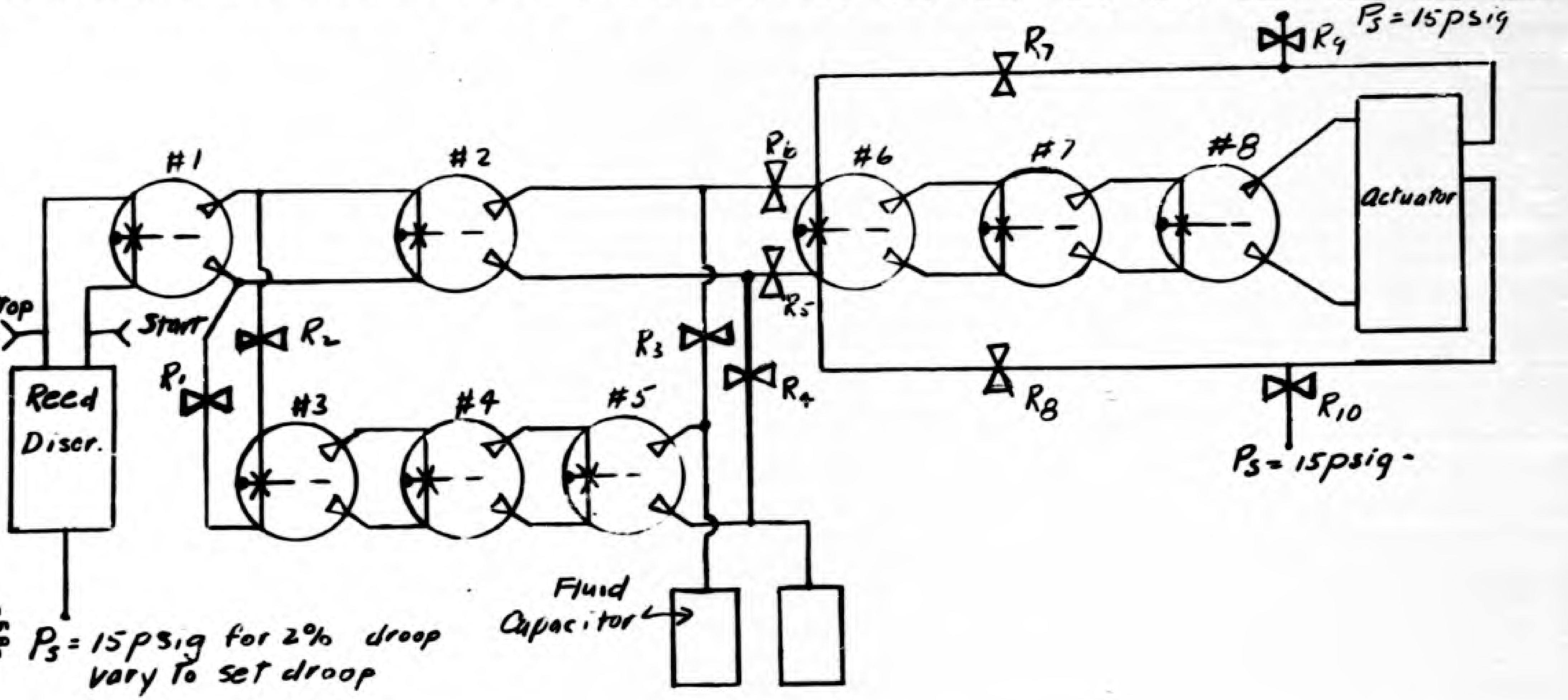
The final configuration is as shown in Figure 3.5.2.1. The three elements used in the high gain loop are photo etched in glass in a planor circuit board. The blocked load gain of the amplifier is 350. The losses across the input and output summing resistors reduce the gain to 60 in the overall circuit. A similar amplifier with a gain of 5 is used in the proportional path.

The response of the loop to step inputs is shown in Figure 3.5.2.2. Reset time constants in excess of that required for this application are feasible. This approach is tolerant of self generated noise in the fluid amplifiers. The output noise is filtered by the capacitor and there is no feedback to the input stages.

3.6 Overall Control Loop

A circuit diagram of the complete speed governing loop is shown in Figure 3.6.1. A total of eight proportional amplifiers are required. All of the elements are basically the same design and size (0.0016 in^2 nozzle). With the exception of the actuator stage the number and size of elements are representative of a full scale design. The full scale actuator requires a 0.013 in^2 nozzle on the output stage and three additional amplifiers with a 0.0016 in^2 nozzle.

Using the nominal values of power supplies and resistance values listed on Figure 3.6.1 the overall loop will have a 2% droop without the reset circuit and 0.2% droop with the reset circuit.



$R_1 = R_2 = 10$ lbs. sec/in^5
$R_3 = R_4 = 10$
$R_5 = R_6 = 10$
$R_7 = R_8 = 10$
$R_9 = R_{10} = 3$

$P_{S1} = 5$ psig
$P_{S2} = 20$
$P_{S3} = 3$
$P_{S4} = 8$
$P_{S5} = 25$
$P_{S6} = 3$
$P_{S7} = 10$
$P_{S8} = 30$

Figure 3.6.1 Overall Speed Loop.

The functions of the various amplifiers are as follows. Amplifier #1 is primarily a buffer amplifier for impedance matching between the reed discriminator pick off and the proportional plus reset loop. It contributes a pressure gain of 2. Amplifier #2 supplies a pressure gain of 5 for the proportional path. The reset gain of 60 is obtained with amplifiers 3, 4, and 5. Amplifiers 6, 7, and 8 comprise the actuator loop. The open loop gain of the actuator stage is 4000 lbs./in^2/in. The closed loop gain is 0.025 in/lbs/in^2 when equal input and feedback resistors are used. The open loop pressure gain from the input of amplifier #1 to the output of #8 is 36000.

The proportional gain or droop of the system is varied by changing the supply pressure to the reed discriminator pick off nozzles. The gain is directly proportional to the supply setting.

The reset loop can be deactivated or activated by closing the variable resistors R_3 and R_4. Removal of the reset loop is accompanied by a 50% increase in proportional gain due to unloading of the proportional gain amplifier.

Starting and stopping of the turbine is accomplished by opening one of two push button valves on the output of the reed discriminator. Opening the valve shunts one side of the error signal to ground and either opens or closes the air valve to the turbine depending on which side is open. The start valve is momentarily opened to bring the turbine to a range of 20 to 50% of rated speed. In this speed range there is sufficient output from the reed discriminator to accelerate the turbine to its set speed.

The scale turbine was sized and designed on the basis of obtaining an adequately fast time constant for test purposes. It was designed for a minimum time constant of 2 sec. for the combined inertia of the speed sensor, electrical generator used for loading and the turbine impellor. Longer turbine time constants are obtained by adding inertia to the shaft. Figure 3. 6. 2 is a photo of the scale turbine and speed sensor. The air supply is brought into a plenum and directed at a high speed radial impellor through four nozzles 1/2 inch by 0. 03 inch. The plenum volume and the piping volume between the air valve and the turbine was selected to simulate a 0. 02 sec. steam valve time constant.

The air valve to the turbine was designed and sized to supply 40 watts to the turbine for a 1/4 inch displacement.

Figures 3. 6. 3 through 3. 6. 7 are response tests on the complete loop with various turbine time constants, reset times and proportional gain settings.

Figure 3. 6. 3 shows the no load response from zero to 10, 000 rpm for a 1. 8 sec. turbine time constant. The lower trace shows speed vs. time for a step displacement of the air valve and is a measure of the turbine time constant. The supply pressure has been reduced to prevent excessive over-speed. The upper trace shows the closed loop response with rated air supply to the turbine. There is a 2% over shoot followed by a small 3 cycles per second oscillation. The overall crossover of the loop is approximately 20 rad/sec and is representative of the maximum crossover with this hardware. This is adequate for any foreseeable turbine- generator application.

Figure 3. 6. 4 shows a zero to full speed response on a 10 sec. turbine with an overall loop crossover of 6 rad/sec. There is no overshoot which indicates a highly damped system. The small purturbations on the speed trace are due to electrical noise from the tachometer used to monitor speed.

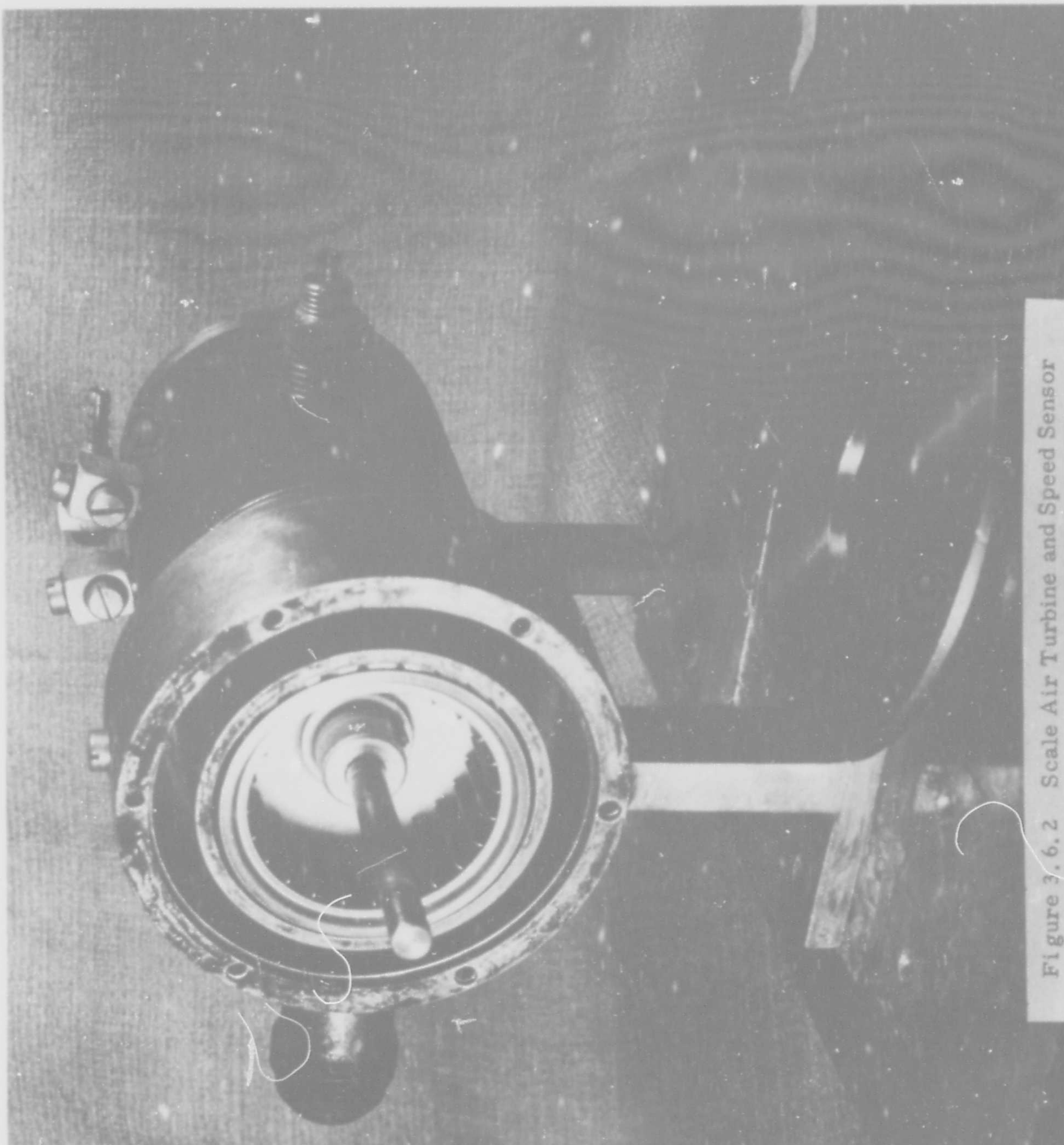


Figure 3.6.2 Scale Air Turbine and Speed Sensor

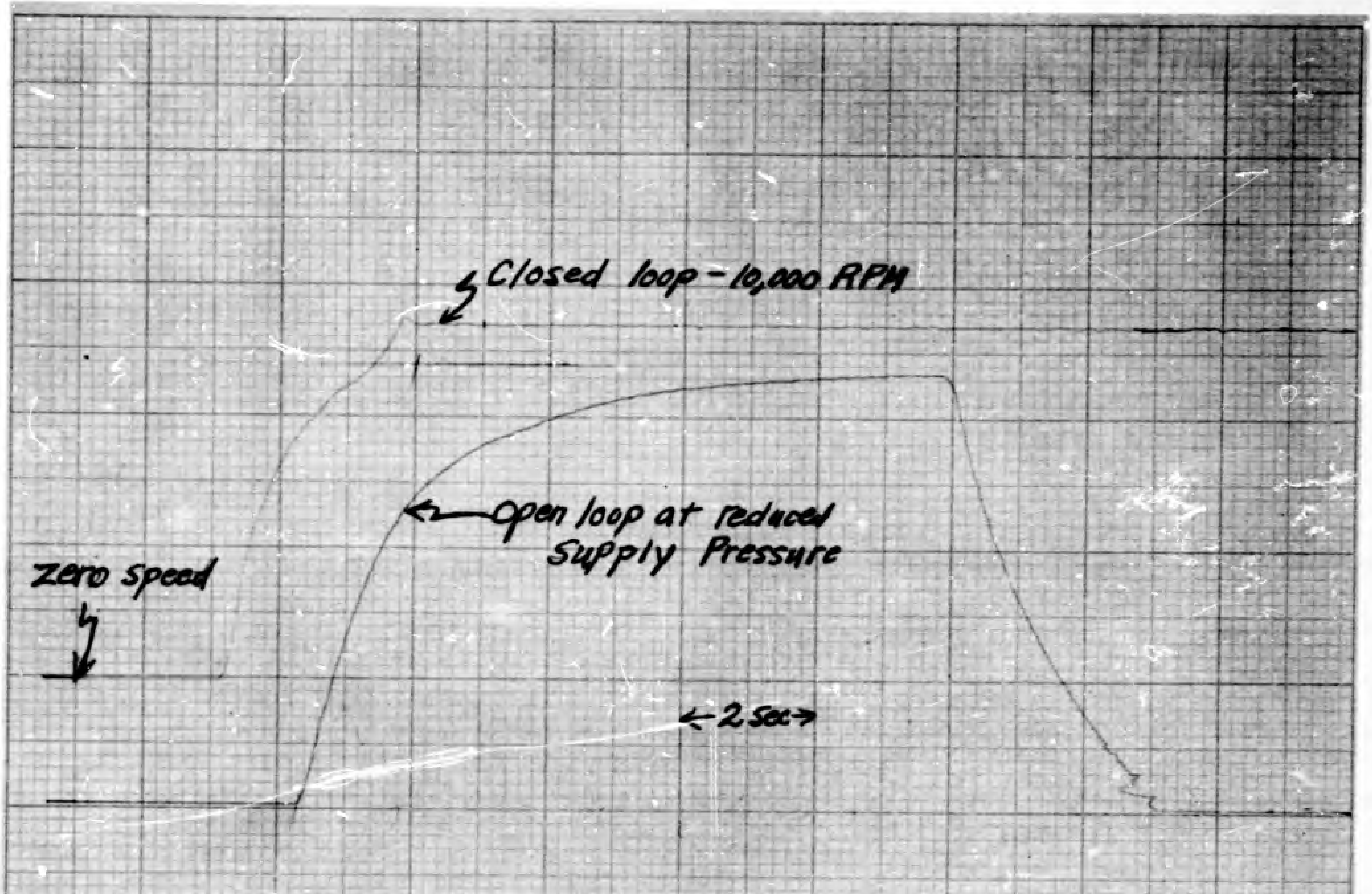


Figure 3. 6. 3 Response of 1. 6 sec. Turbine from Zero to Full Speed.

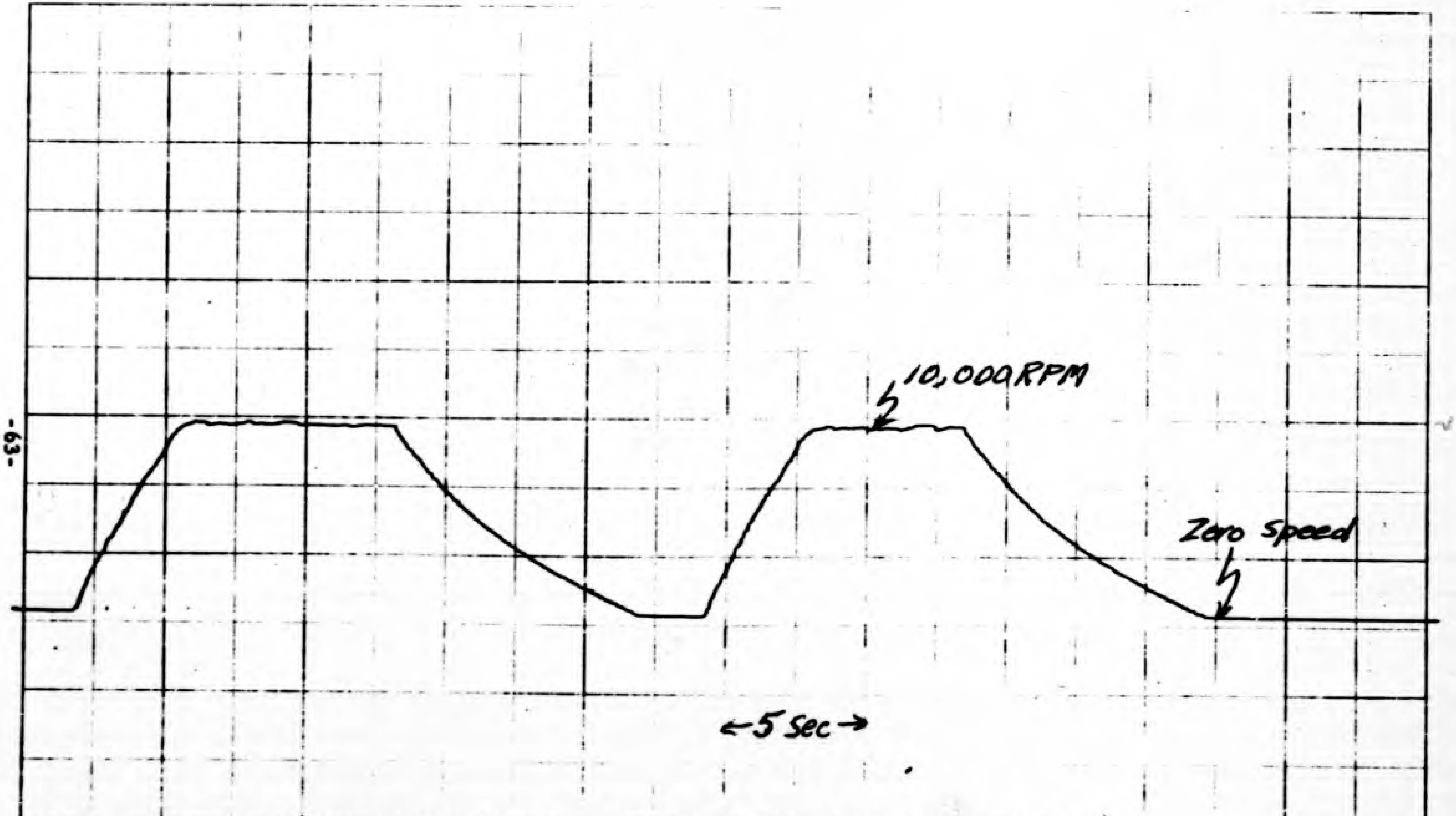


Figure 3. 6. 4 Response of 10 Sec. Turbine from Zero to Full Speed.

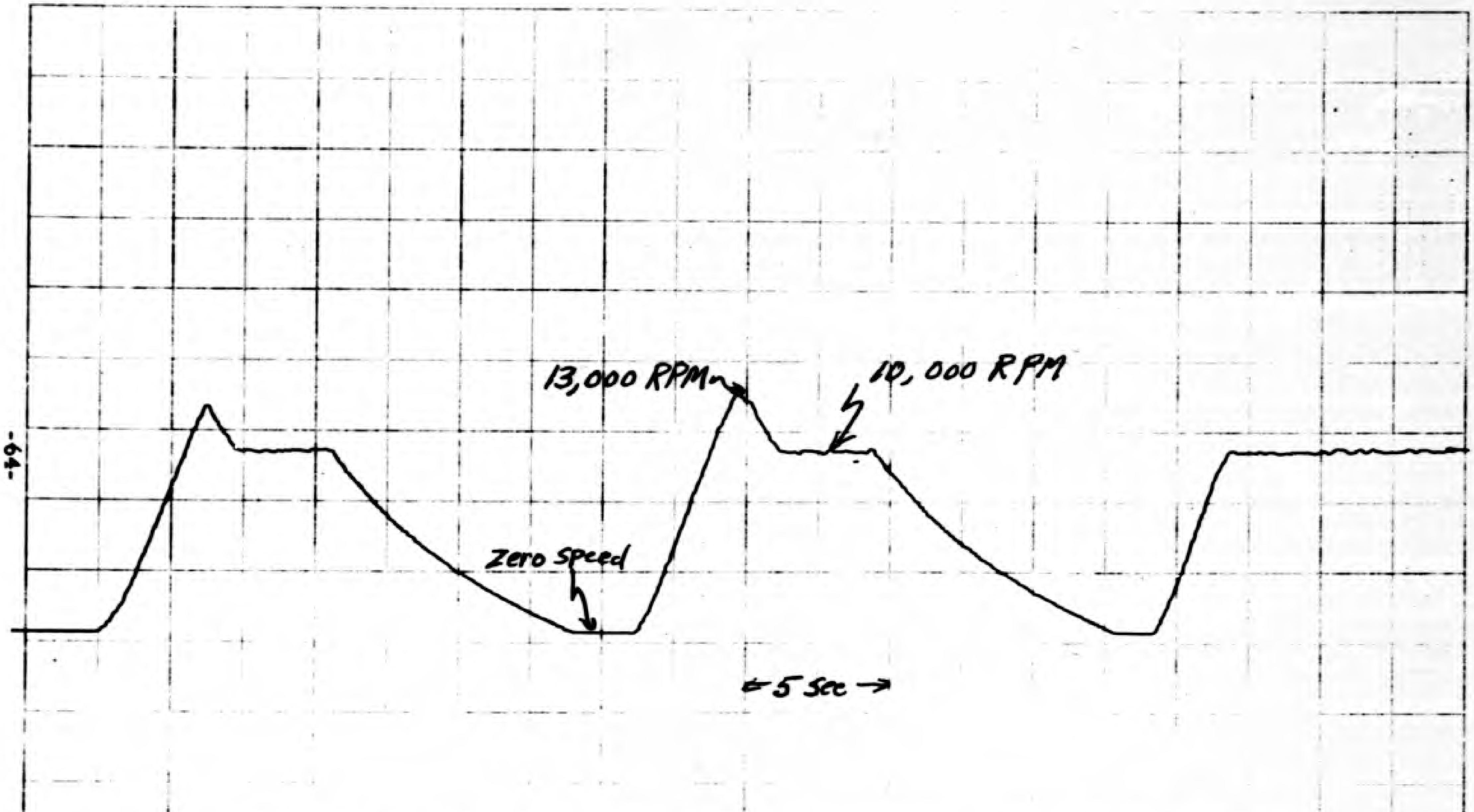


Figure 3.6.5 Response of 10 sec. Turbine to Overspeed. Prop. Plus Reset Control.

