Presented at the 1995 SPIE Conference in Orlando, FL

A Voice Coil Actuated Air Valve for Use In Compressor Forced Response Testing

Duane Mattern
NYMA, Inc.
NASA Lewis Group
Brookpark, OH 44142 USA
mattern@lerc.nasa.gov

A.Karl Owen
U.S. Army Vehicle Propulsion Directorate
NASA Lewis Research Center
Cleveland, OH 44135 USA
fsako@lims01.lerc.nasa.gov

ABSTRACT

A 0-450 Hertz bandwidth, voice coil actuated, proportional sleeve valve is designed to modulate air mass flow by controlling the throat area of a choked flow. The valve was designed to deliver a mass flow of 0.072 kg/s (0.16 lbm/s) with a maximum valve throat area of 41 mm^2 (0.064 $in.^2$), a 689 kPA (100 psid) pressure difference across the valve, and 20^o C, (68^o F) air supply. The valve was developed with inexpensive, off-the-shelve components for use in ground-based forced response testing of compression systems. The design and operation of the valve are discussed and experimental test data of a prototype valve and air injector are compared to a mathematical model. Implementation of a set of 8 of these valves in the compression system of a jet engine is discussed.

Keywords: voice coil actuator, valve, air injector, actuator, control system, forced response testing

1. INTRODUCTION

Compression systems for jet engines have long been vexed by a region of flow uncertainty as shown on a typical compressor map in Figure 1. Current compressor operating strategies avoid this region of operation by providing a stall margin between the surge line and the normal operating limit line. To better understand the fluid dynamics in this region of the compressor map, different models have been developed. These models need to be validated against experimental data to gain confidence in the predictions made by these computer codes. An experimental study of the fluid dynamics in a compression system requires a means by which to perturb the flow. In low speed compressors variable inlet guide vanes, bleed ports, the exhaust nozzle, and fuel flow have been used to excite the system. These actuation methods have bandwidths which are insufficient to excite some of the faster circumferential dynamics in high speed compression systems, (>6000 rpm). A theoretical study of various actuation schemes indicated that a circumferential array of jets performs significantly better than other schemes considered[1]. Such jets would require a high speed valving system in order to synchronize the circumferential modulation of air into a compressor. A brief market search in the 4th quarter of 1993 revealed no commercially available high bandwidth valves for this application*. Thus, the development of a high speed valve for injecting air into a multi-stage compression system was undertaken to provide an enabling technology for high speed compressor research.

In this paper, the design specifications and constraints, trade-offs, and approach are discussed. The valve is described and a mathematical model of the valve is presented. Experimental valve test results are examined and compared to the model predicted data. Finally, the valve development is summarized and some areas for future improvement of the valve are discussed.

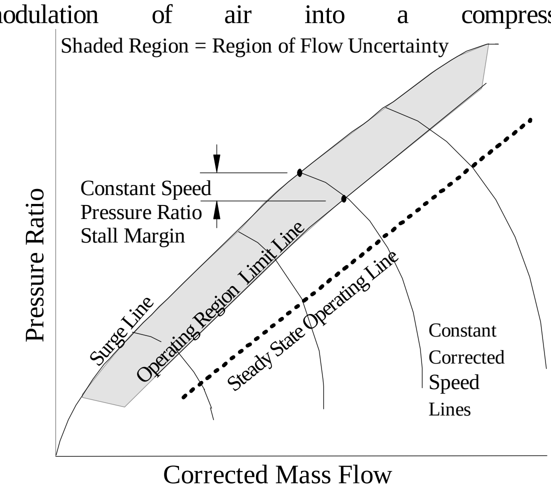

Figure 1 Typical Compressor Map Showing the Region of Flow Uncertainty

* Additional product information has recently been obtained and is discussed in the appendix.

2. DESIGN

2.1. Specifications

The valve was designed for ground-based, forced response testing of an axial compressor in a jet engine. It was desired to excite the first three fluid modes represented by a linear circumferetial modal model by modulating a supplementary mass flow of air into the compressor at 8 uniformly spaced circumferential positions. From previous tests[2,3], it was known that the rotating stall modes would appear at circumferential speeds less than 50% of the rotor speed. The design speed for the test compression system is about 18,000 rpm. Thus, an actuator with a bandwidth of about 18,000/60*0.5*3=450 Hertz was desired. This established the actuator bandwidth requirement for open-loop forced response testing. A higher bandwidth valve is desired if this actuation method is to be used for active stall suppression by controlling the first three fluid modes[4] .