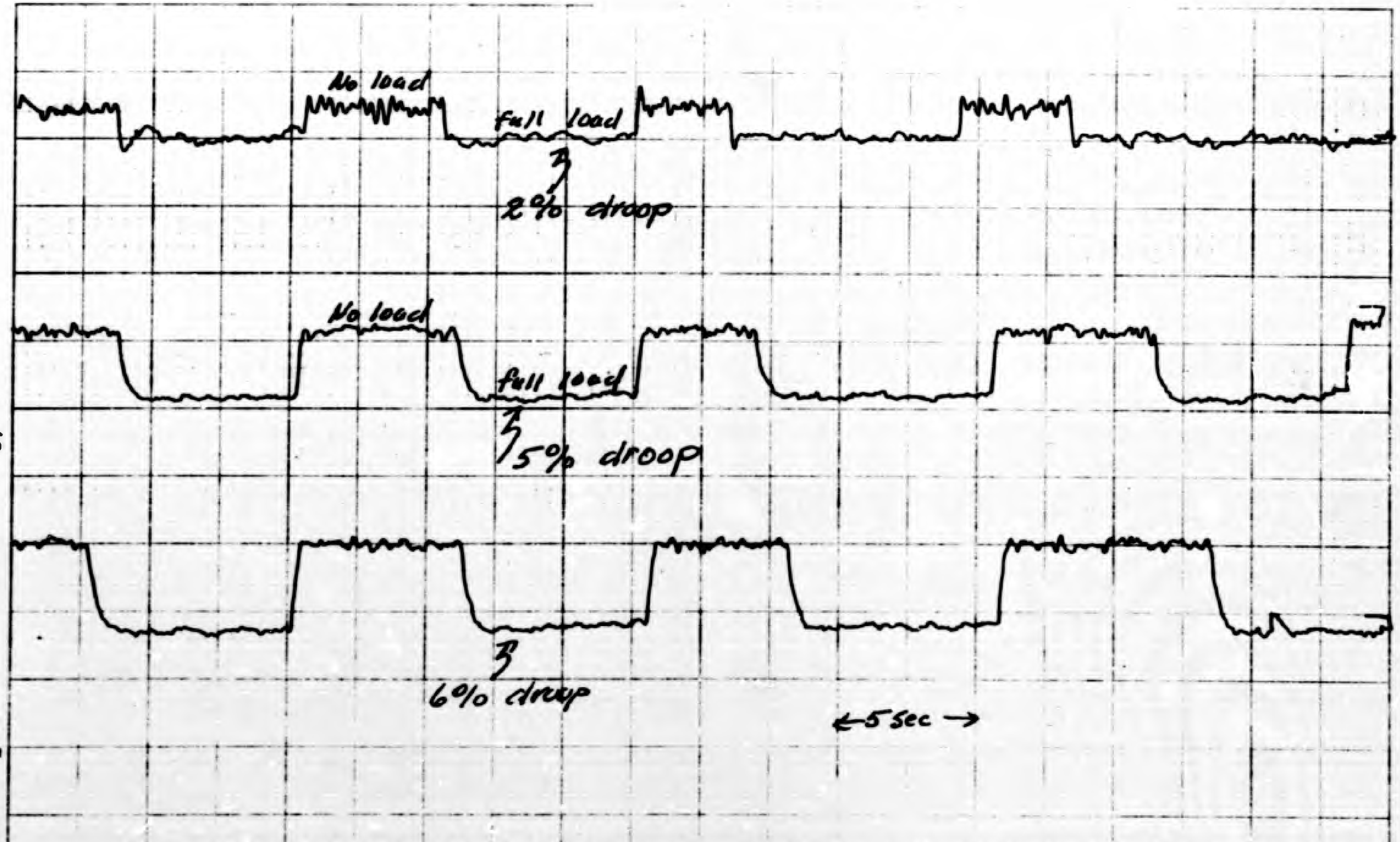


Figure 3. 6. 6 Response of 10 sec. Turbine to Jump Load. Proportional Gain Varied - No Reset.

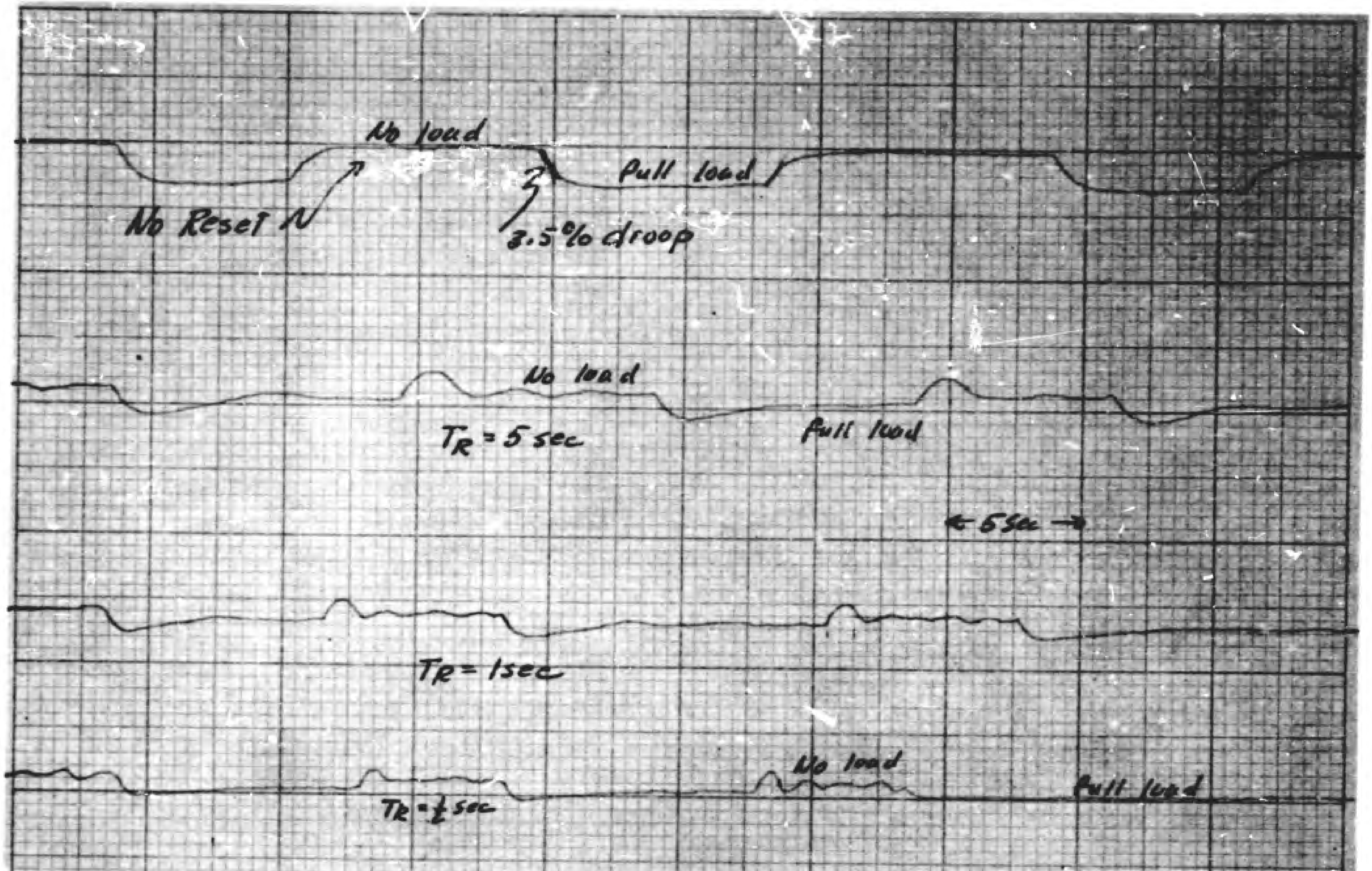


Figure 3. 6. 7 Response of 10 sec. Turbine to Jump Load. Reset Time Constant Varied.

Total time to go from zero to set speed of 10,000 rpm is 5 sec.

Figure 3.6.5 was taken on a 10 sec. turbine and demonstrates the capability of the control to recover from overspeed conditions. The tests were run by holding the start button until the turbine exceeded its set speed. The button was then released and the speed control loop allowed to take over. The control will recover from overspeeds of 25%. This range encompasses the range between the normal set speed and the overspeed trip, hence there is no need for any auxiliary circuitry on the primary speed control.

Figure 3.6.6 illustrates the change in regulation when the supply pressure to the reed discriminator pick offs is varied. The high frequency purturbation are primarily due to recorder and tachometer noise.

Figure 3.6.7 shows the effects of varying the reset time constant with the proportional gain held constant. The upper trace is with no reset while the lower traces are for various reset time constants. With a 1 sec. reset time the system has settled to essentially its steady state value in 1.5 sec. and the maximum speed deviation is 1.75%. The system will meet the transient requirements of MIL-G-21410, however, the steady state droop characteristics are marginal. The true gain ratio between the proportional and reset gain paths should be 1.5 times the measured because of the loading effects on the proportional path when the reset circuit is added. The measured droop with reset is 0.5%. While without reset it is 3.5%. The gain ratio is then $\frac{3.5}{.5}(1.5) = 10.5$. This agrees with independent pressure gain measurements on the proportional plus reset loop and agrees with the original design goal. It is evident that a larger ratio should be selected to give more flexibility to the system. With the present system 0.2% steady state droop can be achieved only by increasing the proportional gain and the system crossover.

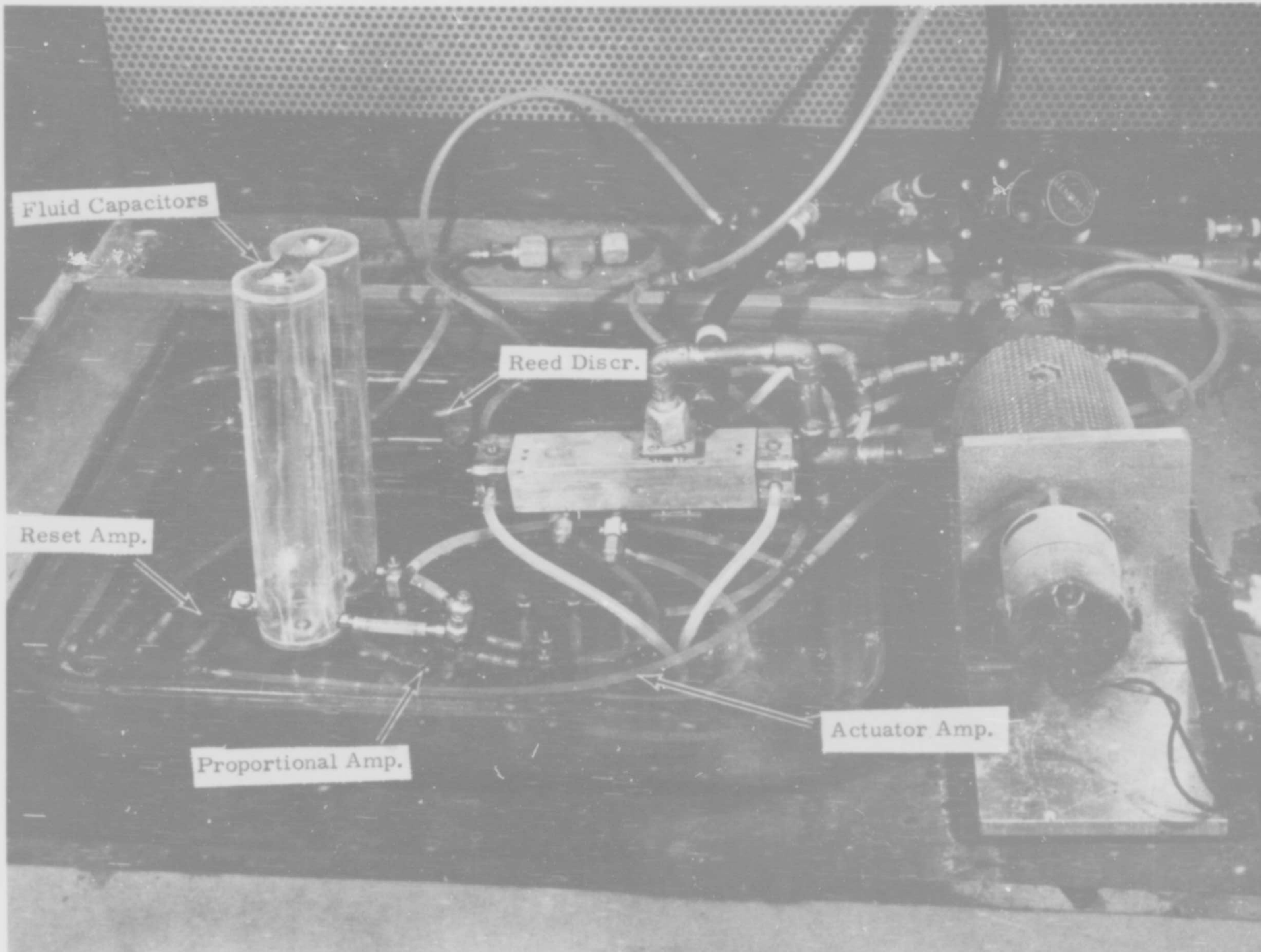


Figure 3.6.8 Assembled Speed Control Loop

Figure 3.6.8 is a photo of the assembled speed control loop. No attempt was made on this contract to make an integrated package design. The primary object was to retain flexibility in selection of supply pressure levels, summing resistors and addition or deletion of amplifier stages as needed. Individual amplifiers or blocks of amplifiers were interconnected with tubing and all summing resistors and dropping resistors for power supplies are adjustable needle valves. Operation with water requires that the amplifiers be completely immersed in water to avoid entrapment of air bubbles. An open pan with sufficient depth to cover the amplifier and reed discriminator is used for the feasibility model.

A control designed for installation on a full scale turbine generator set would use stainless steel for all the component parts. The amplifiers would be functionally grouped so that all the fluid amplifiers would be contained in two circuit boards approximately 1/4" by 2" by 7". Power supply manifolds, power supply dropping orifices and fixed summing orifices would be an integral part of the circuit board. The two circuit boards would be stacked and enclosed in a common manifold. The external connection to the manifold would be two input lines from the reed discriminator, two input lines from the actuator position pickoff, two output lines to the actuator, one supply line for the amplifier power supplies and a drain line. The drain would be restricted so that the manifold operates at a small positive pressure relative to the return line. The least tenable portion of the design are the fluid capacitors required for the reset loop.

3.7 Conclusions

1. The response time of fluid amplifiers operating on water is fast enough for any foreseeable turbine-generator speed control application.

2. The mechanization of a closed loop actuator stage with sufficient loop gain to cope with the negative steam reactions gradients and valve cracking forces of typical poppet type steam valves is feasible. The fluid circuitry required is relatively simple and can be readily extended to an oil operated actuator.

3. The mechanization of the reed discriminator with the reeds fully immersed in water is feasible. Adequate bandwidth and gain can be obtained with the proper reed design.

4. The signal to noise ratio of fluid amplifiers operating on water is marginal. The low frequency components of noise, inside the bandwidth of the speed control loop, cause random peak to peak fluctuations in turbine speed of 0.2%.

   A Company funded noise program identified the primary sources of noise. Corrective design modifications were recommended, however, relative timing between the noise program and the hardware development did not allow for incorporation of the recommended changes in the final hardware. Results of this program were sufficiently encouraging to predict that self generated noise will not set a limitation to overall performance on future designs.

5. Water operated amplifiers are susceptable to shock and vibration. The effects of shock and vibration can be confined to the input stages by paralleling two push-pull amplifier stages. The input connections to the amplifier are made so as to cancel vibration and shock induced noise at the summed outputs of the parallel stages.

6. Mechanization of a "no-moving-parts" reset circuit without a water to air interface is beyond the present "state of the art". Low signal to noise ratio and rigid balancing requirements on the fluid amplifiers limited the reset time constant to 1/4 sec. on the circuits evaluated on this contract. The basic concepts are fundamentally sound and show promise for long range applications.

7. Mechanization of the reset circuit using a pneumatic capacitor is relatively straight forward. There are no apparent limitations to the reset times that can be obtained. The approach is unattractive, compared to the time constant multiplication circuits, because of the introduction of a water to pneumatic interface.

   The specific reset circuit mechanized on this contract has marginal steady state gain. Future designs will require an additional stage of gain and longer reset times.

4.0 STEAM TESTS

The objective of this phase of the contract was to determine the feasibility of using steam as the working fluid in a fluid amplifier control system. The effort was primarily directed at determining the effects of erosion and the degree of erosion vs. operating time and steam conditions. A secondary task involved an analytical study to determine whether fluid amplifier circuits could be used in a steam-operated actuator stage for a steam valve. This task was concerned with the dynamic response capabilities of the amplifiers. The practical problems involved in the design and fabrication of a reliable actuator for steam operation were beyond the intended scope of the study.

4.1 Materials and Design Study

Fluid amplifiers will be subjected to steam conditions similar to those encountered by steam turbine buckets. The pressure ratio which would normally be used across the output amplifier and some of the intermediate amplifiers in a control loop will give steam velocities in excess of the critical erosion velocities of some materials. In a typical fluid amplifier the spacing between the jet nozzle and receivers will be sufficiently large to permit small water particles or solid particles to reach a terminal velocity approaching the velocity of the stream. The conditions required to cause erosion are then present, particularly at the fluid amplifier receivers.

Considerable data is available on the erosion properties of materials commonly used in steam turbine buckets and to a lesser degree on materials considered potential candidates for turbine buckets. Data has been accumulated from inspection and measurements on turbines with years of field service.

This data has been supplemented by laboratory life tests on samples under controlled and known conditions. Empirical equations relating rate of erosion to the material, steam velocity and quality have been derived. These equations can be used with a high degree of confidence to predict erosion rates under specific conditions.

The primary unknowns in the application to fluid amplifiers are the effects of geometry on rate of erosion and the changes in element performance due to finite amounts of erosion.

The materials selected were based on a review of data accumulated by the General Electric Large Steam Turbine & Generator Department and consultation with their experts. Of primary concern was selection of materials, for the elements to be life tested, which would yield the maximum information in the 12 weeks allocated for the test. Two materials with a large and well-established spread in erosion characteristics were selected. A chrome steel was selected for one end of the scale and carbon steel for the other. The two materials have approximately a 40: 1 difference in erosion rate under identical steam conditions. Preliminary predictions of erosion indicated that carbon steel would exhibit significant and measurable erosion over a 12 weeks period. This would give a quantitative measure of erosion vs. time from which the useful life of amplifiers, fabricated of other materials and operated under different steam conditions, can be predicted. One other material with unknown characteristics was selected primarily because of its potential for lowering fabrication costs. A relatively new process has been introduced which permits deposition of titanium carbide on a base metal by the flame spray process. Titanium carbide has the general characteristics required to make it a potential candidate for use in highly erosive fluids. The deposition can be made on base metals more easily fabricated than the chrome steels.

The design study was directed at minimizing the effects of erosion on the element characteristics. The most likely spot for erosion to occur is at the tips of the receiver entrance. Normally these tips are made relatively sharp to permit location of the receiver entrance at the maximum gain point on the jet. The basic supposition was that the tips would erode and the gain of the element would degrade with time. Elements which simulated various degrees of erosion were made and tested. The tests indicated that the initial rate of change of gain vs. time would be fairly rapid but would decrease with time. This suggested that the tips should be given a generous radius to decrease the initial rate of change of gain vs. time. The primary design effort was directed at recovery of the gain loss due to radiusing the tips. The parameters varied were the angular spacing between the receiver center line axis and the spacing between the receiver entrance and the jet nozzle. Best results were obtained with the angular spacing maintained at the optimum value for a sharp tip design and increasing the receiver to nozzle spacing.

The initial gain of the amplifier has been reduced by 30% by the 0.02" radius selected for the final design.

4.2 Selection of Test Conditions and Test Samples

The life test conditions were selected to give accelerated erosion on the elements. The critical velocities (velocity below which no significant erosion occurs) and the relationships between moisture content and erosion are well-established on the chrome steel and carbon steels selected for test samples. A steam velocity of 1000 ft/sec and a 6% moisture content were selected for the test.

In normal usage the fluid amplifiers will be supplying loads ranging from wide open to fully blocked. The load will influence the steam velocity at the receiver tips. Test conditions were selected to simulate the two extremes in loading.

Originally the intended scope of the life test program was to run the test on only wet steam conditions to accelerate erosion. However, past experience has shown that contamination from water soluble products in the boiler feed water is aggravated under dry steam conditions and both wet and dry steam tests seemed desirable. It was also desired to evaluate design changes made to minimize the effects of erosion by testing designs with both sharp tips and rounded tips.

Thirteen test samples were selected to obtain sufficient data to relate cause and effect in the variables involved.

The selected conditions of test and a listing of the samples tested are tabulated in Figure 4.2.1 .

The following criteria were selected for evaluation of the test results.

1) Gain and overall performance. Gain would be checked at two power supply settings and two loading conditions.

2) Measurement of the critical dimensions such as nozzle to receiver spacing, nozzle widths and control port geometry.

3) Microscopic inspection of the elements.

4.3 Design of the Life Test Fixture

Two fixtures are required to test under both wet and dry steam conditions. To obtain any useful results the conditions of the test must be known and controlled. The prime purpose of the test fixture is to accept steam from the supply mains under widely varying conditions, convert the steam to constant and controlled conditions and apply it to the test elements. Sufficient monitoring capability should be included to detect catastrophic failure or very rapid degradation of performance of the test elements.

Tabulation of Test Samples

Sample #	Material	Geometry	Load	Steam Condition
1	Carbon	ST	B	Dry
2	Carbon	BT	B	Wet
3	Chrome	BT	B	Wet
4	Carbon	ST	B	Wet
5	Chrome	ST	B	Dry
6	Carbon	ST	B	Wet
7	Carbon	BT	O	Wet
8	Chrome	BT	O	Dry
9	Chrome	ST	O	Wet
10	Chrome	BT	O	Wet
11	Chrome	ST	O	Wet
12	Tit. Carbide	ST	O	Wet
13	Tit. Carbide	ST	O	Wet

Nomenclature: ST - Sharp tip
BT - Blunt tip
O - Short circuit load
B - Blocked load

Test Conditions:

Wet steam:

1. Steam velocity - 1000 ft/sec.
2. Quality - 6%
3. Temperature - 425°F

Dry steam:

1. Steam velocity - 1000 ft/sec.
2. Temperature - 450 to 700°F

Figure 4. 2. 1

The facilities of the Large Steam Turbine and Generator Departments testing laboratory were required to conduct the steam tests. The steam supply is at a nominal pressure of 400 psig. Depending on the laboratory test load the steam quality will range from saturated to 300°F superheat. These inlet conditions require the test fixture to have a means of injecting water into the steam to the test samples. The widely varying conditions dictate that the water injection be controlled by a closed loop system. The implementation of the control loop requires a sensor for measuring steam quality, comparison against a reference, amplification of the error signal, generation of a derivative signal for stabilizing the system and finally the control of a water injection valve.

A schematic diagram of the fixture is shown in Figure 4.3.1. Regulator # 1 supplies steam at 360 psig to both the wet and dry steam fixtures. Water for injection into the wet steam fixture is obtained by a condenser at the output of Regulator # 1. The condensed water is forced through the controlled metering valve and the mixing nozzle by the pressure drop established by Regulator #2. The wet steam from the mixing chamber is piped to the nozzle plenums of the individual test amplifiers. Separate lines are used for each amplifier to assure equal division of load between all samples. The vents from the amplifier exhaust into a common manifold. The output from the manifold goes through a throttling valve and then to a throttling Calorimeter. A gas bulb thermometer senses the temperature in the Calorimeter and compares it against a set temperature. The temperature error is amplified in a pneumatic controller which positions the water metering valve so as to maintain the steam temperature in the Calorimeter equal to the set temperature. The set temperature is determined by entering the mollier steam chart with the pressure in the wet steam mixing chamber (output of Reg. #2), and the desired steam quality. The set temperature is then found from the intersection of the constant enthalpy line and the measured pressure in the throttling Calorimeter.

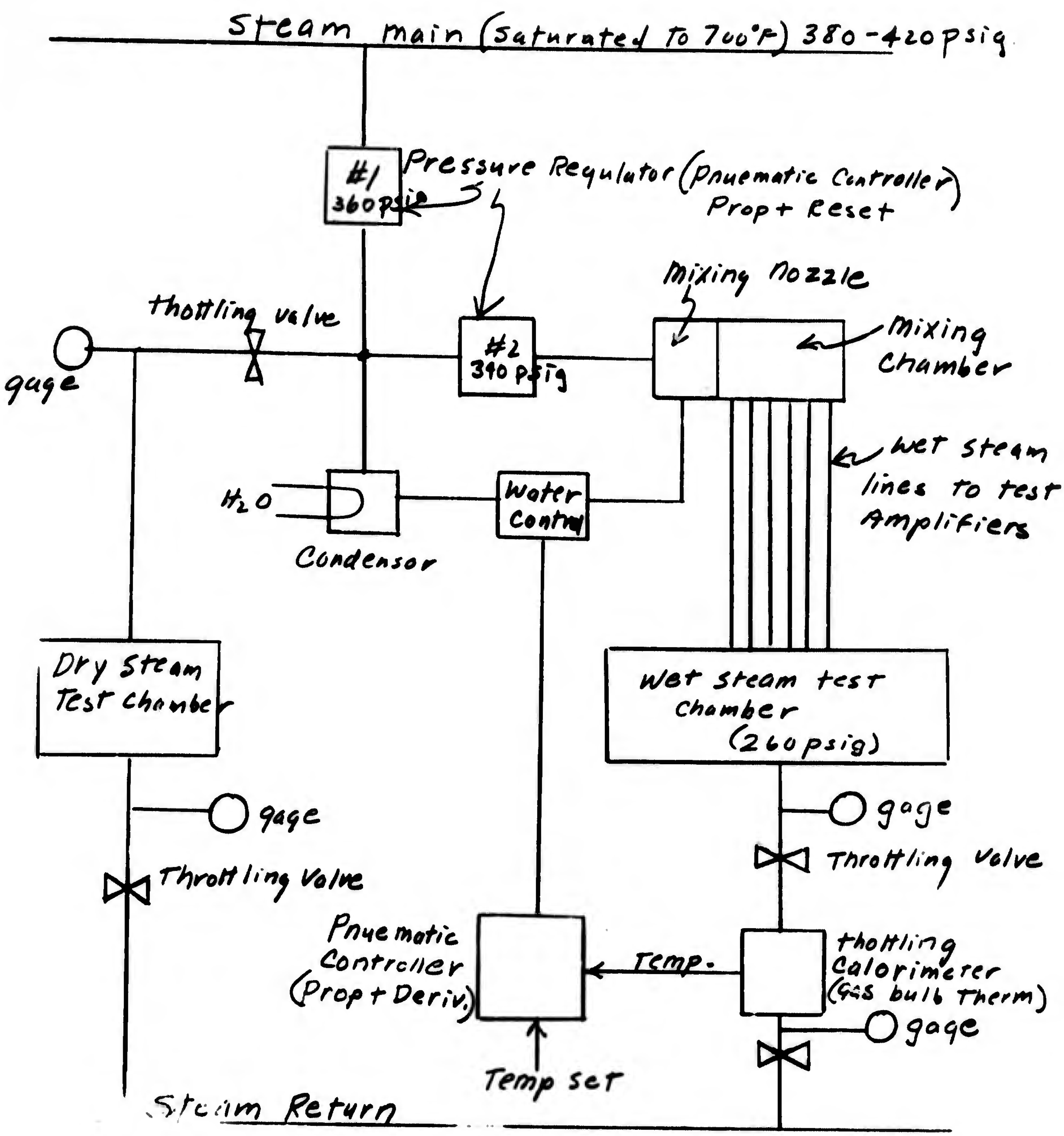


Figure 4.3.1 Steam Test Fixture.

The steam velocity is set by adjusting the throttling valve between the test manifold and the Calorimeter to give the desired nozzle drop across the test sample.

The wet steam fixture was designed to accommodate 12 test samples. Six of the test stations have pressure gages for monitoring the recovery pressure in the receivers of the test sample. These stations would normally be used for blocked load tests on the elements and are primarily used to detect catastrophic types of failure.

The dry steam fixture was designed to accommodate three test samples. Steam is supplied to the test manifold through a fixed throttling valve. The amplifiers exhaust to the common manifold. The manifold back pressure is adjusted to give the desired nozzle drop across the amplifiers. The pressure drop across the throttling valve is high enough to insure dry steam to the test chamber under all inlet conditions.

Figure 4.3.2 shows the test fixture before application of the lagging material to the steam lines and test manifolds. The primary components are identified on the photo. Figure 4.3.3 shows the completed fixture in place at the Large Steam Turbine and Generator Department testing lab.

4.4 Steam Test Results

Initial gain checks, measurements of critical dimensions and photographs were made of the thirteen elements selected for life test. The test was started on January 19, 1965 and terminated on April 12, 1965 after 2080 hours of operating time on the amplifiers. The test was continuous except for a 3 day shutdown after 840 hours for routine examination of the test amplifiers.

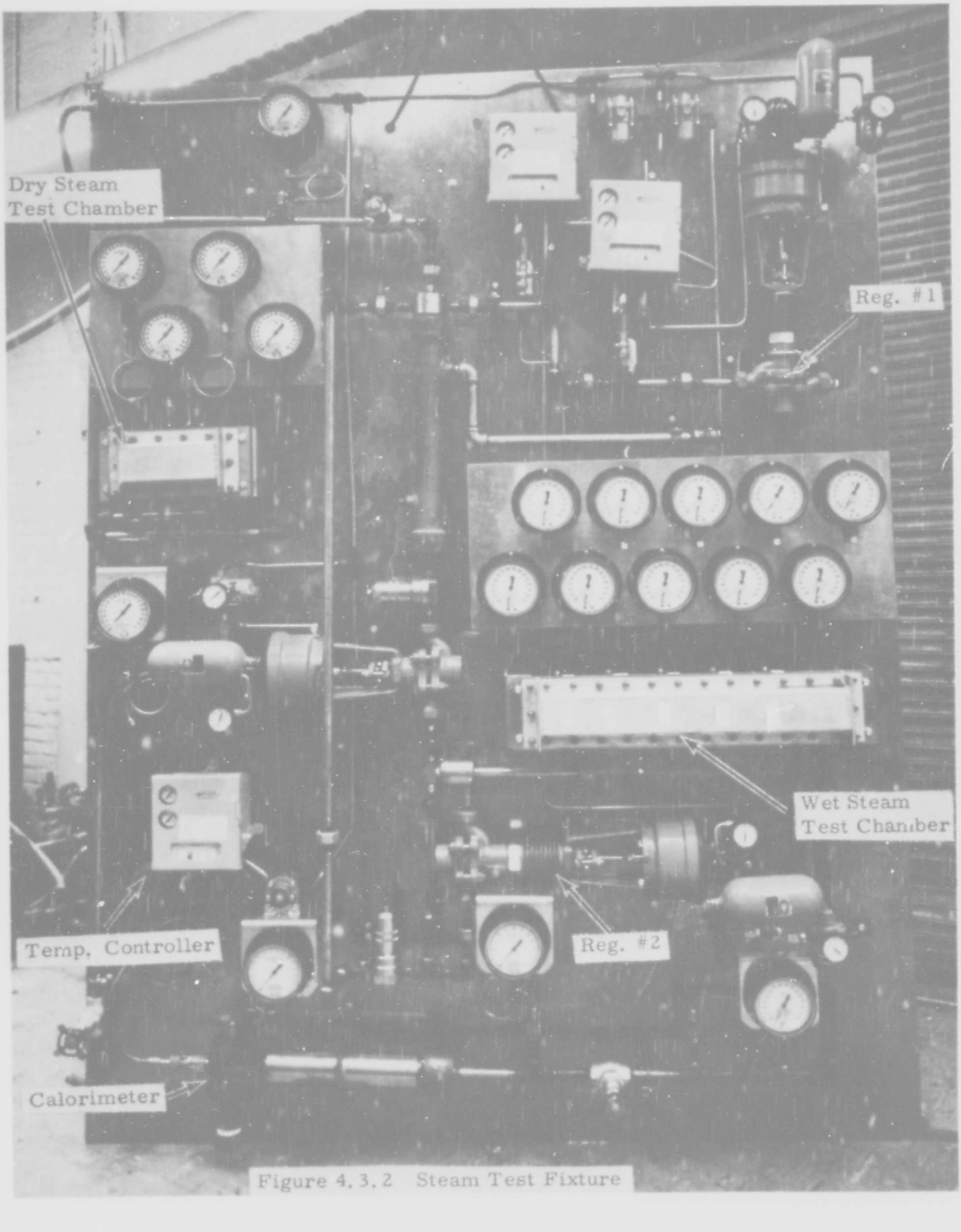


Figure 4.3.2 Steam Test Fixture

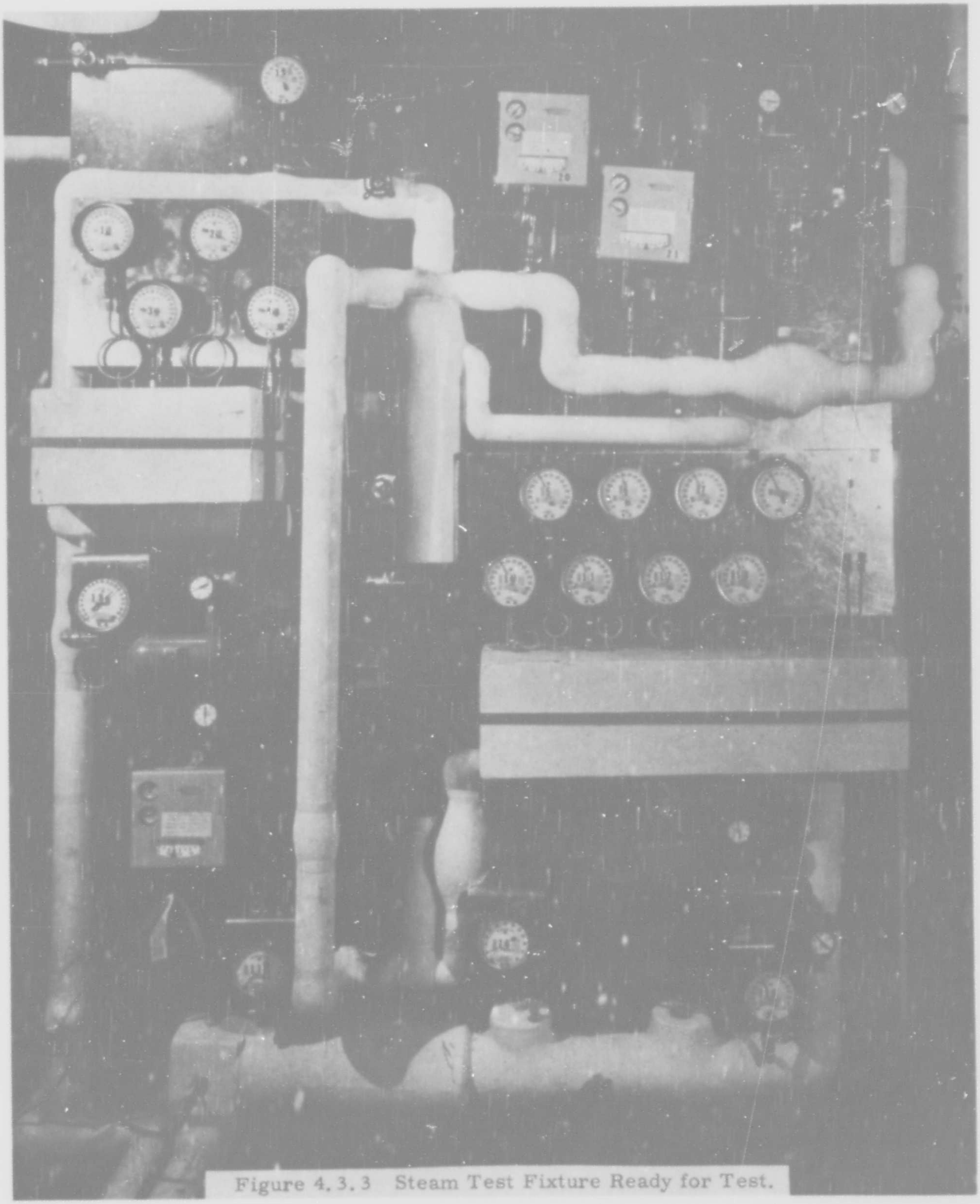


Figure 4.3.3 Steam Test Fixture Ready for Test.

Photographs of representative elements before life test are shown in Figures 4. 4. 1 to 4. 4. 6. A tabulation of the initial gain and significant dimensions is given in Figure 4. 4. 7. The gains and DC offsets were measured with both 5 and 10 psig power supplies and under block load and a 0. 04" diameter orifice load. The tabulated gain is the slope of the output vs. input curve measured at the point where the differential output goes through zero. The DC offset is the differential output with zero differential input.

A tabulation of the changes in dimensions and changes in gain and DC offsets after the life test is presented in Figure 4. 4. 8. Figures 4. 4. 9 to 4. 4. 14 are photographs of representative elements taken after completion of the life test.

4. 5 Interpretation of Results and Conclusions

The gain changes resulting from the life test were significant and not consistent with the expected trend. An acceptable gain variation from the standpoint of a system application will encompass a wide range depending on the system requirements. The speed control loop mechanized on this contract can be made relatively tolerant of gain variations in the amplifiers providing the trend is known and the minimum gain criteria can always be met. Gain variations up to 20% are tolerable. In contrast the less complex speed control loop using an open loop actuator stage for integration requires gain stabilities in the order of 5%. Using 20% as an acceptable variation, eleven of the thirteen elements failed. The only two amplifiers with acceptable gain variations were # 12 and # 13. These were the amplifiers with a deposited coating of titanium carbide.

Several general trends can be noted. After life test the gain has increased on the elements which had low initial gain and decreased on the

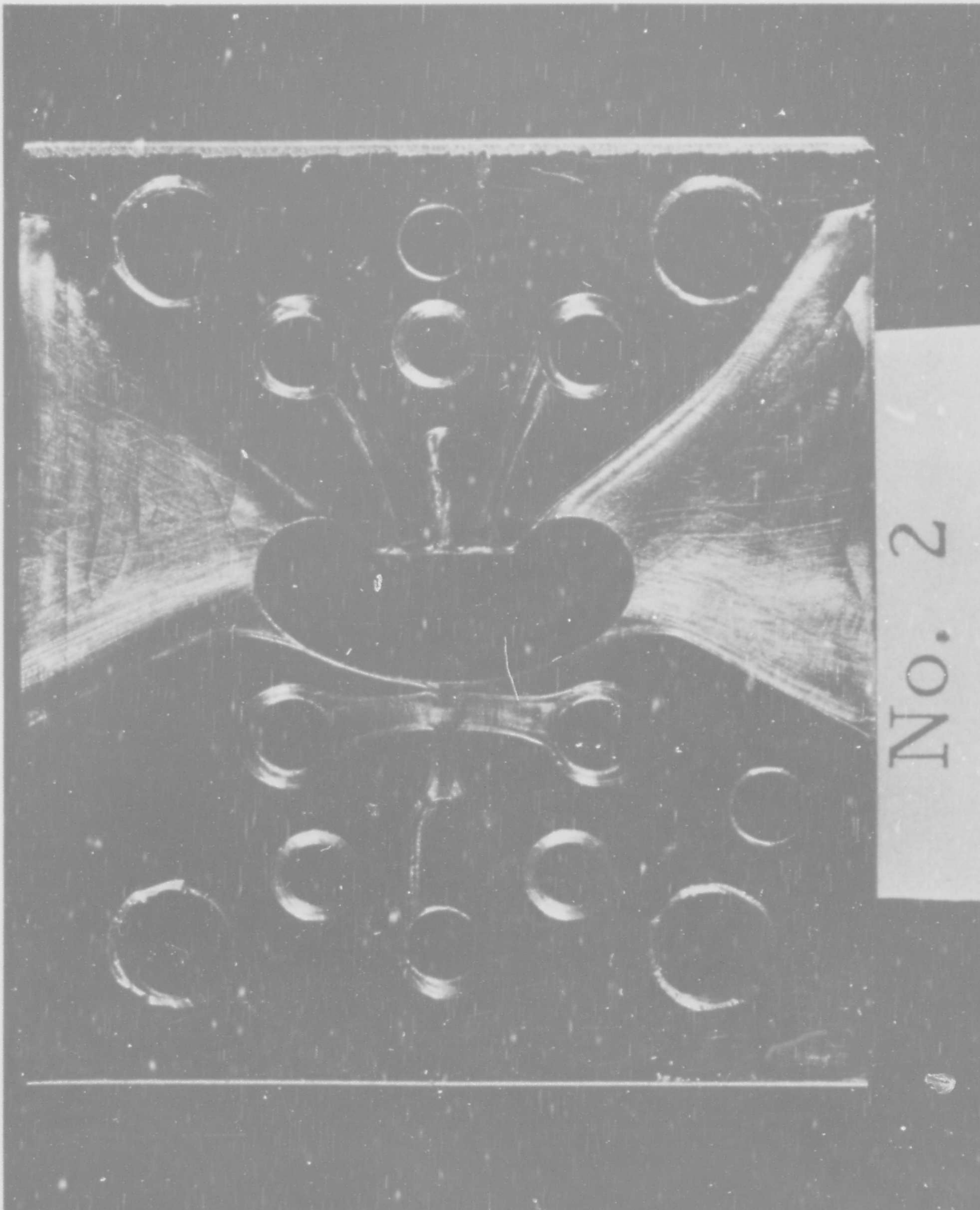


Figure 4.4.1 Sample #2 - Carbon Steel BT.

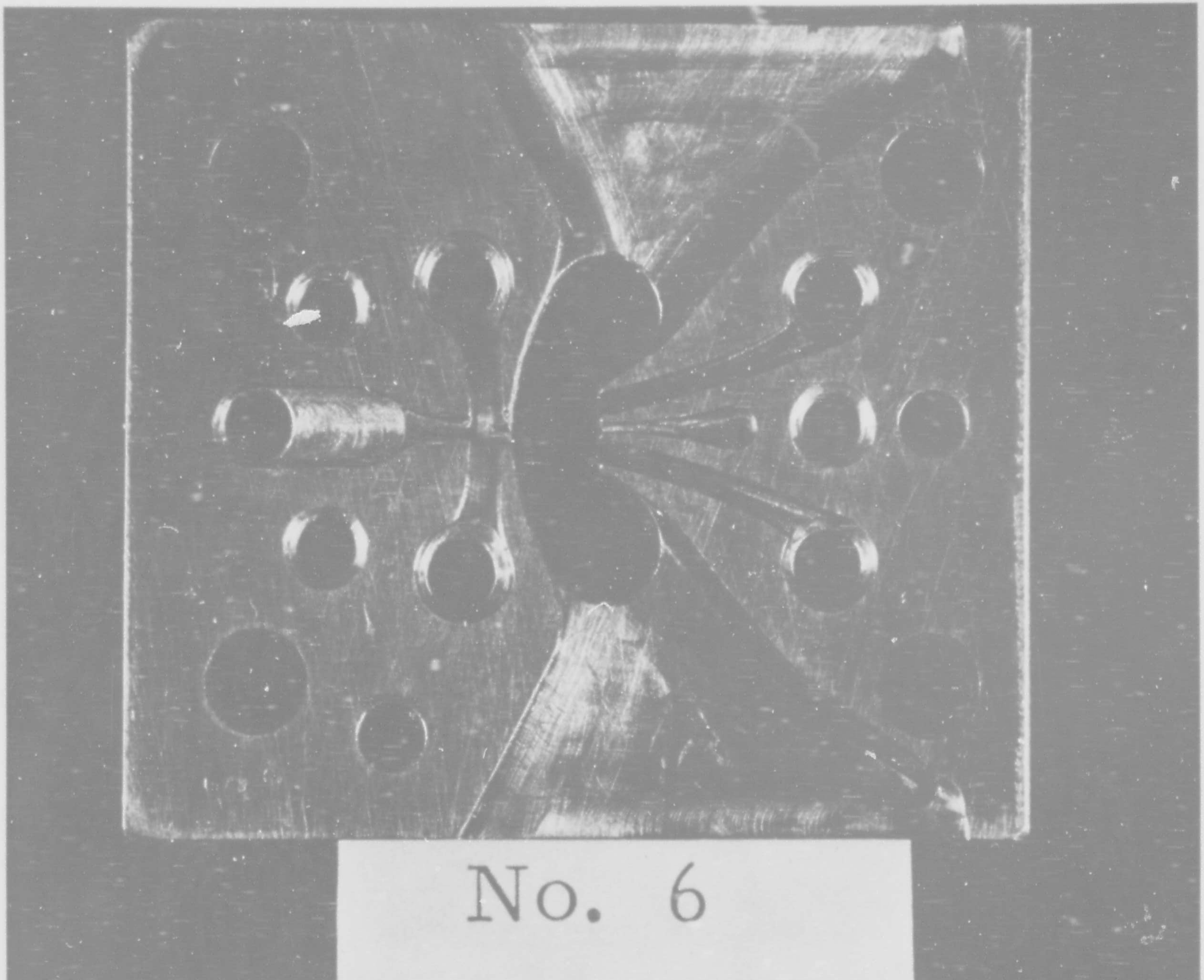


Figure 4.4.2 Sample #6 - Carbon Steel ST.

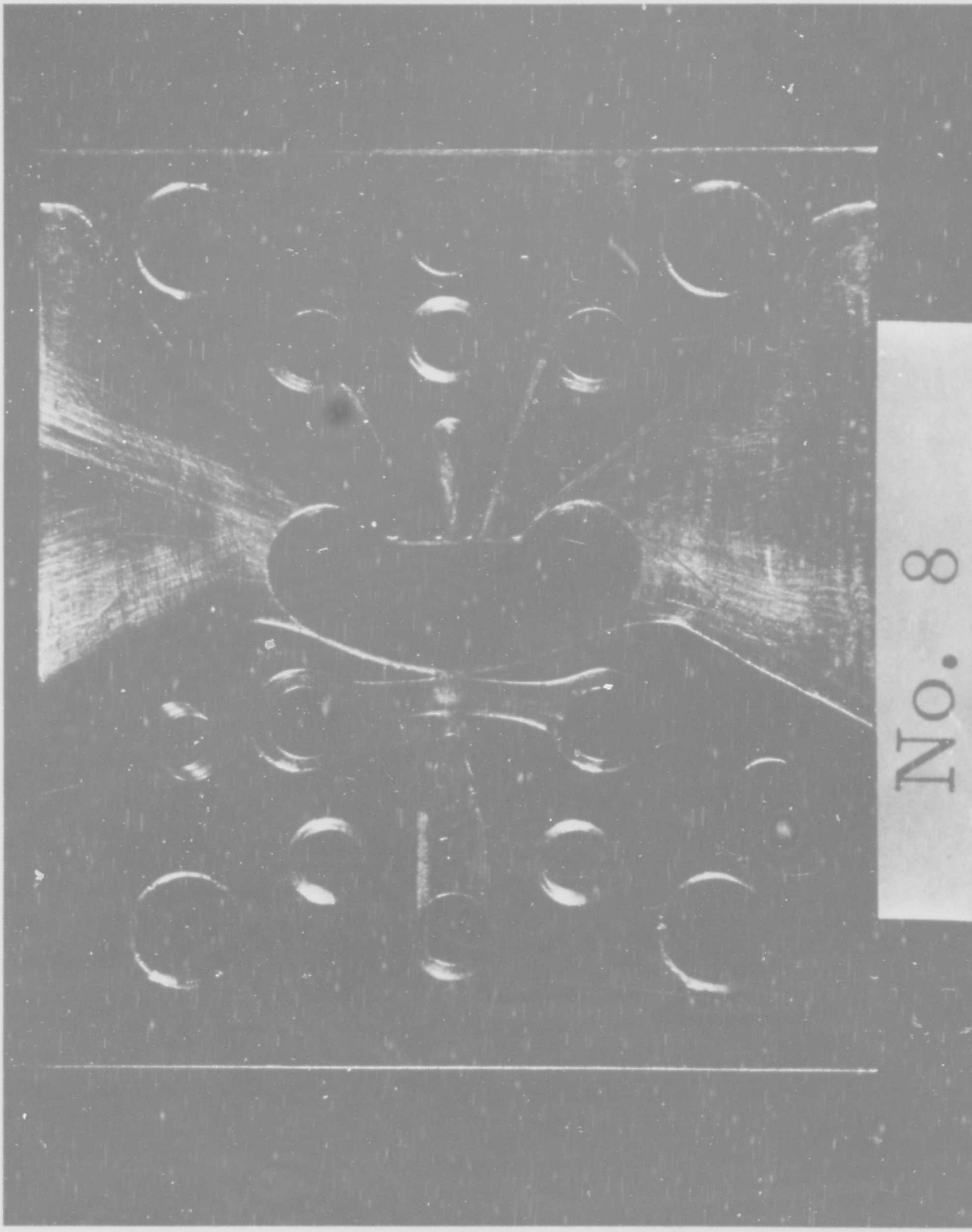
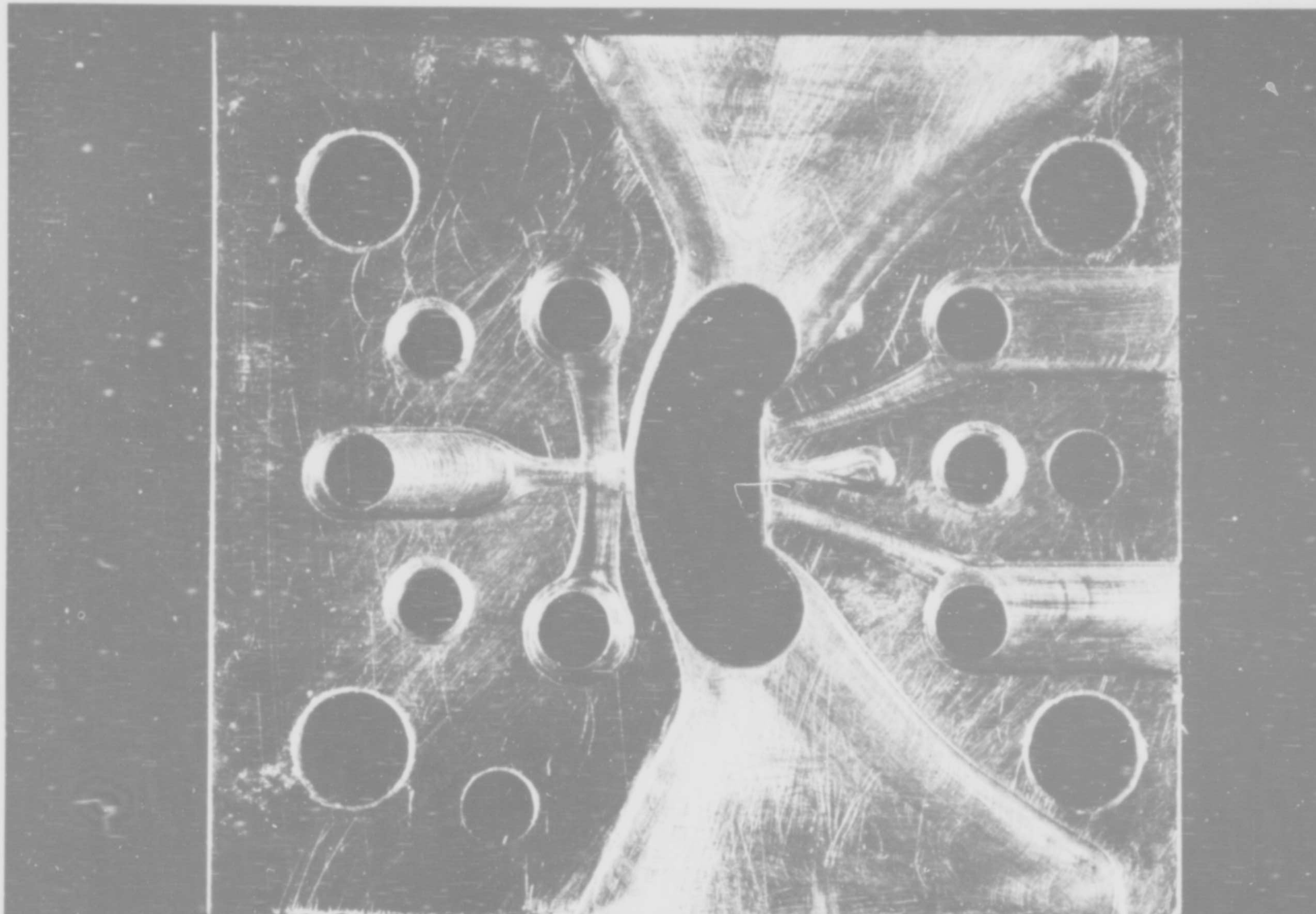


Figure 4.4.3 Sample #8 - Chrome Steel BT.



No. 10

Figure 4.4.4 Sample #10 - Chrome Steel BT.

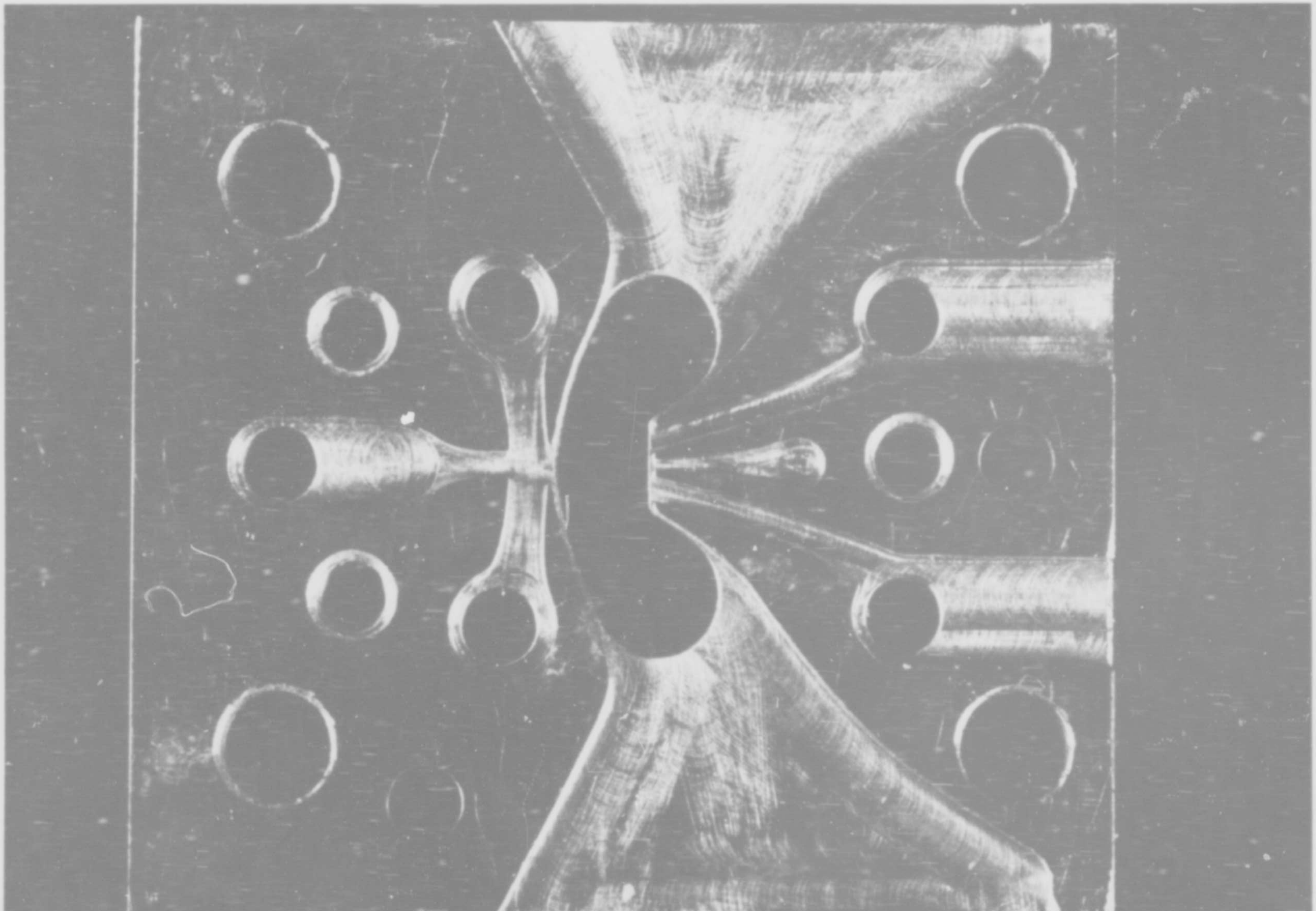


Figure 4.4.5 Sample #11 - Chrome Steel ST.

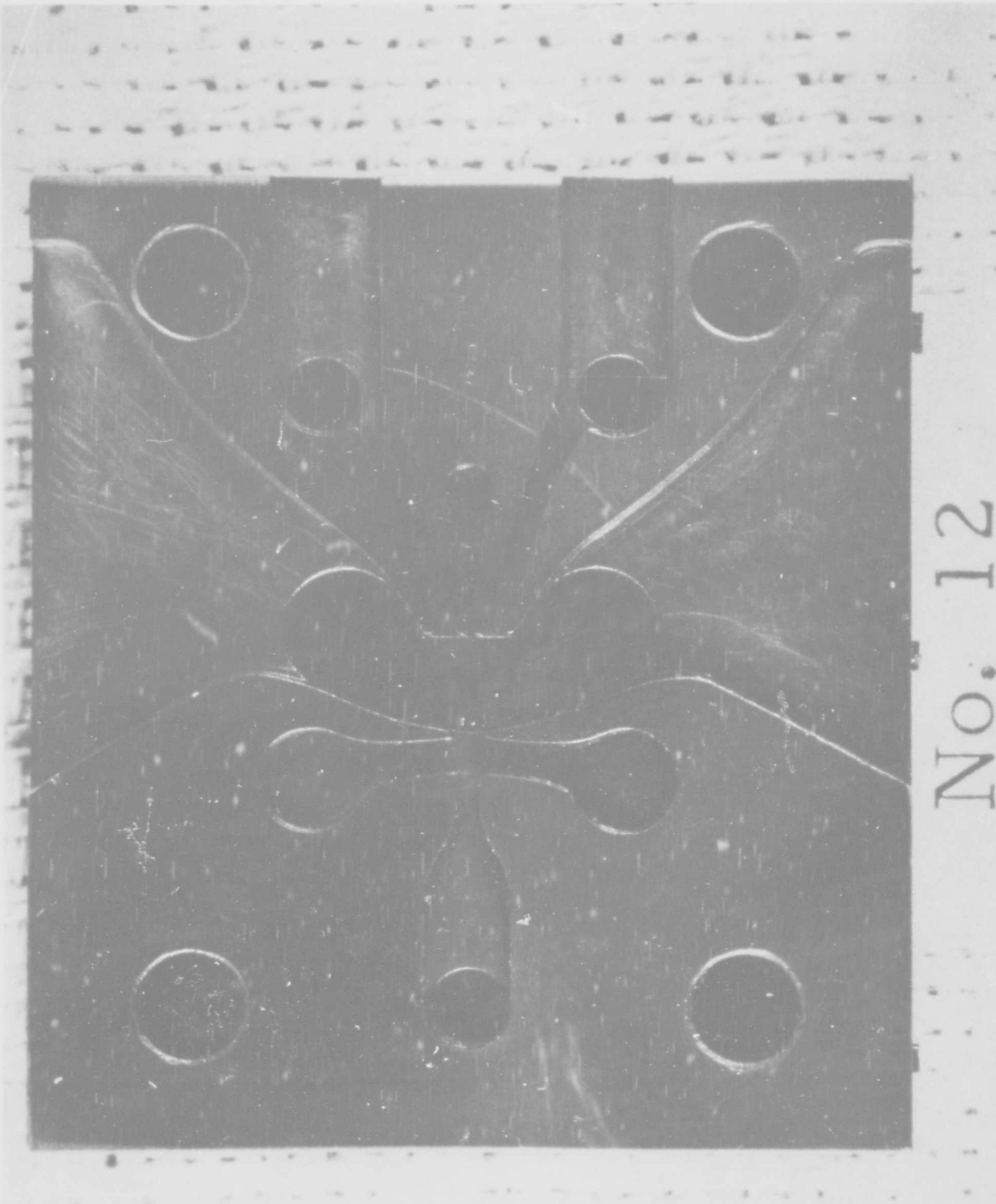


Figure 4.4.6 Sample #12 - Titanium Carbide on Tool Steel ST.

Initial Dimensions and Gain

Amp. #	Dimensions (in)				Gain (G) and Offset (O)							
					Blocked Load				.04" Orifice			
					5 psig		10 psig		5 psig		10 psig	
	A	B	C	D	G	O*	G	O	G	O	G	O
1	.2451	.2442	.0429	.0649	5.0	+.5	6.2	+1.2	2.36	+.05	2.65	+.1
2	.3206	.3206	.0418	.0633	4.2	+.2	4.0	+ .4	2.24	0	2.23	0
3	.3206	.3203	.0428	.0633	3.4	0	4.46	0	2.0	-.05	2.24	0
4	.2447	.2453	.0431	.0648	5.0	+.05	6.67	+1.6	2.5	0	2.87	+.30
5	.2429	.2429	.0415	.0823	4.0	+.5	6.67	+.5	2.24	-.05	2.65	+.15
6	.2447	.2447	.0417	.0642	8.7	+.5	8.70	+1.0	3.35	+.3	2.65	+.1
7	.3203	.3193	.0416	.0625	3.4	+.4	4.0	+ .6	1.70	+.3	1.85	+.4
8	.3214	.3217	.0421	.0622	3.1	+.6	3.36	+ .4	1.55	+.5	1.55	+.6
9	.2434	.2434	.0426	.0634	4.3	+.4	5.0	+ .8	2.5	0	2.5	0
10	.3224	.3217	.0436	.0635	4.8	+.3	3.64	+ .4	2.0	0	1.67	+.3
11	.2432	.2431	.0424	.0629	7.15	+.6	7.15	+ .4	2.67	0	2.67	-.05
12	.2400	.2413	.0395	.0613	6.67	+.4	6.67	+1.2	2.5	+.2	2.22	+.4
13	.2423	.2434	.0401	.0608	7.15	+.05	7.15	+ .1	2.86	0	2.7	0

* Offset measured on
y axis (psi)

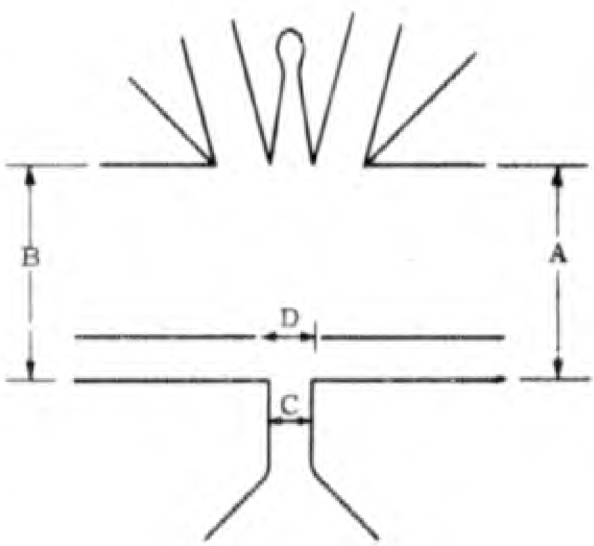


Figure 4. 4. 7

	Dimensional Change X10^{-3} in.						% Gain Change				Null Shift (% of P_s)			
	0 to 2080 hr.			840 to 2080 hr		0-2080	Blocked		.04" Orific		Blocked		.04" Orific	
Sample #	Ave A & B	D	C	Ave A & B	D	Under-cut	5	10	5	10	5	10	5	10
1	3	1.8	0.5	-.7	-.1	2	60	43	-7	-17	11	18	6	5
2	3.6	5.5	3.6	0.7	0.3	14	19	37	-11	-18	6	5	2	2
3	-.4	3.1	1.1	0.1	0.0	0	96	69	10	-18	8	6	5	2
4	2.8	5.4	1.4	0.4	0.2	12	33	0	0	-12	28	16	6	3
5	1.2	.4	1.7	0.4	-.1	0	0	-33	-44	-49	0	7	0	1
6	0.5	3.2	4.2	0.0	-.2	22	-19	-13	-59	-24	14	10	4	2
* 7(840)	0.2	3.5	2.1	---	---	3	67	61	90	19	8	4	6	2
8	1.2	1.0	1.2	-.6	0.3	0	65	49	29	13	12	4	6	3
9	1.9	0.8	1.9	-.7	-.1	0	-15	-20	-18	-18	6	6	8	6
10	-.7	2.1	1.8	-.3	0.4	0	4	37	-9	0	6	8	8	5
11	0.5	2.3	0.1	0.3	0.2	0	7	2	-32	-32	14	16	4	5
12	3.7	3.1	1.5	0.2	0.1	0	0	5	-11	0	0	0	4	2
13	4.1	3.2	1.4	0.6	0.1	3	5	5	-7	-4	8	7	4	4

All Values Positive Unless Otherwise Indicated

* Removal from test after 840 hrs.

Figure 4.4.8 Gain, Null Shift & Dimensional Changes During Life Test.

Figure 4.4.9 Sample #2 - Carbon Steel - 2080 hrs. Wet Steam

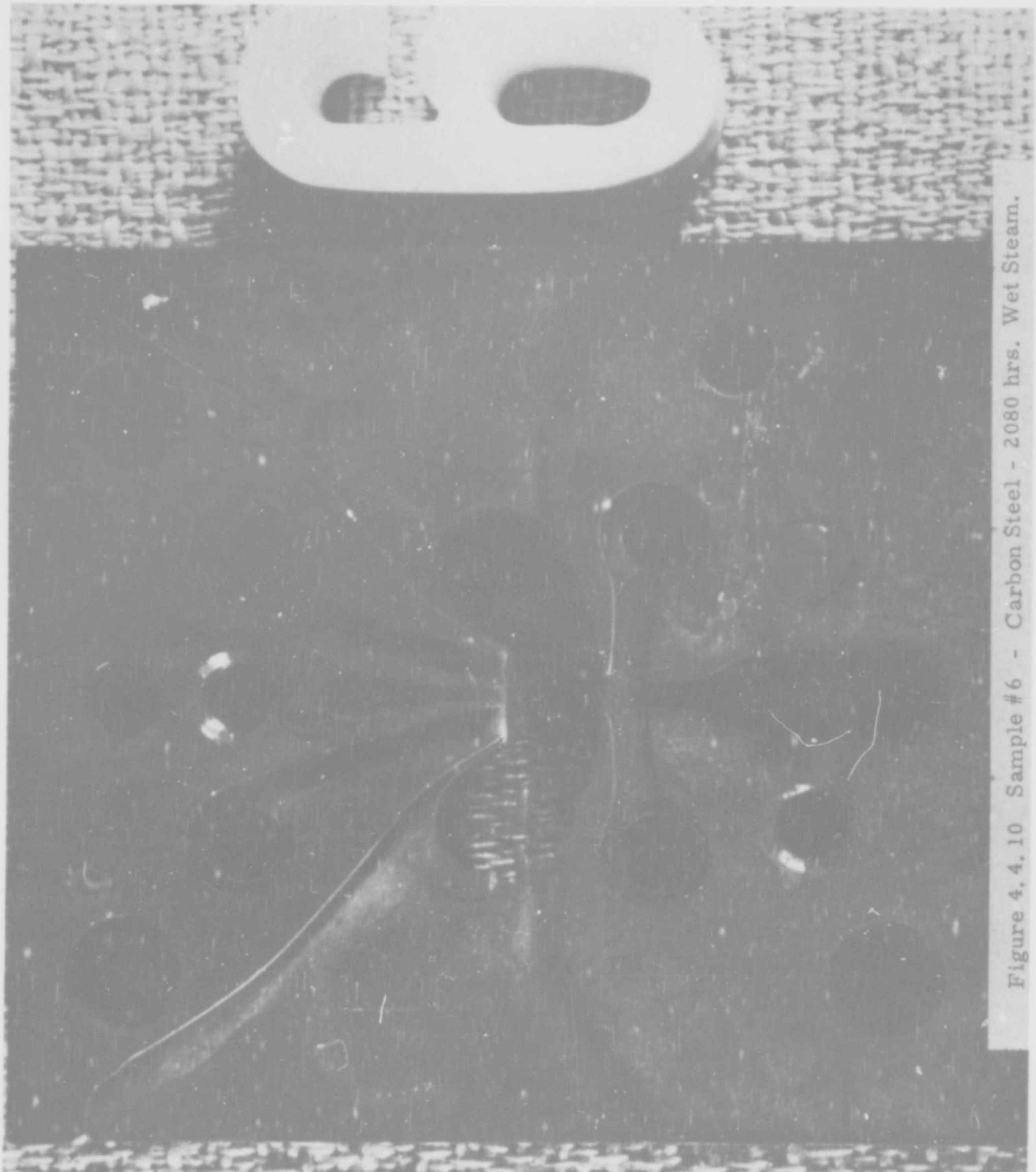


Figure 4.4.10 Sample #6 - Carbon Steel - 2080 hrs. Wet Steam.

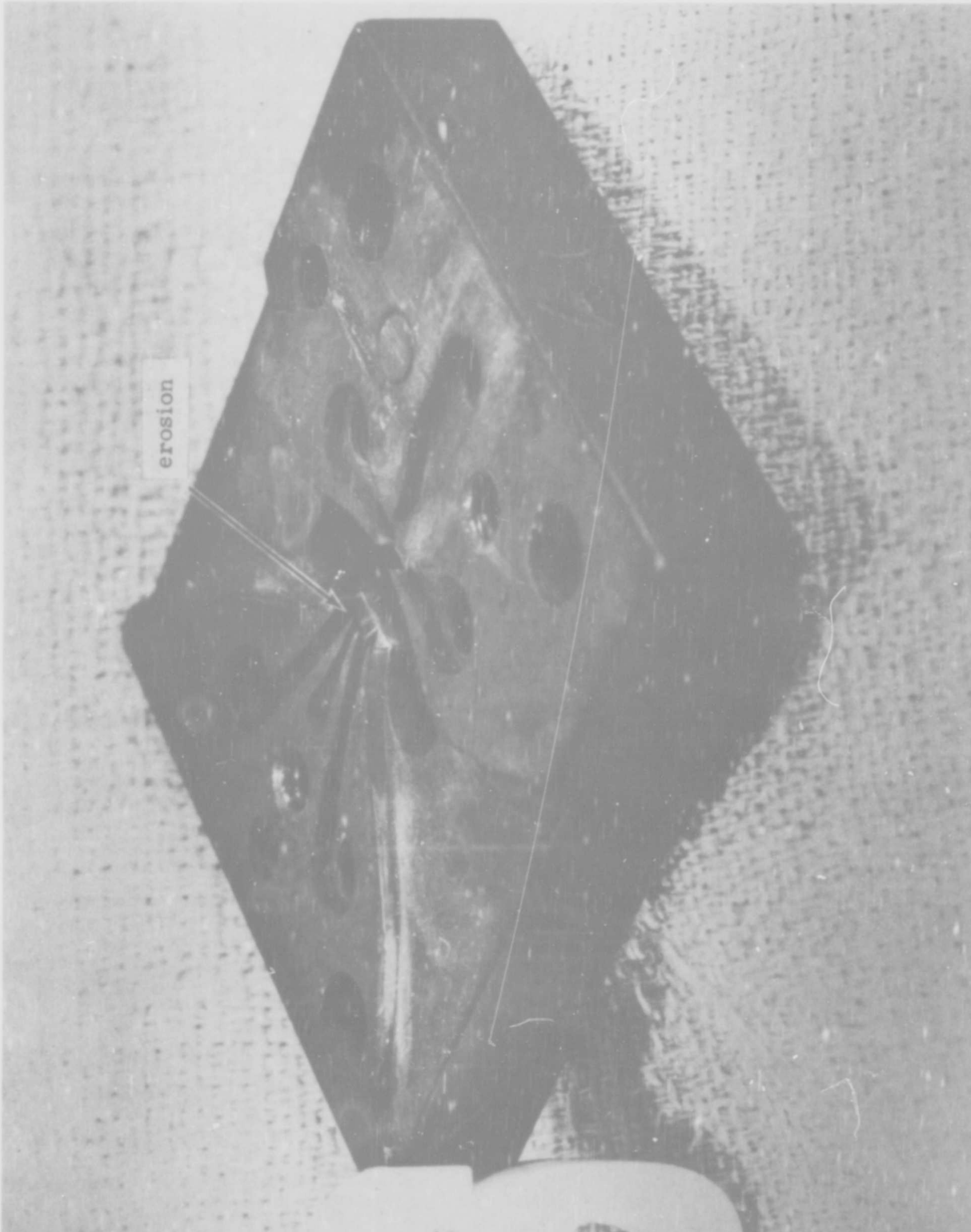


Figure 4.4.11 Sample #6 - Carbon Steel - 2080 hrs. Wet Steam.

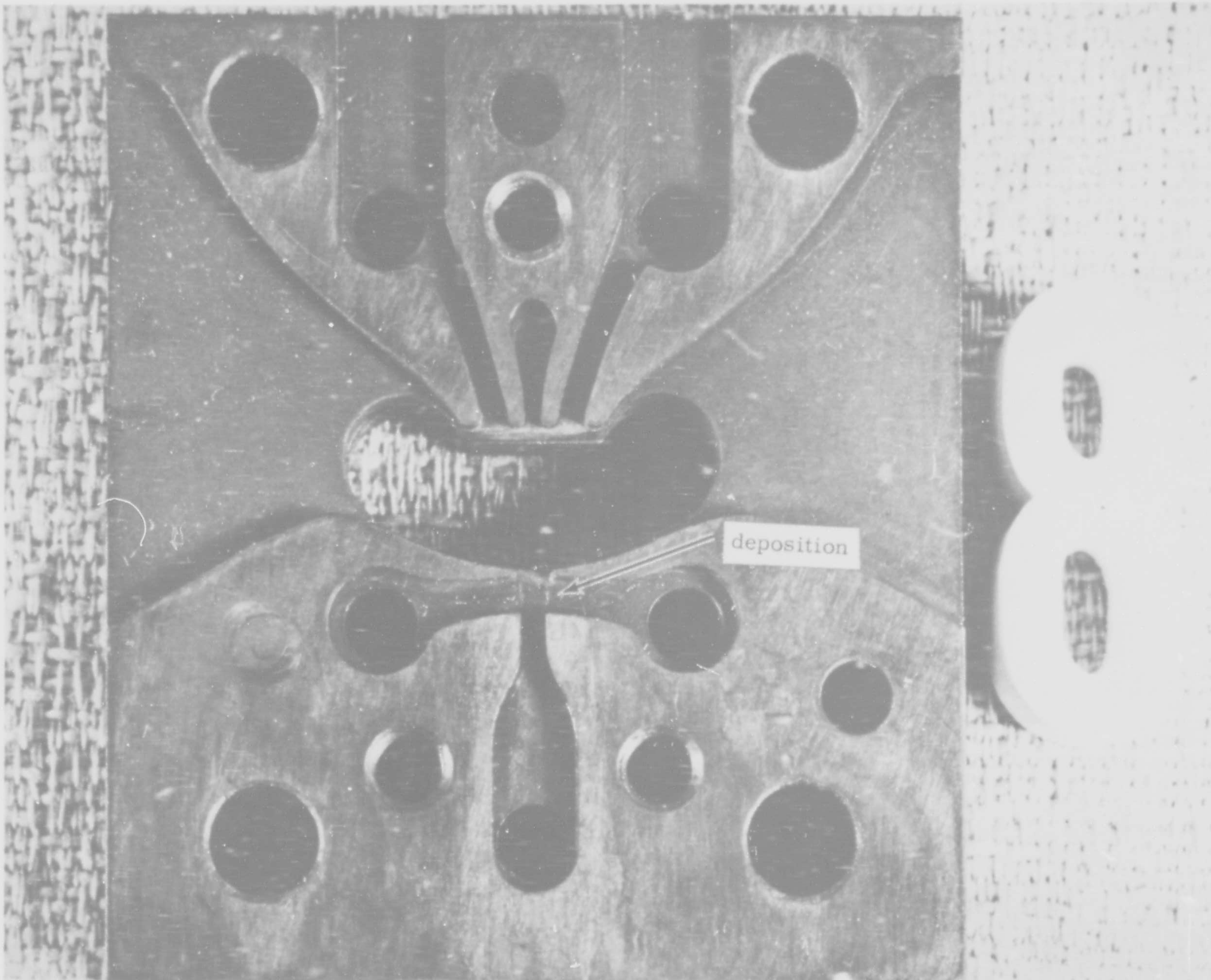


Figure 4. 4. 12 Sample #8 - Chrome Steel - 2080 hrs. Dry Steam.

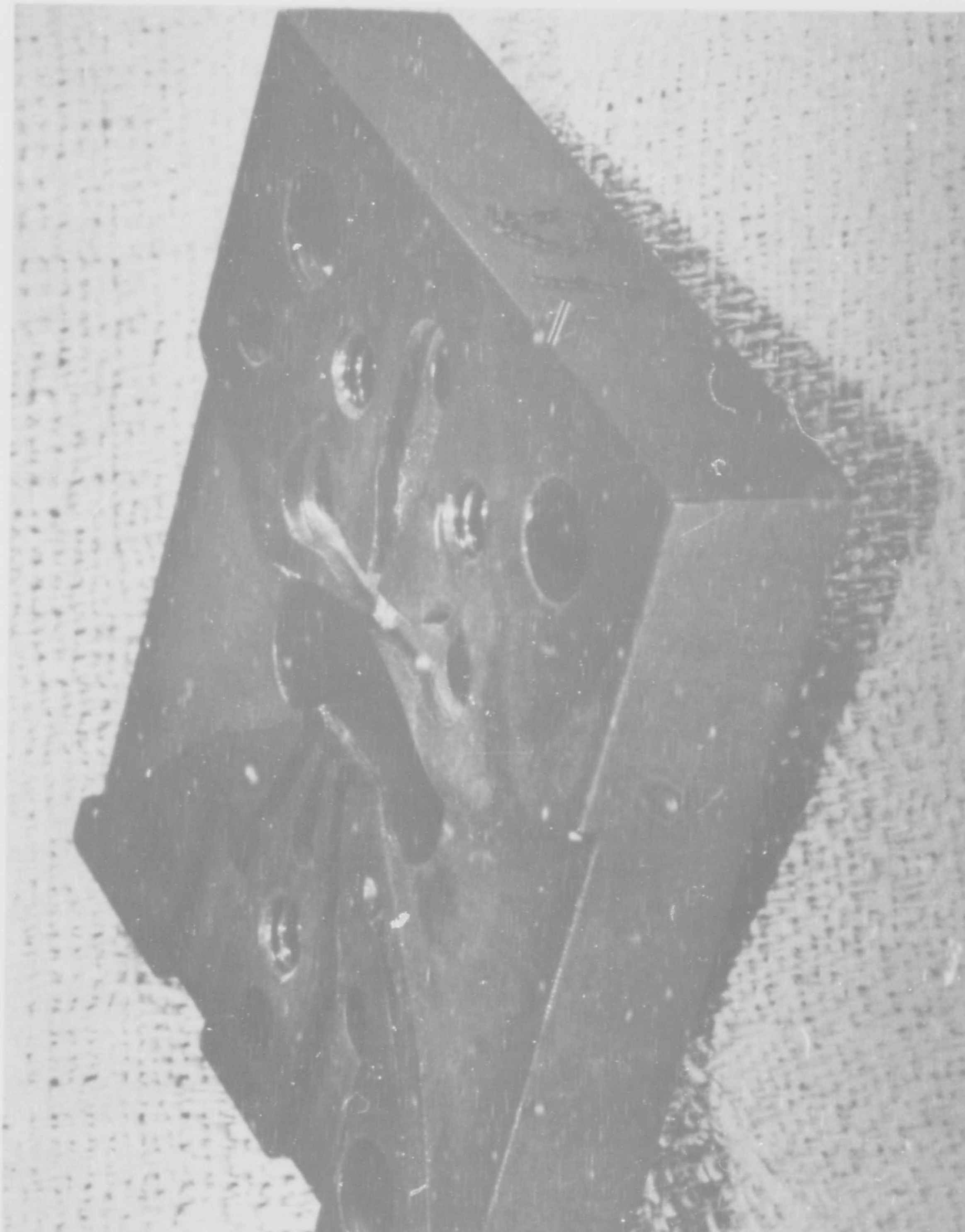


Figure 4.4.13 Sample #10 - Chrome Steel - 2080 hrs. Wet Steam.

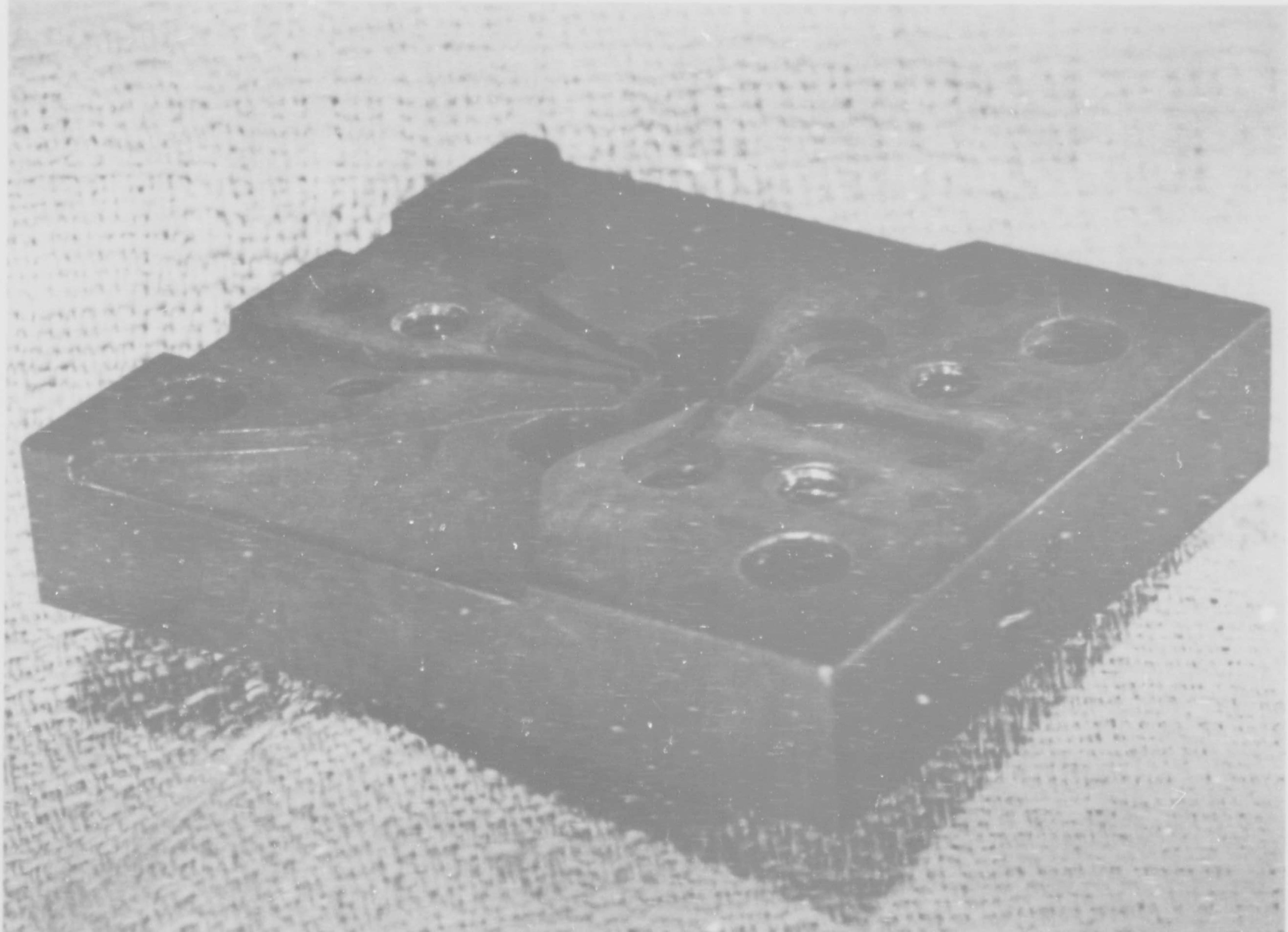


Figure 4.4.14 Sample #11 - Chrome Steel - 2080 hrs. Wet Steam.

elements with high initial gain. The gain variation as a function of power supply has generally decreased after the life test. Of even greater significance is the fact that the gain characteristics after life test are very tolerant of the control port null pressures. With the exception of the two titanium carbide amplifiers, control port null pressures in the range of 15 to 25% of the nozzle supply pressure were required before life test. The linear range of the amplifiers has also improved. Representative gain plots are shown in Figures 4. 5. 1 through 4. 5. 4.

The DC null shift of the amplifiers also changed on the life test. On the speed control system the null shift will manifest itself as a drift of the steady state governing speed. For the specific control loop which has been mechanized the first stage amplifier in the reset circuit is the most critical. The ratio between the percentage change in set speed to a change in bias is 0. 02. A 10% change in DC null shift is the maximum acceptable. By this criteria amplifiers 1, 4, 6, 8 and 11 failed.

The dimensional changes tabulated in Figure 4. 4. 7 are not considered significant. The measurements show a small increase in all the measured dimensions from start to finish of the test. The changes noted from 840 hours to the end of the test show no trend and are within the accuracy limits of the measurement.

The most significant evidence of the effects of erosion is a notching of the receiver tips (See Figs. 4. 4. 9 through 4. 4. 14). The measured maximum depth of the notch is tabulated in Figure 4. 4. 7. In general the rate of erosion follows the jet velocity pattern as expected. The receiver tips near the jet center line have significantly more erosion than the outer tips. The erosion pattern on each tip tends to indicate that the rate at which the tip recedes is not a strong function of the tip geometry. A blunt tip is likely to recede at

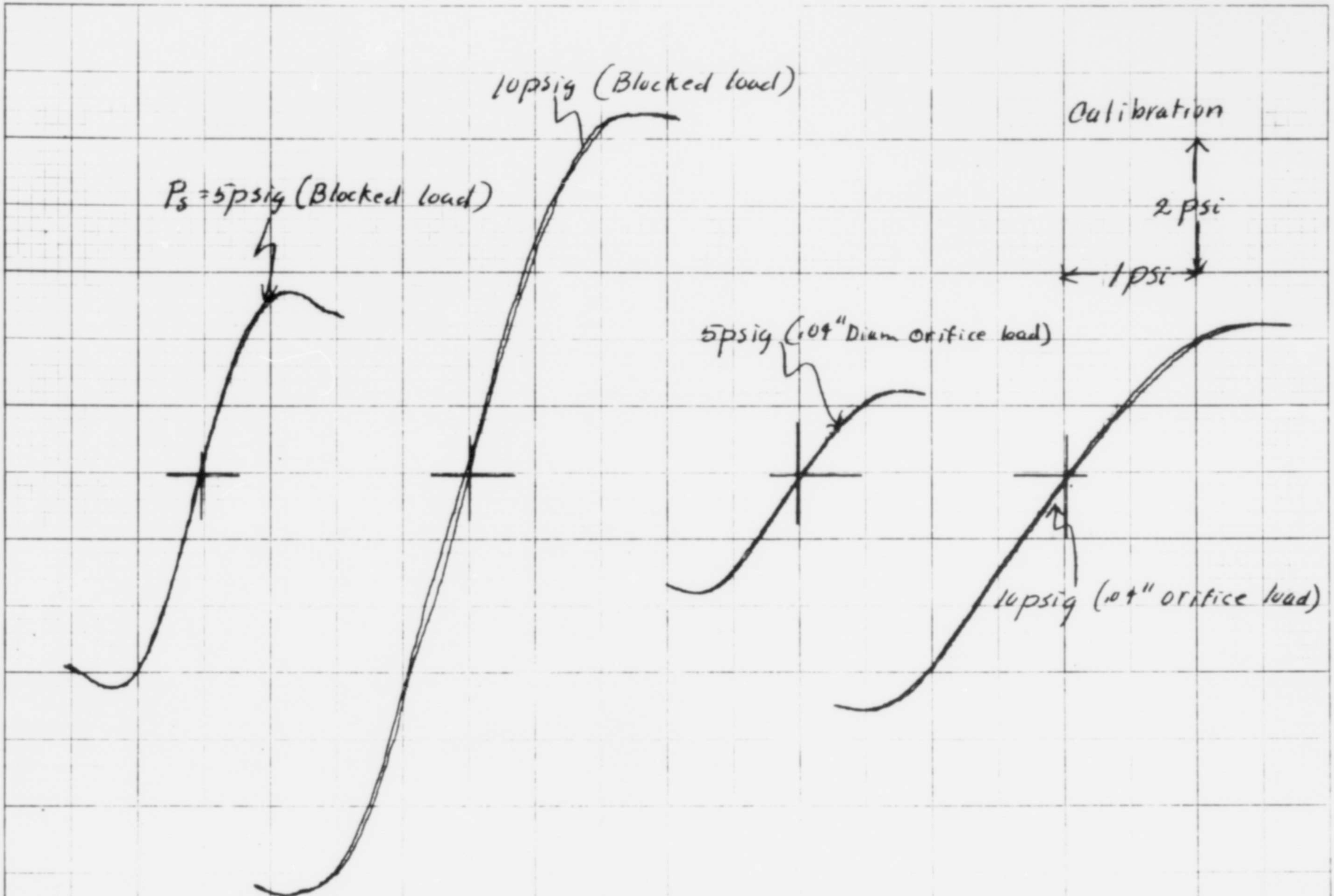


Figure 4. 5. 1 Sample # 13 - Initial Gain Measurement.

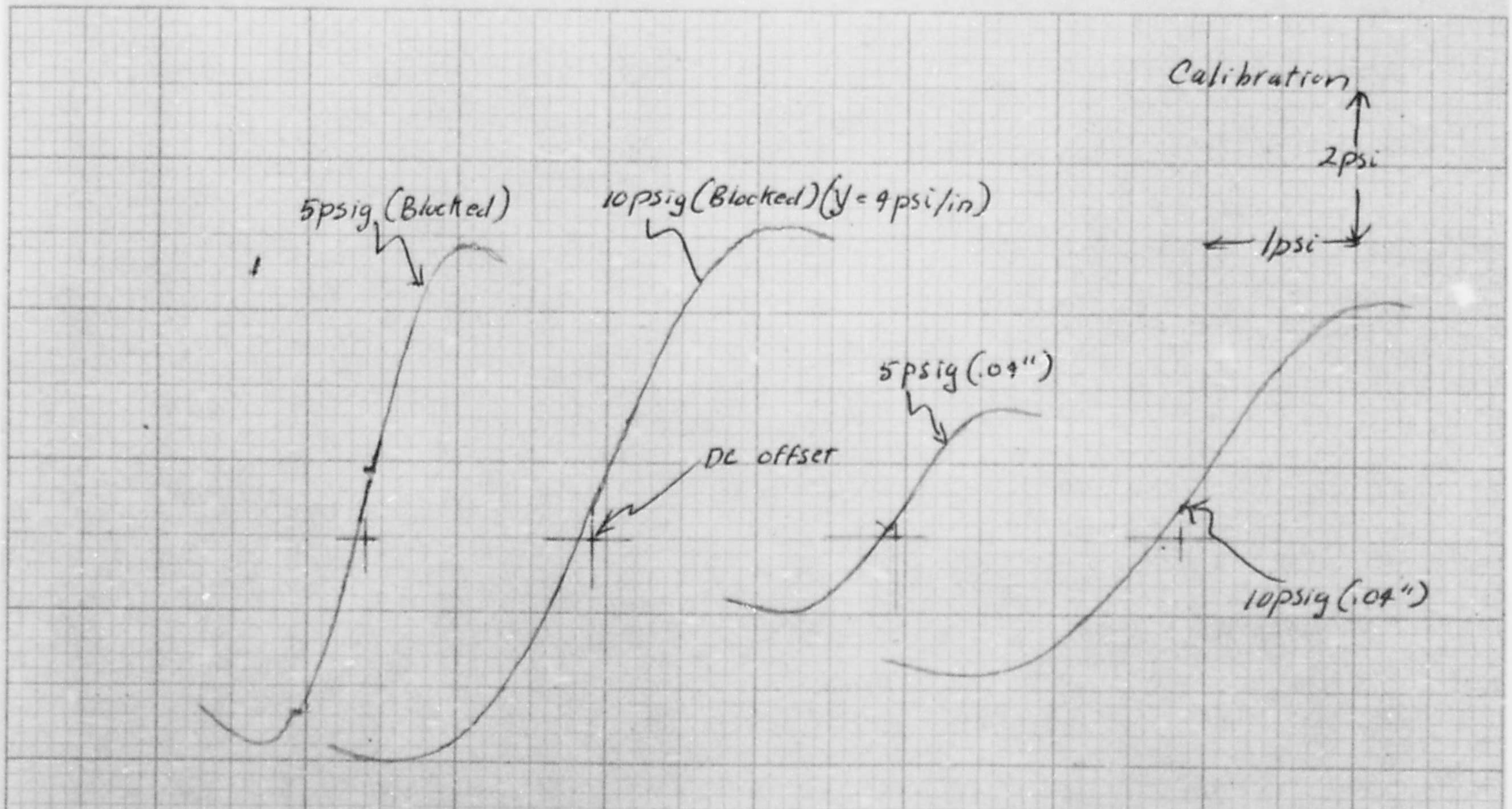


Figure 4.5.2 Sample #13 - Gain Measurement After 2080 hr. Life Test.

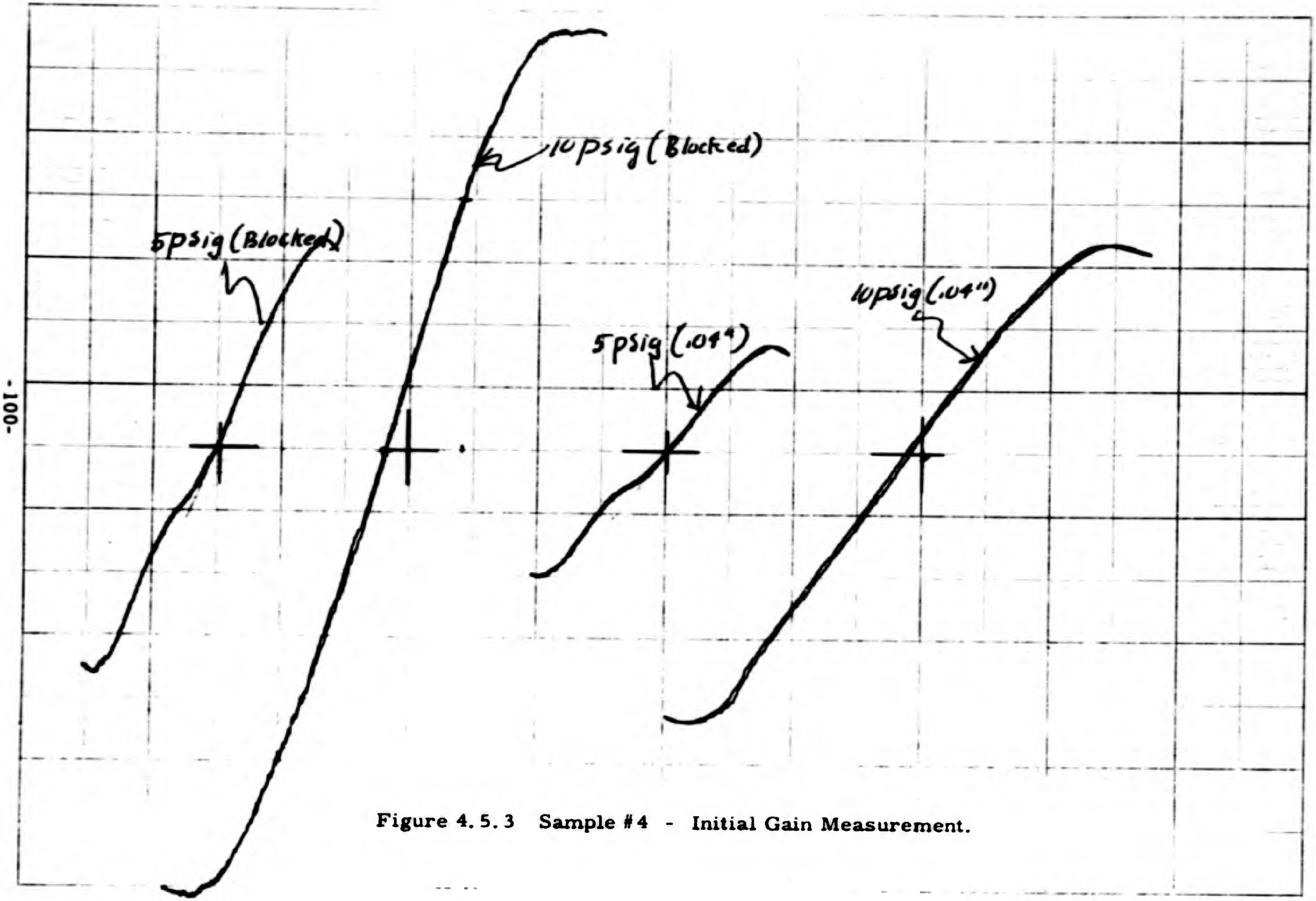


Figure 4.5.3 Sample #4 - Initial Gain Measurement.

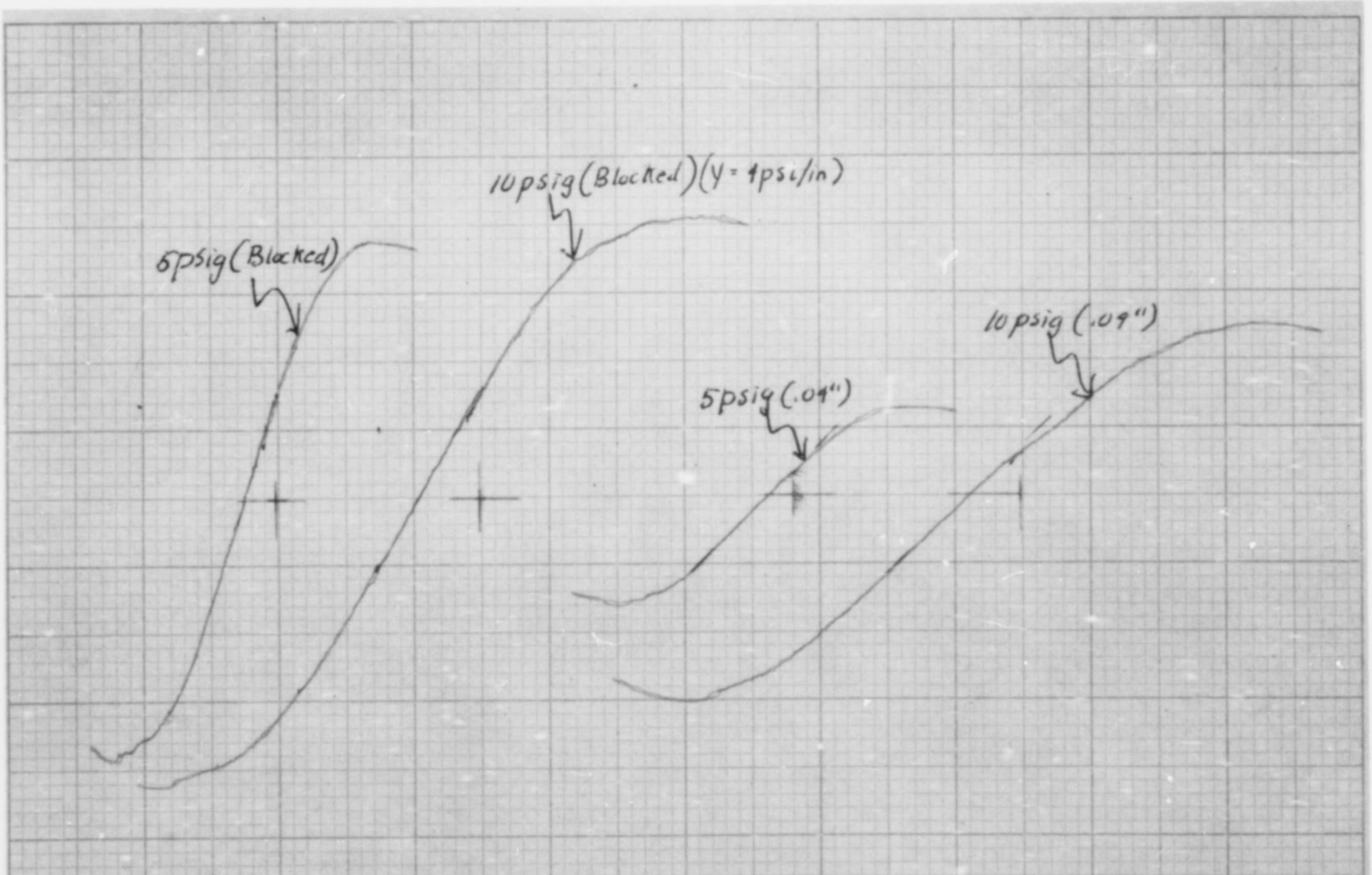


Figure 4.5.4 Sample #4 - Gain Measurement After 2080 hr. Life Test.

the same rate as a sharp tip if they are exposed to the same steam velocity. Comparison of a blunt tip sample (Figure 4. 4. 9) with a sharp tip sample (Figure 4. 4. 11) shows no significant difference in the amount the tip has receded. The blunt tip has been exposed to slightly lower steam velocities because spacing between the nozzle and the receiver is increased on the blunt tip design. These observations do not support the original supposition used to justify the blunt tip amplifier design and the tip design which yields maximum gain should be used.

All of the carbon steel elements including the one operated on dry steam had significant erosion. No detectable erosion is apparent on any of the chrome steel amplifiers. The titanium carbide elements have eroded to a degree where the base metal is exposed and will be the controlling factor in rate of erosion. Titanium carbide is not considered a satisfactory solution to the erosion problem. The test time required to erode the chrome steel element to the same degree as the carbon steel element can be predicted to be approximately 10 years. This is based on the established 40:1 difference in erosion rates in the two materials.

The data required to establish the maximum acceptable erosion was obscured in the life test results by other contributing factors. This data was obtained by taking a brass element identical to the sharp tip design used on the life test and filing the tips to the observed erosion pattern. Figure 4. 5. 5 shows successive gain plots and null shifts observed as a function of the depth of the notch representing the erosion pattern. The gain decreases with notch depth, a 20% reduction occurs when the notch is approximately equal to the maximum observed in life test. A conservative estimate of useful life on a chrome steel amplifier is at least 10 years when subjected to 6% wet steam and steam velocities of 1000 ft/sec.

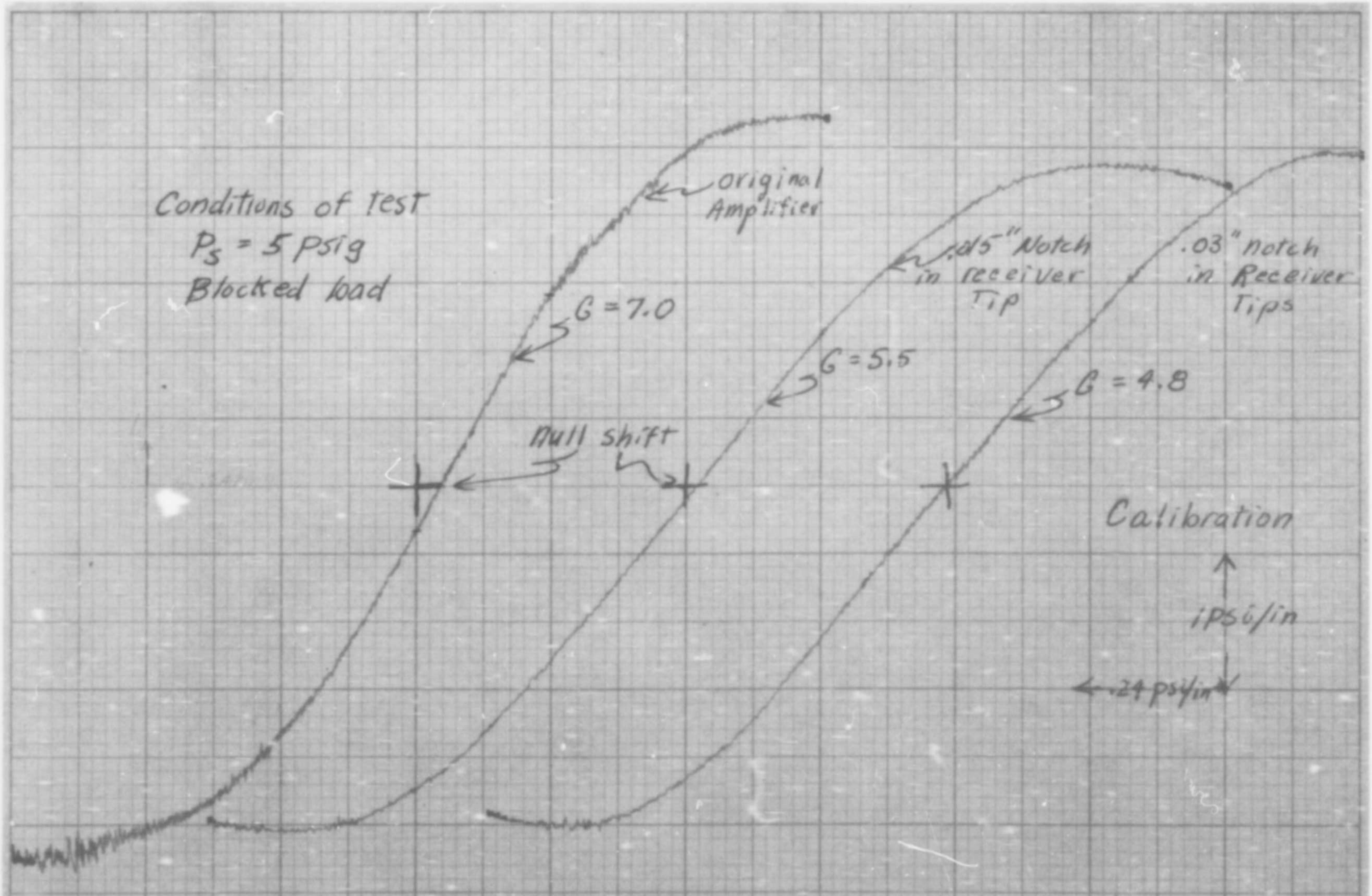


Figure 4.5.5 Gain Measurements on Amplifier with Simulated Degrees of Erosion.

The erosion on the carbon steel amplifier exposed to wet steam is approximately 4:1 greater than on the amplifier exposed to dry steam. A useful life of 40 years can be expected on dry steam elements considering only the effects of erosion.

All of the amplifiers exposed to dry steam showed a pattern of deposited contamination which differed from the wet steam amplifier. Both the wet and dry steam had a fine sprinkling of iron oxide in the vent areas, in addition to this the dry steam had a build-up of hard slag on both sides of the jet as it exits from the nozzle. The build up was less than 0.001/inches during the course of the test and would have no significant effect on performance. If it were to continue to build up at the same rate significant change in performance might be expected after approximately 3 years of operation. It is reasonably certain that on dry steam, useful life will be limited by deposition of carryover rather than erosion. The build up is due to the water soluable products in the boiler feed water and there are possibilities of minimizing the problem by periodic water flushing of the amplifiers. During normal shut down and start up there will be condensate formed in the amplifiers, this may partially alleviate the problem.

Purging the condensate water out of the amplifier presents no particular problem, providing the power supply levels are sufficiently high. Tests were run on an amplifier with a 0.04" x 0.04" nozzle to determine the minimum pressure supply required to purge the amplifier at start-up or to insure that a water slug injected during normal operation would not cause the amplifier to malfunction. On the size and configuration of amplifiers tested a 2 psi pressure drop should be the minimum across any port in the amplifier. To satisfy this condition on the amplifier control ports requires that the minimum nozzle power supply be on the order of 6 psig. A recorder trace of the amplifier output vs. time when water is injected into the nozzle of the amplifier is shown in Figure 4.5.6. A transient lasting for several seconds and equal to approximately 6% of maximum output of the amplifier results from the injection of 1/2 cc. of water. This is a much larger slug of water than would normally be expected under any conditions of operation.

Summarizing the specific conclusions drawn from the steam tests:

1. The conditions required, for erosion by small water particles or solid particles, are satisfied in a fluid amplifier operated on steam. The condition will be satisfied to some degree at the minimum practical nozzle supplies and on both wet and dry steam operation. Any amplifier intended for steam operation should use a chrome steel with high resistance to erosion. With the proper material erosion is not considered a serious problem.

2. Erosion is not strongly influenced by the geometry of the amplifier receiver tips . The same design techniques used in an amplifier designed for operation on air can be applied to an amplifier for steam operation.

3. Contamination of the amplifier when operated in dry steam may be a problem. Contamination will affect other circuit components such as orifices, inductances, passageways and actuators as well as the amplifiers. Complete circuit assemblies and other circuit components will have to be tested under dry steam conditions to make an exact appraisal of the problem.

4. Water logging of the elements is not a problem providing the power supplies are maintained above a specified minimum.

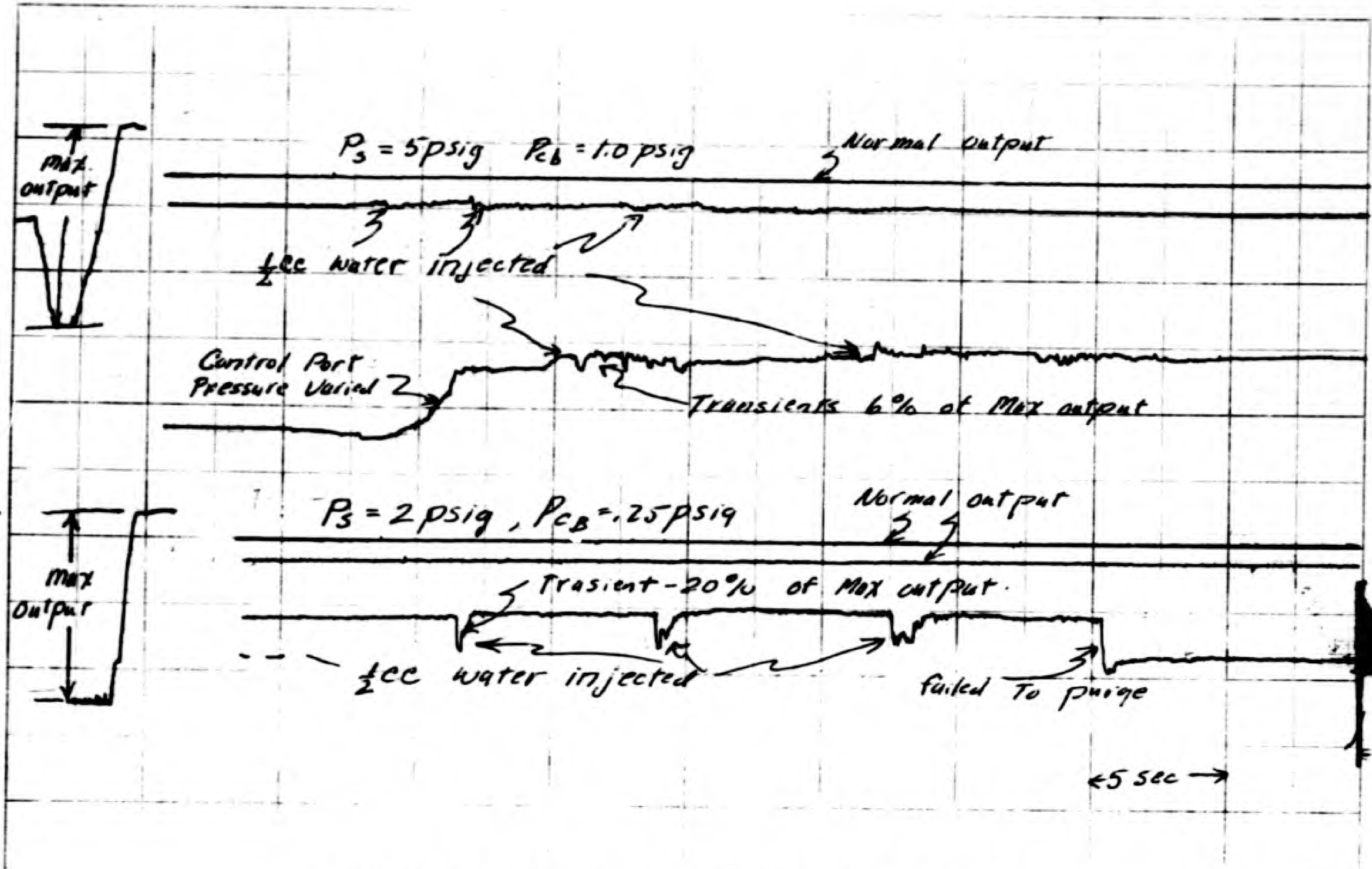


Figure 4. 5. 6. 2 Fluid Amplifier Output - Water Slugs Injected in Nozzle.

5.0 DYNAMIC ANALYSIS OF A STEAM OPERATED ACTUATOR STAGE

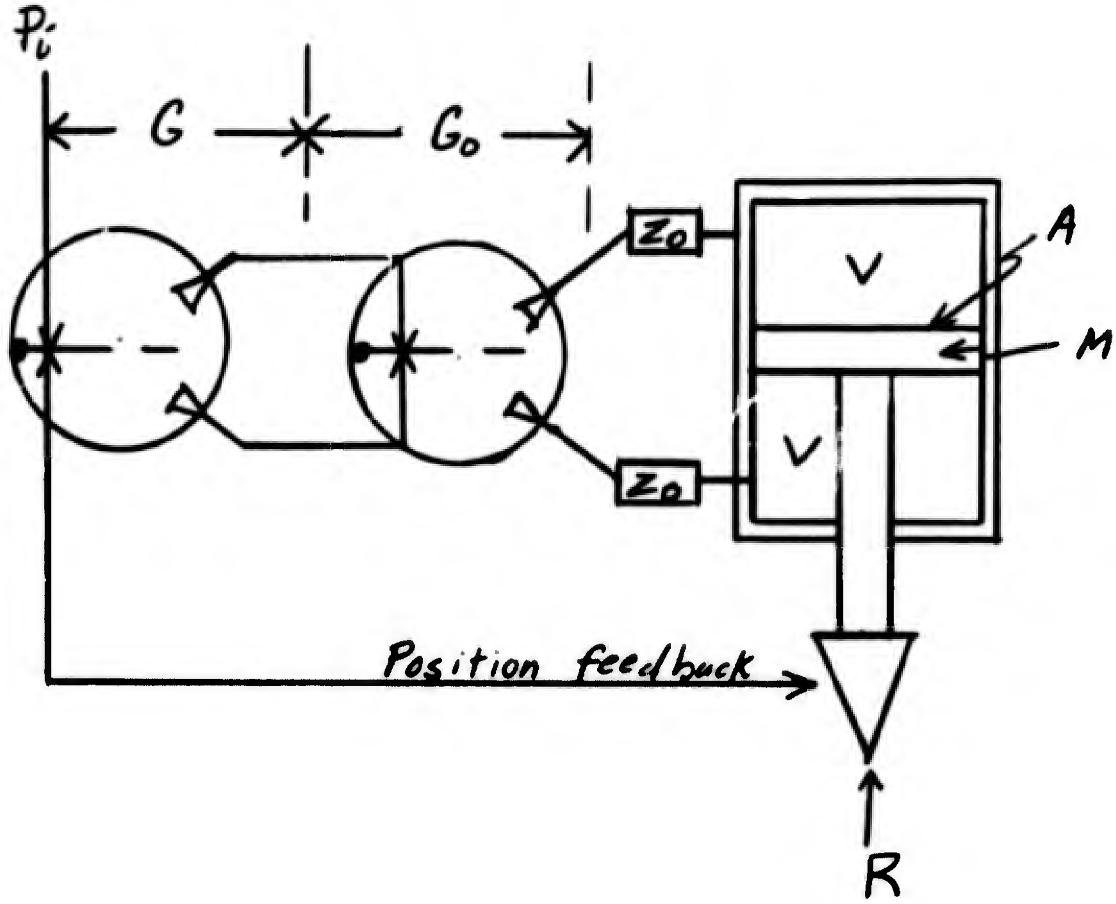
From the standpoint of overall system performance and damping, the requirements of the steam operated actuator stage are the same as for the water actuator discussed in Section 3.4. There are even stronger arguments for using a closed loop actuator configuration. The actuator friction levels will be higher and less predictable than in a water or oil actuator, also in a steam operated loop the integration required for isochronous control can be quite readily obtained in the fluid amplifier circuit; hence, there are no great benefits to be derived from an open loop integrating actuator stage. The closed loop stage is the only approach considered in this analysis. Both high reaction poppet steam valves and balanced low reaction steam valves will be considered.

Figure 5.1 is a simplified schematic of an actuator stage from which the equivalent block diagram can be derived. Only two amplifiers are shown on the schematic, it should be recognized that the gains G and G_o may be the overall gain of a block of amplifiers.

The equivalent block diagram is shown in Figure 5.2. This block diagram applies directly to an actuator driving a high reaction poppet steam valve. The block diagram for the actuator stage of a balanced low reaction valve is shown in Figure 5.3. It can be derived from Figure 5.2 by letting the steam valve reaction gradient go to zero.

5.1 Mechanization of Actuator Stage for a Low Reaction Steam Valve

The closed loop bandwidth requirement on the actuator is assumed to be the same as for the water loop or 70 rad/sec. This defines the open loop crossover on the actuator loop and sets the minimum pneumatic spring mass resonance and the damping factor.



R = Steam valve reaction (lbs.)

H = Steam valve piston travel (in.)

M = Total mass connected to piston (lbs. sec^2/in)

A = Piston area (in^2)

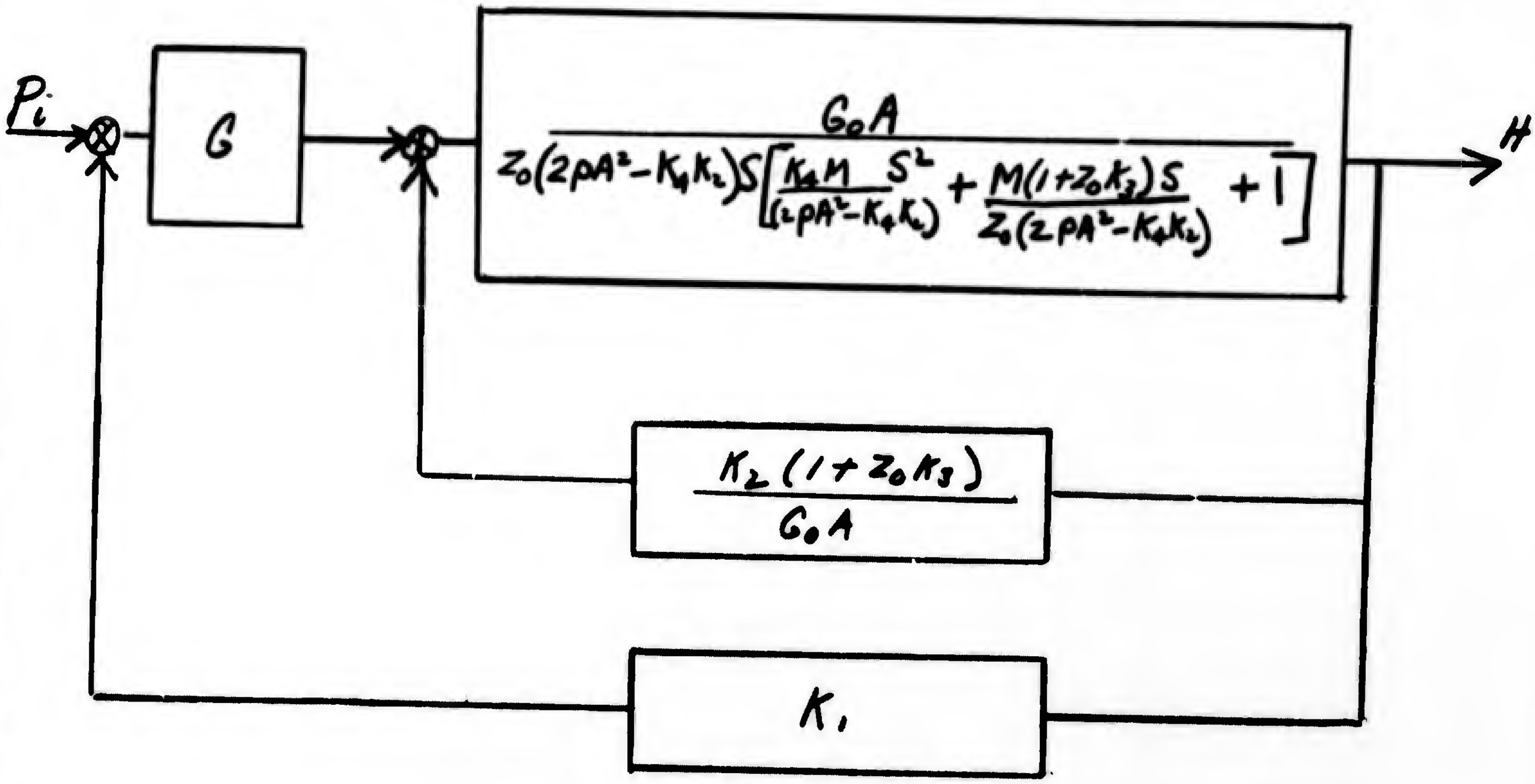
V = Cylinder volume (in^3)

Z_o = Output impedance of last stage

G_o = Blocked load pressure gain of last stage

G = Pressure gain of 1st stage

Fig. 5.1 Simplified Schematic of Actuator Stage



G = Pressure gain of stages preceeding output stage

G_o = Output stage blocked load pressure gain

Z_o = Output impedance of last stage

K_1 = Position feedback

K_2 = R/H = Steam valve reaction gradient

K_3 = Leakage Coefficient

$K_4 = \dfrac{V}{\gamma R_o T}$ = fluid capacitance of actuator volume

ρ = density

Fig. 5.2 Actuator Stage for High Reaction Steam Valve

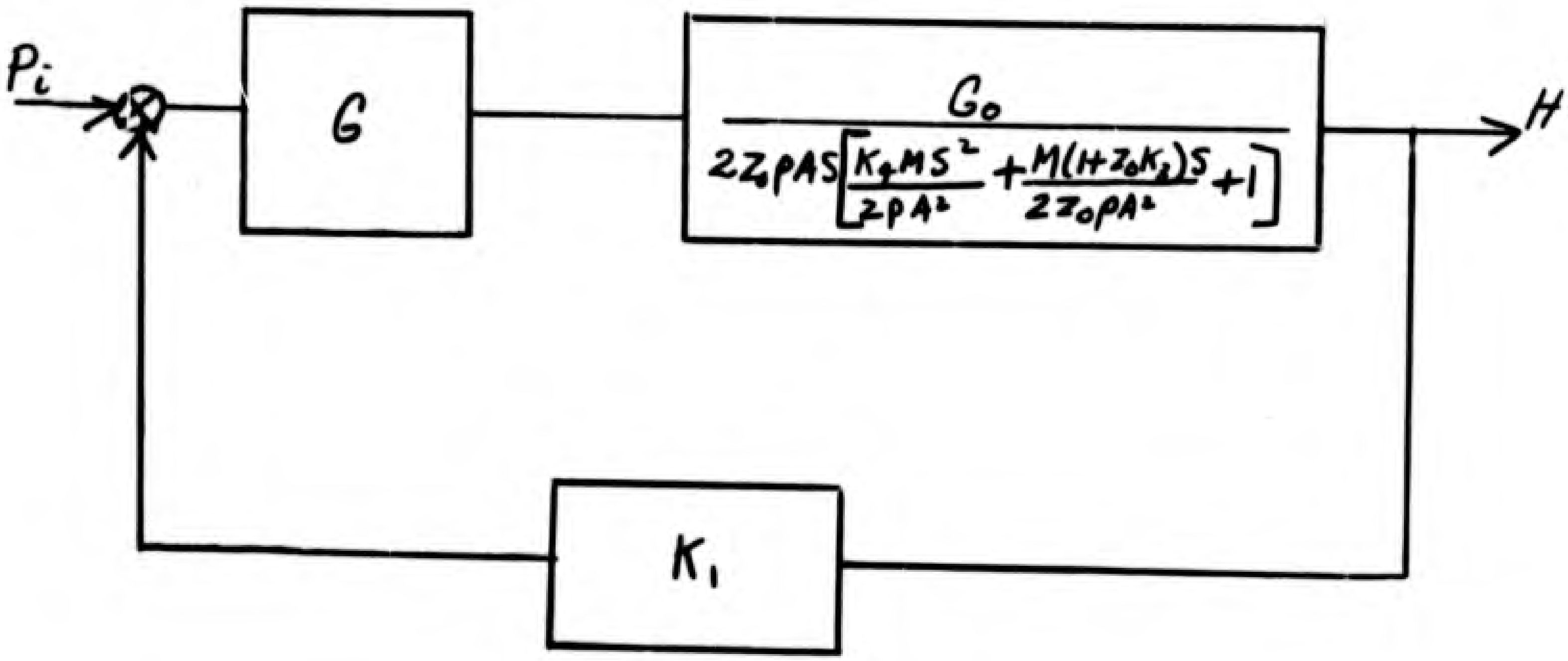


Fig. 5.3 Actuator Stage for Low Reaction Steam Valve

Figure 5.1.1 is a general plot of the open loop characteristics. To maintain stability in the system, the gain at the spring mass resonance must remain below unity under all conditions of operation.

The following assumptions are made:

1. Design for a nominal gain at resonance of -6 db . This will permit a 2:1 gain variation in the open loop gain.
2. Design for a resonance frequency 4 times the open loop crossover frequency or 280 rad/sec. This allows a 6 db rise at resonance to satisfy assumption #1 .
3. Assume the actuator must have a total stroke of 0.5 inches and move a total mass of 15.4 lbs. and has zero leakage.
4. Assume $700^{\circ}F$ steam as the working fluid.

The required actuator area can now be found from:

$$\omega_R = \sqrt{\frac{2\rho A^2}{K_4 M}}$$

For the specific case where the maximum stroke of the actuator has been specified K_4 can be expressed as:

$$K_4 = \frac{V}{\gamma R_o T} = \frac{DA}{\gamma R_o T} = K_4' A$$

where D is the distance between the actuator face and cylinder end wall with the actuator centered. D is assumed equal to the full actuator stroke.

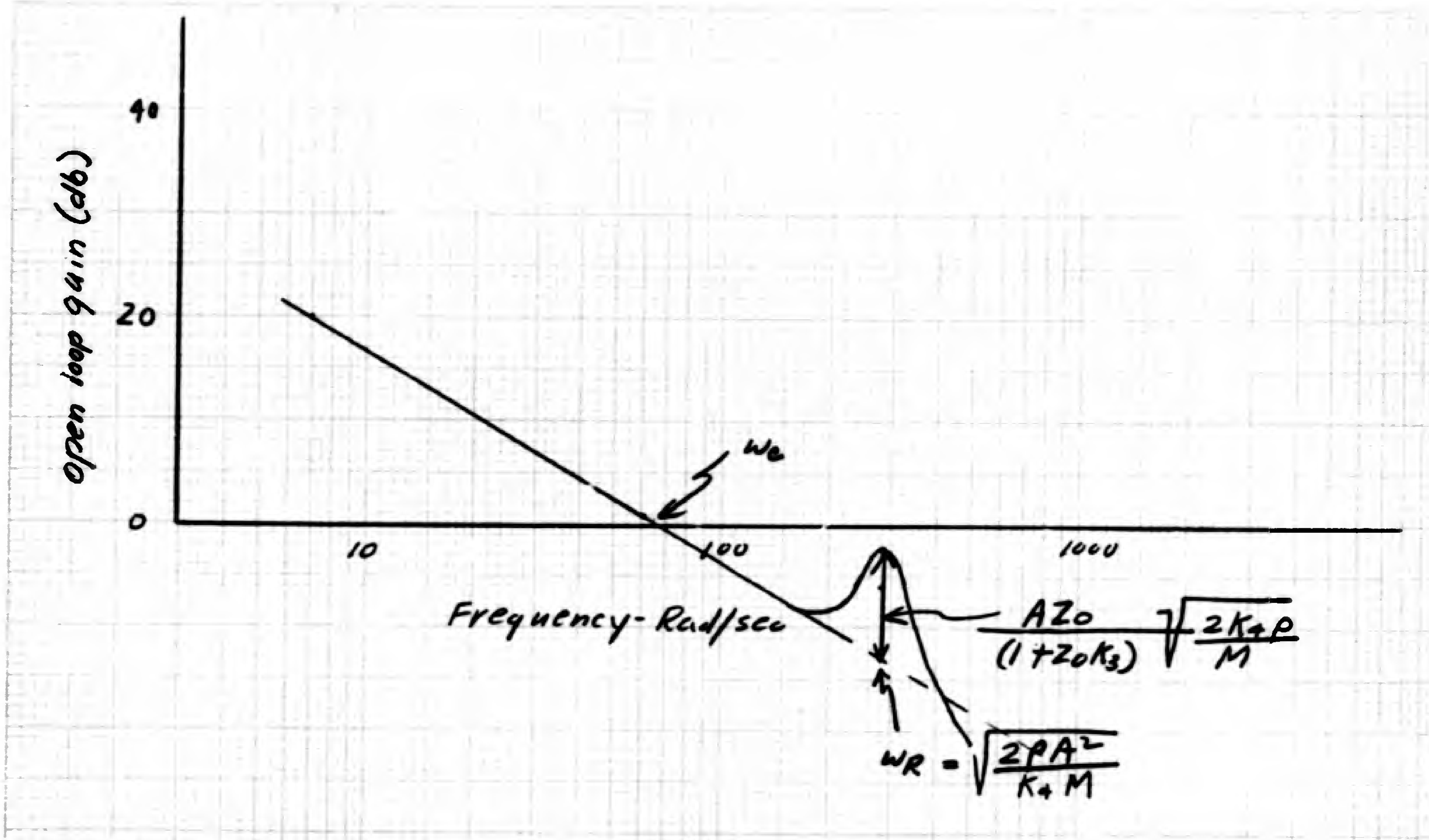


Fig. 5.1.1 Open Loop Characteristics of Actuator Stage

$$\omega_R = \sqrt{\frac{2\rho A}{K_4' M}} = 280 \text{ rad/sec}$$

$$A = \frac{\omega_R^2 K_4' M}{2\rho} = \underline{\underline{21}} \text{ in}^2 \text{ for the assumed conditions.}$$

The damping requirements are derived from:

$$\frac{A Z_o}{(1 + Z_o K_3)} \sqrt{\frac{2 K_4 \rho}{M}} \leq 6 \text{ db}$$

K_3 is assumed equal to zero and expressing K_4 in terms of the actuator stroke,

$$Z_o \leq 2 \sqrt{\frac{M}{2 K_4' A^3 \rho}} = 1000 \text{ sec/in}^2$$

Using the same force capability from the actuator as for the water loop, the power supply required on the output stage is <u>80 psig</u> .

The size of output stage required to satisfy Z_o can now be determined from the empirical equation:

$$Z_o = \frac{0.66}{A_N} \sqrt{\frac{P_s}{2\rho}} \text{ sec/in}^2$$

for the assumed conditions

$$A_N \geq 0.042 \text{ in}^2$$

Using the same criteria as the water loop for the time to travel full stroke, the maximum output impedance to satisfy this requirement can be found.

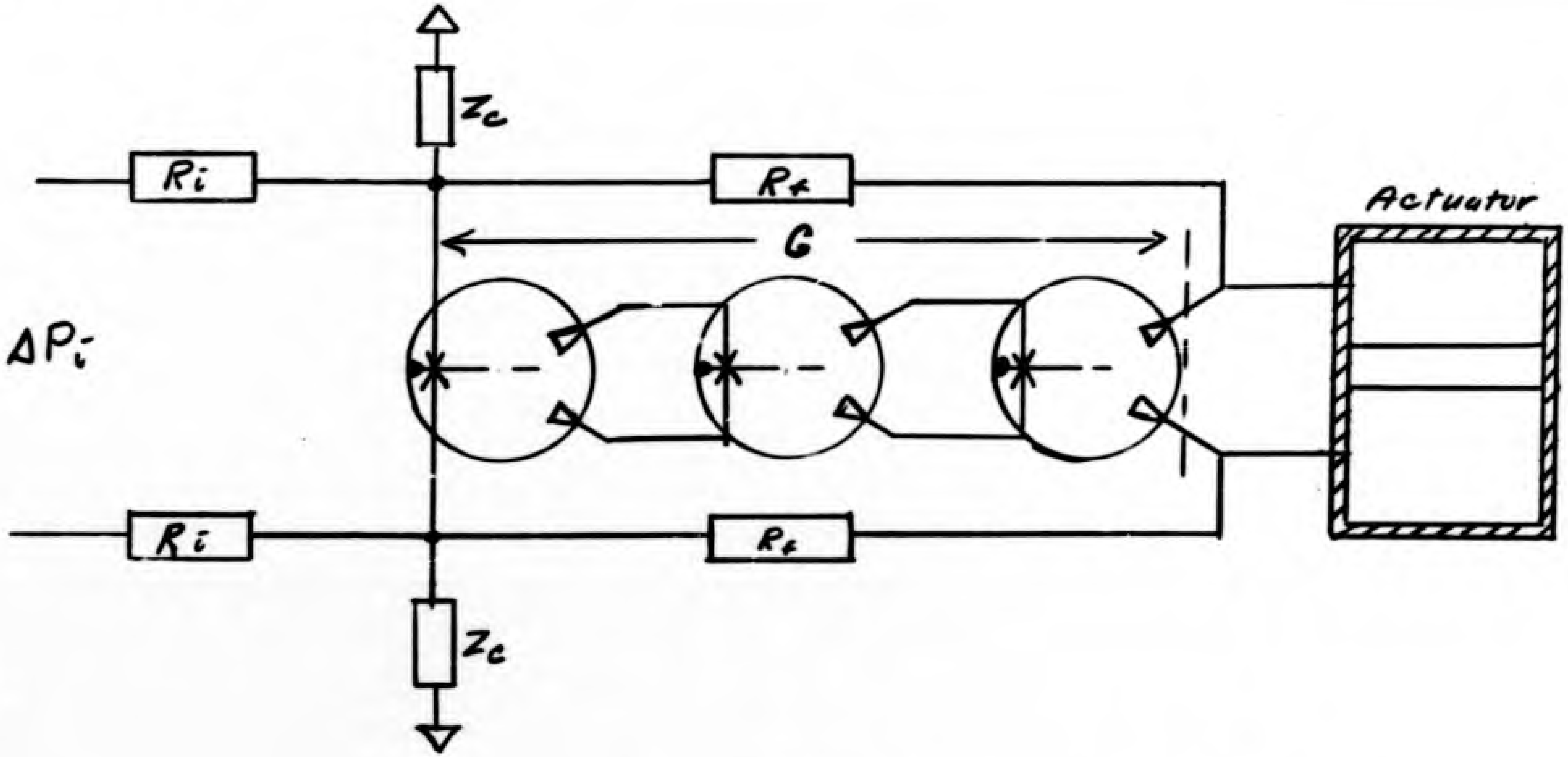
$$Z_o \leq \frac{P_s (K)}{Q} = 3.45\,(10^4)\ \text{sec/in}^2$$

where (K) relates the power supply pressure on the last stage to the effective pressure across the actuator.

The nozzle area required to satisfy this impedance is $\underline{\underline{0.0012}}$ in^2. The power requirement is 0.62 KW which compares favorably with the water actuator. However an amplifier sized to meet the damping requirements would require 21 KW of power.

The output impedance of an amplifier can be lowered by the use of feedback around the amplifier. The output impedance is effectively lowered by the loop gain of the circuit. Application of this technique is not effective in lowering the impedance limiting the maximum stroke capability, this impedance must be made sufficiently low by amplifier design to supply the flow. However the damping requirements have no maximum flow requirements -- only dynamic requirements. Hence this technique can be used to lower the output impedance of the 0.0012 in^2 amplifier by the ratio of 34:1 required to meet the damping requirements.

Figure 5.1.2 is a schematic of the circuit required to satisfy the loop gain requirements of 34 . The loop gain is the product of the forward gain and the feedback gain. The feedback gain is normally limited to about 0.1 to satisfy D.C. level requirements on the amplifier. A forward gain of 340 is then required. This is a minimum of 3 amplifiers.



$$\text{Open loop gain} = \frac{G\,Z_c\,R_i}{R_f\,Z_c + R_f\,R_i + Z_c\,R_i} \approx \frac{G\,Z_c}{R_f}$$

$$\text{Closed loop gain} = R_f/R_i$$

Fig. 5.1.2 Damping Circuit for Steam Actuator

This amplifier is only one functional block in the complete actuator loop -- it performs the required damping on the pneumatic spring-mass resonance. The closed loop gain of this amplifier will be unity if equal feedback and input resistors are used. From the standpoint of linearity it is desirable to make the resistors equal and accept unity gain.

The gain required to satisfy the actuator loop crossover of 70 rad/sec can be deduced from Figure 5.3 . The calculated design parameters have reduced the quadradic in the transfer function to unity gain at the crossover frequency.

The actuator transfer function can then be represented by:

$$\frac{C\,G_o\,K_1}{2\,Z_o\,\rho\,A\,S}$$

and the crossover frequency becomes

$$\omega_C = \frac{C\,G_o\,K_1}{2\,Z_o\,\rho\,A}$$

$\frac{G_o}{Z_o}$ has been fixed by the damping requirements.

The actuator area has been determined, hence $G\,K_1$ is the only independent variable.

$$G\,K_1 = \frac{2\,\omega_C\,Z_o\,\rho\,A}{G_o} = 74 \text{ lbs/in}^2\text{/in}$$

On a steam system where all the pressure supplies must be derived from the high pressure steam it is inefficient to take a high gain (K_1) in the position pickoff. One stage of fluid amplification with a gain of 5 will be used. The required position gradient is then $\frac{74}{5}$ = 15 $lbs/in^2/in$. This can be obtained with a 10 psig supply.

The complete actuator loop will be as shown in Figure 5. 1. 3 A minimum of 4 amplifiers are required. The output amplifier has a 0. 0012 in^2 nozzle operating at 80 psig and driving a 21 in^2 actuator.

The tradeoffs that can be made in the design are somewhat limited. For example, if the actuator stroke and total mass is fixed by the steam valve, then the only way of decreasing the actuator area is to lower the ratio between the open loop crossover and the resonant frequency. When this is done the damping must be increased to maintain the same stability margin for gain variations: to a close approximation the loop gain required to damp the resonant is inversely proportional to the square of the resonant frequency. A 2:1 reduction in the resonant frequency would reduce the actuator area to 5. 25 in^2 and the loop gain required to reduce the output impedance would increase to 136. Current "state-of-the-art" on feedback amplifiers limits loop gain to the order of 50. These gains are obtained with four to five cascaded stages yielding forward gains on the order of 10^4. The 21 in^2 actuator is then fairly representative of the present "state-of-the-art".

5.2 High Reaction Steam Valve

Basically this loop must fulfill all the requirements of the low reaction actuator loop plus being able to cope with the negative steam valve reaction gradients. The specific requirements can be readily extrapolated from the low reaction design by assuming the same actuator and output stage.

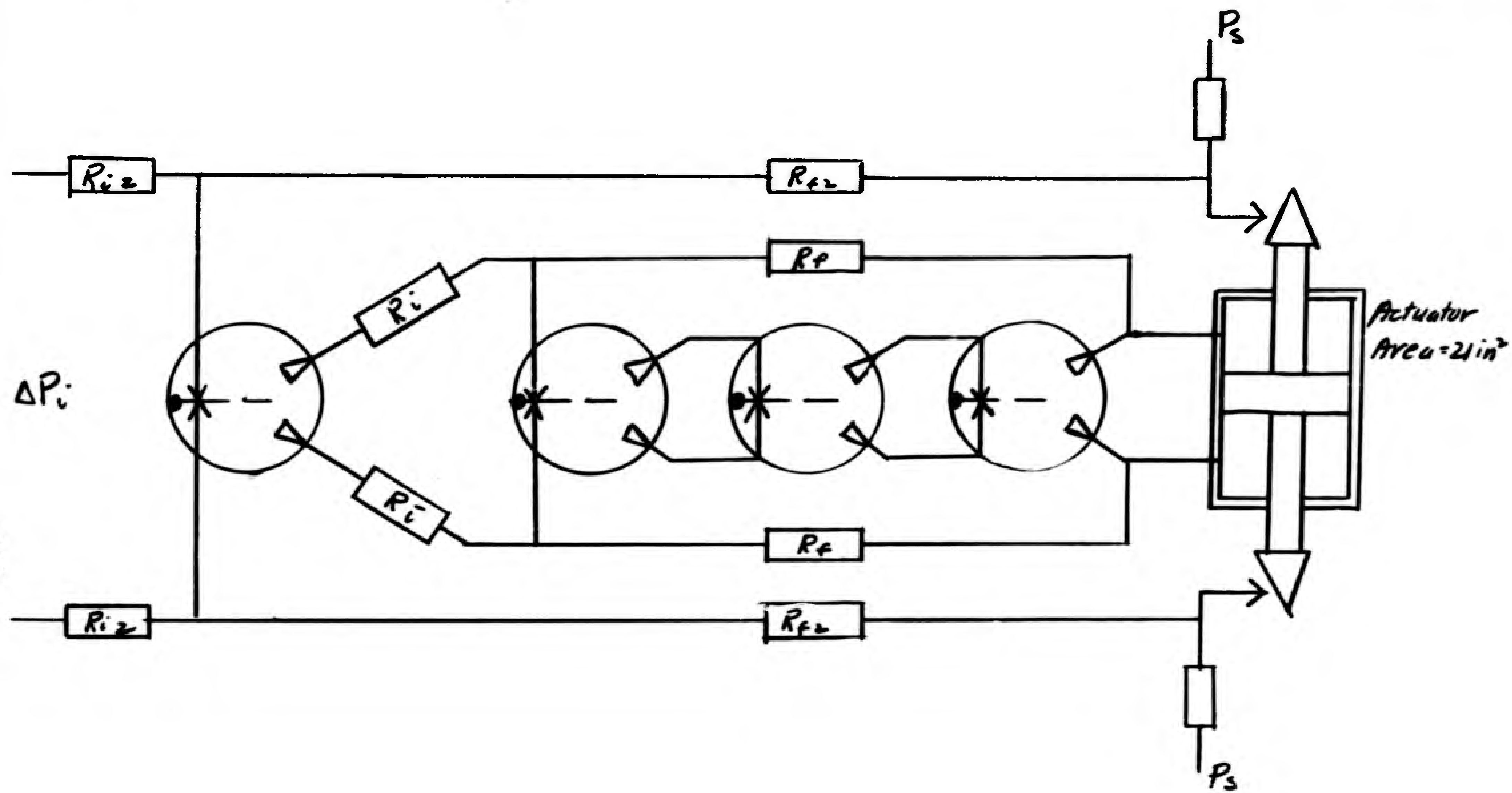


Fig. 5.1.3 Actuator Circuit for Low Reaction Steam Valve

Because of the negative gradient this loop must meet several additional criteria.

Referring to Figure 5.2 it is apparent that

$$G K_1 \geqq \frac{K_2 (1 + Z_o K_3)}{G_o A}$$

and for $K_3 = 0$

$$G K_1 \geqq \frac{K_2}{G_o A}$$

Using a typical value of $K_2 = 1000$ lb/in and $G_o = 1$ from the low reaction design,

$$G K_1 \geqq 48$$

Using a 4:1 margin as in the water actuator loop,

$G K_1 = 192$ which is higher than the 74 required in the low reaction loop. One additional stage of amplification will be required.

Also $2 \rho A^2$ must be greater than $K_4 K_2$.

$K_4 K_2 = 0.007$ lbs/in with $K_2 = 1000$ lbs/in, this compares to $2 \rho A^2 = 0.022$ lbs/in, hence the low reaction design satisfies this criteria with sufficient margin.

The pneumatic spring mass resonant frequency is reduced by the factor:

$$\left(\frac{2 \rho A^2 - K_4 K_2}{2 \rho A^2}\right)^{1/2} = \left(1 - \frac{K_4 K_2}{2 \rho A^2}\right)^{1/2}$$

The gain at resonance then remains at - 6 db and there is no need to add loop gain to increase damping.

With the exception of an increase in the G K_1 product and the addition of one more amplifier outside the pressure feedback loop, the high reaction valve actuator stage is identical to the circuit of Figure 5. 1. 3.

5. 3 Conclusions

The dynamic requirements of a steam-operated actuator for the steam valve can be satisfied with fluid amplifier circuits by the use of high gain feedback circuits. The efficiency compares favorably with electro-mechanical hydraulic systems. Present "state-of-the-art" in fluid circuit technology limits the minimum actuator area to approximately 20 in^2 for a 70 rad/sec. closed-loop bandwidth on the actuator stage. The practical problems associated with the actuator are the primary obstacles to the mechanization of the actuator stage.

APPENDIX A. SYSTEM ANALYSIS

On a steam turbine generator control system severe disturbances may be introduced at the steam valve actuator by the steam reaction forces on the valve. Steam valves fall into two general categories; the balanced low reaction valve and the high reaction multiple poppet valve. The high reaction poppet valve has found wide acceptance because of its high efficiency and low manufacturing costs.

The reaction forces are of sufficient magnitude where they must be considered in the system design. Figure A. 1 is a typical plot of steam reaction forces vs. valve travel. The characteristic shown has a positive force gradient due to a preload spring, i. e. the force increases with displacement and removal of the driving force restores the valve to its neutral position. Superimposed on the spring gradient is the steam reaction force. There is a finite cracking force associated with each poppet valve which must be supplied by the actuator, the steam reaction forces then decrease with further valve displacement. If the actuator force is maintained constant at its cracking value the valve will continue to accelerate to its wide open position. In a closed loop speed control one or more poppet valves will be chattering between limits unless certain gain and bandwidth criteria are met.

The criteria for preventing valve chatter in a control loop utilizing an open loop actuator stage can be deduced from the simplified block diagram of Figure A. 2. It is assumed that the actuator uses an incompressible fluid and the resonance due to valve mass and compressibility can be neglected relative to system cross over frequencies. Second order lag time constant associated with the system have also been neglected.

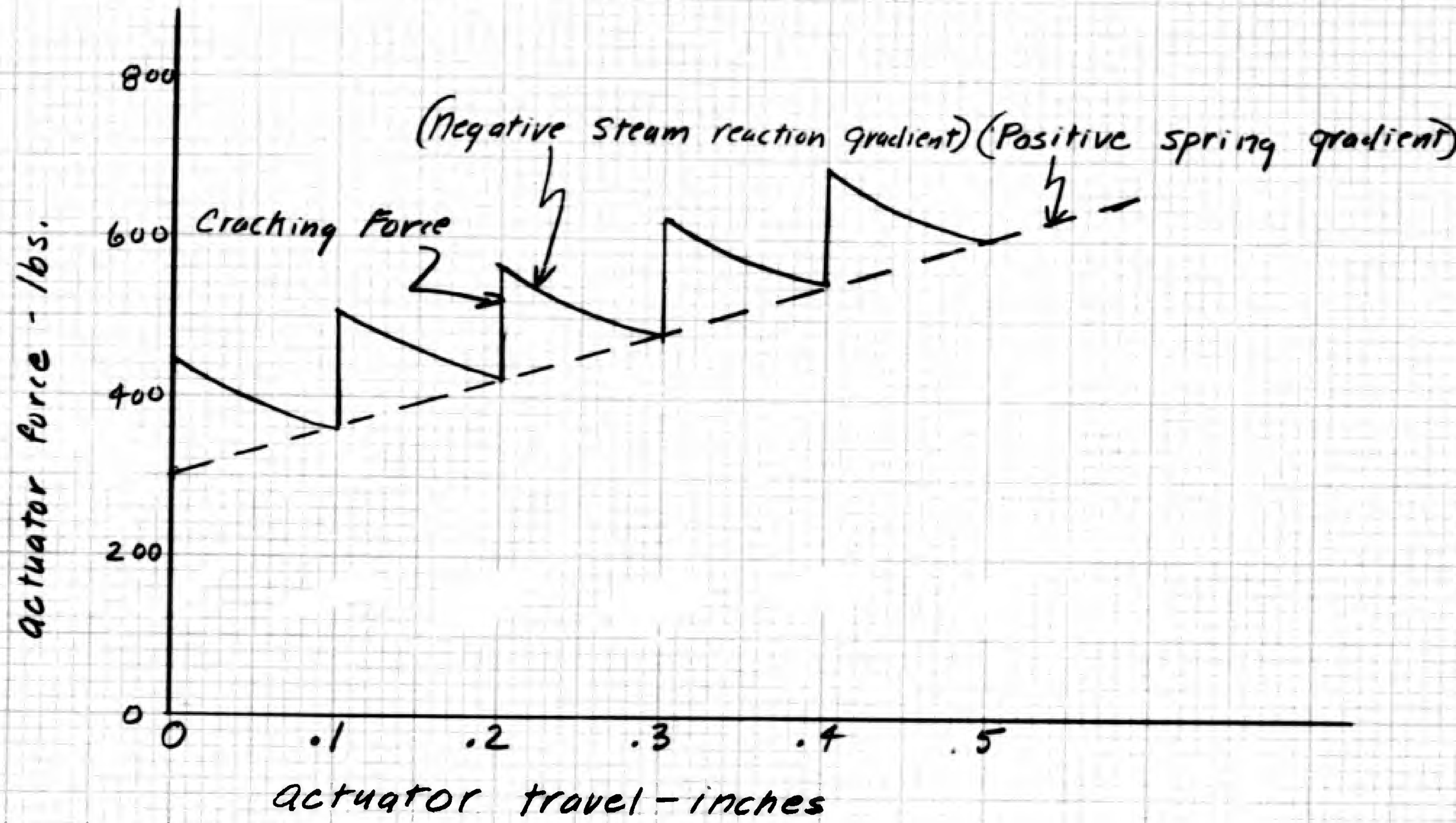
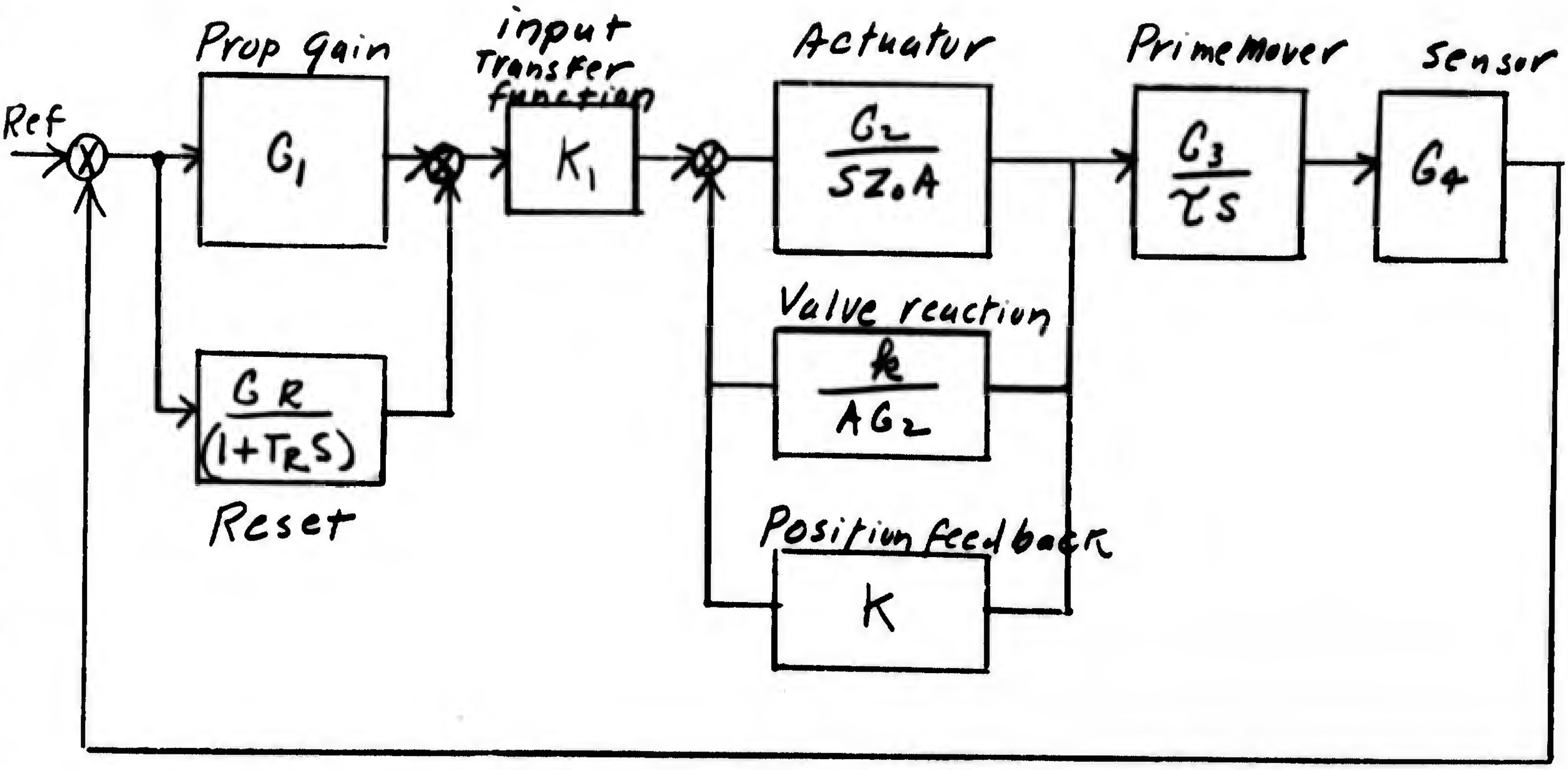


Figure A. 1 Typical Steam Valve Reaction Force.



G_1 - Proportional gain.

G_R - Reset gain.

T_R - Reset time constant.

K_1 - Input transfer function.

G_2 - Actuator area.

Z_o - Output impedance of actuator drive.

A - Actuator area.

k - Position feedback transfer function.

G_3 - Turbine and Valve gain.

G_4 - Speed sensor gain.

Figure A.3 Simplified Block Diagram Speed Control - Closed Loop Actuator.

The loop gain of the system is:

$$G_L = \frac{G_1 G_2 G_3 G_4 A\,(1 + T_L S)}{S \tau k \left(1 + \frac{S Z_o A^2}{k}\right)}$$

The loop cross over frequency (W_c) is obtained by equating the loop gain expression to unity

$$W_c = \frac{G_1 G_2 G_3 G_4 T_L}{Z_o A \tau}$$

The characteristic equation from which the stability criteria can be deduced is $\left|1 + G_L\right| = \quad Z_o A^2 S^2 + (\tau k + G_1 G_2 G_3 G_4\ A T_L)\, S + G_1 G_2 G_3 G_4 A$

The steam valve reaction gradient (k) is negative and to insure that no positive real roots exist in the system:

$$G_1 G_2 G_3 G_4 A T_L > \tau k$$

putting this in terms of the crossover frequency.

$$W_c > \frac{k}{Z_o A^2}$$

Generally nominal gain is selected for a gain factor of at least 4: 1 over the minimum. This insures stable operation under all conditions of operation.

In the practical case

$$W_c = \frac{4\ k}{Z_o A^2}$$

Typical values for k are 1000 lbs/in. and the maximum Z_o for use with an 8 in^2 actuator for a 10 sec. turbine is 5 lbs sec/in^5.

The required crossover then becomes

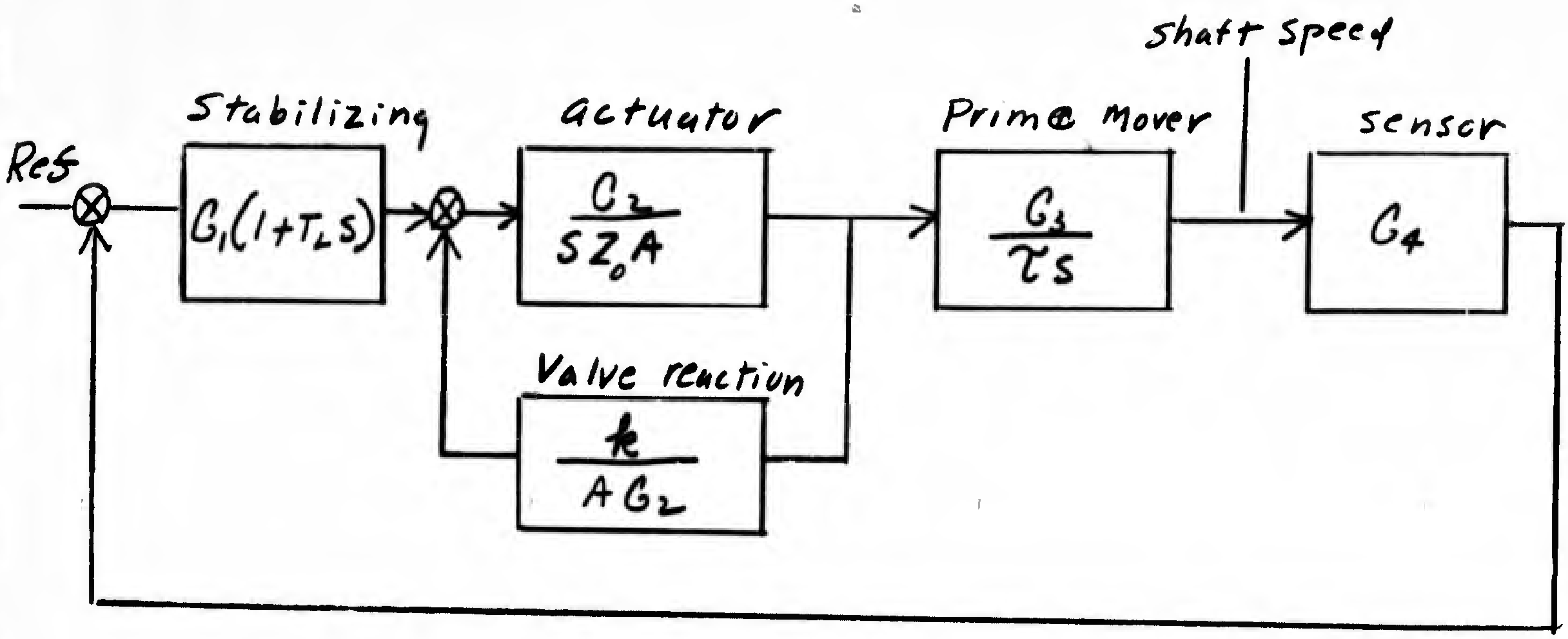
$$W_c = \frac{4\ (1000)}{5\ (8)^2} = 12.5 \text{ rad/sec.}$$

This is a crossover frequency approximately 2.5 times that required for overall system dynamics and is directly set by the specific valve characteristics. The minimum crossover will be directly related to the reaction gradient and inversely to the turbine time constant. The crossover requirement can be reduced by increasing Z_o or the actuator area. However, Z_o and the actuator area are not independent. Z_o is essentially set by the maximum stroke velocity requirements of the actuator. As the area increases Z_o must decrease. If the output stage is designed for maximum efficiency then Z_o must vary inversely as the square of the area and the product $Z_o A^2$ remains constant. The open loop actuator stage is not feasible to use with the high reaction poppet type valve. It is limited to applications where the steam reaction forces are essentially zero and a net positive gradient can be provided by a spring preload.

Figure A.3 is a simplified block diagram of a closed loop actuator system. In this diagram K represents a position feedback from the actuator. K_1 is the input transfer function.

The overall system crossover for this system is:

$$W_c = \frac{G_1\ G_3\ G_4\ K_1}{S\ K\tau}$$



G_1 - Proportional gain.

T_L - Lead time constant.

G_2 - Proportional gain in actuator loop.

Z_o - Output impedance of actuator drive.

A - Actuator area.

k - Steam valve force gradient lbs/in.

G_3 - Turbine and steam valve gain.

τ - Turbine time constant.

G_4 - Speed sensor gain.

Figure A. 2 Simplified Block Diagram. Speed Control - Open Loop Actuator

The characteristic equation for the actuator loop alone is:

$$S Z_o A^2 G_2 + G_2 k + G_2^2 K A$$

To avoid positive real roots with negative k

$$G_2 KA \geqq k$$

and for a practical design

$$G_2 KA = 4k$$

The stability criteria can be satisfied by making G_2 sufficiently large. G_2 does not appear in the expression for the overall speed loop crossover hence, the system crossover can be selected independently of the steam valve actuator stability criteria.

The actuator open loop gain is

$$\frac{k + K A G_2}{S Z_o A^2}$$

If the practical stability criteria of $KAG_2 = 4$ k is satisfied, the open loop gain expression reduces to:

$$\frac{3 K G_2}{4 S Z_o A}$$

and the minimum crossover frequency becomes:

$$W_c = \frac{3 KG_2}{4 Z_o A}$$

Using the same values for k, Z_o and A as used in the open loop actuator approach

$$W_c = 28 \text{ rad/sec.}$$

This is not an excessive crossover for the actuator stage, overall system damping requirements will normally require actuator loop crossover in the range of 50 - 70 rad/sec.

For all practical purposes the overall requirements of the loop will result in $G_2 KA >> k$ and the loop gain and crossover can be expressed as:

$$G_L = \frac{K G_2}{S Z_o A}$$

$$W_c = \frac{K G_2}{Z_o A}$$

For a general purpose speed control loop, i. e. one that does not have to be tailored to the particular steam valve used, a closed loop actuator stage is a prerequisite.

Allocation of Component Bandwidths

The analog computer studies completed on Phase I of the current contract defined the attenuation vs. frequency characteristics of the overall control loop. The system to be mechanized will be somewhat different than the configuration studied in Phase I, however, the crossover frequency and phase margin at crossover can be used as a guide for dynamic performance.

Figure A. 4 shows a plot of the acceptable gain plot from the Phase I studies. Two plots which are representative of the system to be mechanized

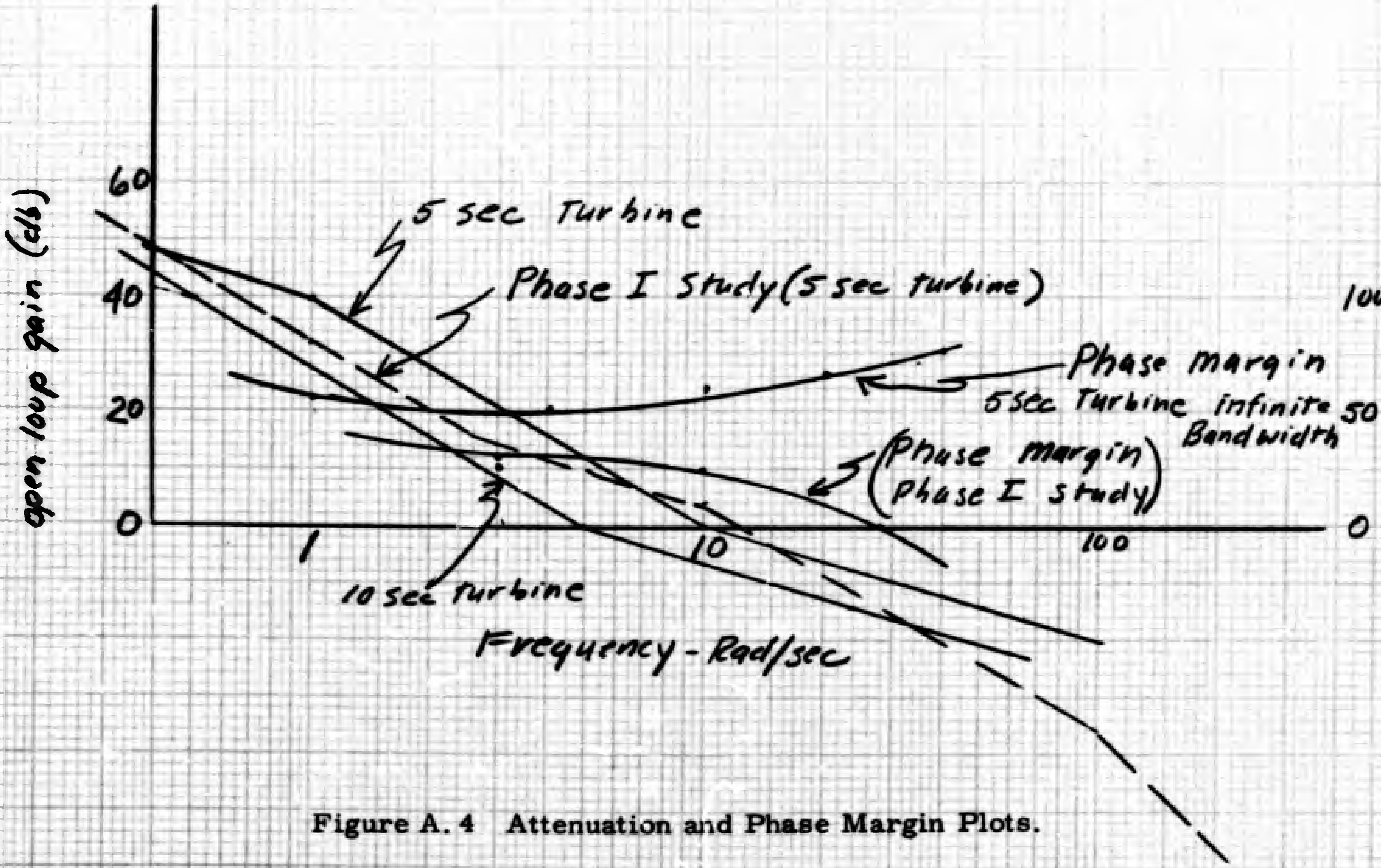


Figure A. 4 Attenuation and Phase Margin Plots.

are also shown. The system which will be mechanized will have a finite reset time constant as contrasted to the Phase I system which assumed a perfect integrator. The recommended system from Phase I has a phase margin at crossover of 25°. The proposed system has a phase margin of 60° at the system crossover assuming infinite bandwidth in all the components. The permissible decrease in phase margin at crossover due to finite component bandwidth is then 35°. This phase shift must be allocated between the steam valve, steam valve actuator stage, the reed discriminator and speed sensor and the fluid amplifiers in the proportional gain path. Typical steam valve fill times will range from 0.02 to 0.04 sec. Using the pessimistic time constant leaves 15° of phase shift to be allocated between the components which will be mechanized. A 5 sec. turbine time constant is used to set the design goals as this is a limiting case.

The phase shift is somewhat arbitrarily allocated as follows:

Reed discriminator and speed sensor - 5° phase shift @ 10 rad/sec.
Closed loop actuator stage - 8° phase shift @ 10 rad/sec.
Proportional amp. - 2° phase shift @ 10 rad/sec.

Actuator Driver Requirements

In the Phase I computer studies a linear system was assumed. No component was permitted to saturate within the demands made by step load changes. In an actual system the actuator for the steam valve and the actuator drive stage parameters must be selected so that the demands of the turbine on step load changes can be satisfied. This condition can be met by specifying a maximum zero to full load stroking time on the actuator.

The actuator drive requirements can be deduced by the use of the simplified block diagram of Figure A.5. This system is similar to the one that will be mechanized. It has a reset time control (t_R) and a lead time constant (t_L) for stabilization. The lag breaks contributed by component bandwidths have been neglected in that they primarily effect system settling time and do not have a strong influence in the maximum transient.

The transient function relating T_o and T_L is:

$$\frac{T_o}{T_L} = \frac{(1 + t_L S)}{\left(\frac{\tau t_R}{K g} S^2 + \left(t_L + \frac{t_R}{Kg}\right) S + 1\right)}$$

The transient solution relation T_o to T_L can be found and the transient solution for error Torque (T_E) is then $T_L - T_o$.

The transient solution for an under damped system is:

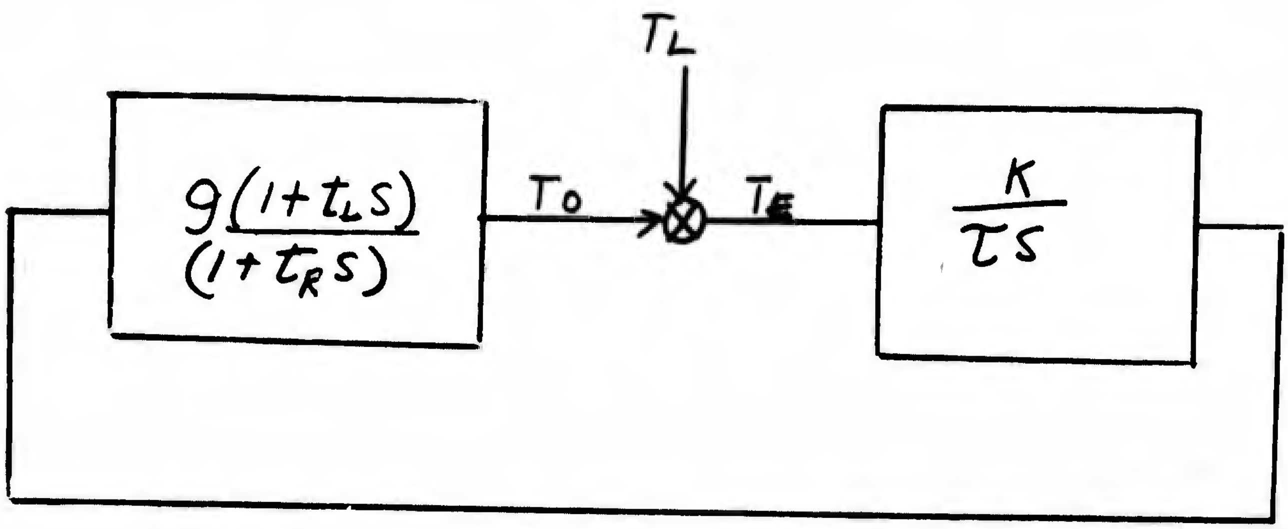
$$T_E = T_L \epsilon^{\left(\frac{t_L gK + \tau}{2\tau t_R}\right) t} \operatorname{Cos}\left(g \frac{\left(\frac{t_L gK + \tau}{g}\right)^2 - \frac{4\tau t_R}{g}}{2\tau t_R}\right) t$$

for a critically damped system.

$$T_E = T_L \epsilon^{-\left(\frac{t_L gK + \tau}{2\tau t_R}\right) t}$$

Using constants from Figure A.4 for a 5 sec. turbine.

$$GK = 500$$
$$t_L = 0.1 \text{ sec.}$$
$$t_R = 1 \text{ sec.}$$
$$\tau = 5 \text{ sec.}$$
$$T_E = T_L \epsilon^{-5.5t} \operatorname{Cos} 8.5t$$



K - Proportional loop gain.

g - Proportional plus reset gain.

t_L - Lead time constant.

t_R - Reset time constant.

- Turbine time constant.

T_L - Load torque.

T_o - Torque applied by control system

T_E - Error torque.

Figure A. 5 Simplified Block Diagram - Speed Control Loop.

The maximum speed transient can be obtained by integrating the error torque from the time equal zero to the time (t_1) where the error torque goes to zero.

$$\text{Speed transient} = \int_0^{t_1} T_E \, dt = T_L \int \epsilon^{-5.5t} \cos 8.5t \, dt$$

t_1 is very nearly equal to $\frac{2\pi}{8.5}\left(\frac{1}{4}\right) = 0.184$ sec.

which after integration yields

$$T_{E_{ave.}} = 0.46 \, T_L$$

As the overall damping factor of the system approaches zero

$$T_{E_{ave}} = \frac{2T_L}{\pi} = .632 \, T_L$$

The maximum speed transient can be found by the product of the average torque and the time of application divided by the turbine time constant. For the specific system considered.

$$\Delta S = \frac{0.46\,(0.184)\,(100)}{5} = \underline{\underline{1.7\%}}$$

The velocity requirements on the actuator can now be estimated. The rate of change of error torque vs. time defines the velocity.

$$S\left(T_E\right) = T_L \, S\left(\epsilon^{-5.5t} \cos 8.5t\right)$$

In general where the damping time constant is equal to or less than $1/w$ the maximum velocity demand will be at time = zero.

The stroking velocity (V_S) of the actuator can then be expressed in terms of the full stroke displacement (D)

$$V_S = \left(\frac{t_L g K + \tau}{2 \tau t_R} \right) D$$

Typical steam valve strokes are on the order of 0.5 in.
Hence, for the 5 sec turbine

$$V_S = 5.5\ (0.5) = 2.75 \text{ in/sec.}$$

and for a 10 sec. turbine time constant

$$V_S = 3.9\ (0.5) = 1.95 \text{ in/sec.}$$