It was desired to keep a short distance from the valve orifice to the compressor interior to avoid the fluid dynamics associated with a column of air at high frequencies. Rough calculations using the resonant frequency of a tube open at both ends yields a maximum length of about 28 cm. (11 in.) from valve orifice to the point of injection to provide a 150 Hertz margin of operation under a corresponding pipe mode at 600 Hz. This distance requirement could best be met by physically locating the valve on the engine casing. During early planning of the engine test it was acknowledged that an engine mechanical pump would have to be relocated to provide access to two ports on the bottom of the engine. Clearance between the engine casing and the relocated pump constrained the valve size to fit in a 7.7cm by 7.7 cm by 12.7 cm volume (3”x3”x5”).

From a previous experiment with air injection jets[5], it was estimated that sufficient excitation authority could be provided with a maximum supplementary mass flow of about 5% of the compressor design mass flow, (11.5 kg/s). Using 8 valves for the forced response test would required each valve to provide a maximum mass flow of 11.5/8*0.05=0.072 kg/s (0.16 lbm/s). The availability of a sufficient supply of compressed air dictated supply conditions at 100 psig and about 68^o F air. Given these conditions and assuming choked flow at the valve orifice it is possible to calculate the theoretical required valve orifice area to provide the maximum mass flow rate[6]. Table 1 summaries the valve design specifications.

Table 1 Summary of Valve Design Specifications

bandwidth	450 Hz	
mass flow rate	0.072 kg/s	0.16 lbm/s
physical size	7.7cm x 7.7cm x 12.7cm	3” x 3” x 5”
supply air pressure	689 kPAg	100 psig
supply air temperature	20^o C	68^o F
maximum orifice area	41 mm^2	0.064 in.^{2}

Early in the design it was decide that a “true” broadband frequency response was desired, which precluded the use of a variable speed motor that could sweep through the various frequencies. This approach was taken to allow this valve concept to be used for closed-loop active surge suppression in future applications.

2.2. Trade-off’s

There were several trade-offs that took place during the valve mechanical design stage. The key difficulty of this design would be obtaining the desired bandwidth. The bandwidth of a positioning system is limited by the maximum possible acceleration. Assuming that a rectilinear (vs rotary) motion would be used, it is possible to calculate the acceleration for a free moving mass during sinusoidal motion. Assume that the position of the valving mechanism can be described by $x=\alpha\sin(\omega t)$ and thus the velocity is $\dot{x}=\alpha\omega\cos(\omega t)$, and acceleration is $\ddot{x}=-\alpha\omega^2\sin(\omega t)$, where α is the maximum displacement and $\omega=2\pi f$, where f is the frequency in Hertz. For an example of the magnitude of the acceleration assume that α=1 mm, then a -3dB response at f=450 Hertz yields $\ddot{x}$=5660 m/s^2. To achieve this acceleration for a 0.01 kg moving mass would require a maximum force, $F=m\ddot{x}_{max}$=56.6 N (12.9 lbf). A example of the required power, P, can be obtained from $P=F\dot{x}=m\ddot{x}\,\dot{x}$. Multiplying the above terms and using a double-angle trigonometry relation results in the power equation, $P=0.5m\alpha^2\omega^3\sin(2\omega t)$. Using

α=1mm, m=0.01kg, and f=450 Hertz yields an estimate for P_{RMS}=80 Watts. This power estimate can be used to size the power supply and examine constraints like voltage and current limits. While these estimates neglect many fine details like friction and suspension stiffness, they do provide an initial means for trading-off the various approaches for obtaining the required orifice area. From the above estimate, it can be observed that while a small mass is important, the displacement plays a larger role in the power requirements. The maximum displacement, force, mass, and stiffness are all interrelated, which complicates the design. A reduction of the mass also impacts the stiffness of the moving mass and associated supporting structure, both of which play an important part on how clearances are maintained for the valving system.

Cost was another major consideration in this design. The goal was to use available off-the-shelve components. It was acknowledge that the off-the-shelve approach would have an impact on reliability, but a long life was not necessary for a possibly one time, ground based test. It was planned that after the successful use of the valve, that design modifications could be made to address reliability issues, (digital electronic and materials selection). This "low-ball" approach precluded the use of some new, but relatively expensive high force actuator made from piezo-strictive and stacked piezo-ceramic materials, and also commercially available voice coil actuators. Cost was also a driver for the position sensing system that was selected. While reliability is critical during an engine test, cost was given more consideration than a long product life.

2.3. Design Approach

Considering the above trade-offs, a voice coil actuator was selected for the linear motor. It would be built with inexpensive, commercially available components (magnet, voice coil, suspension) from the loudspeaker industry. The voice coil actuator would be the spool for modulating the valve orifice area. An easy way to minimize the actuator moving mass while providing the valve throttling was to use the voice coil as the moving sleeve of a sleeve valve. The main issue with this approach is preventing galling from the sliding contact or providing the necessary radial clearance to avoid the high speed sliding contact. Typically, a linear motor of this type would use a set of flexures to provide the radial stiffness that would allow a radial clearance with tight tolerances to be maintained. Alternatively, sliding contact of the moving sleeve was selected to provide the radial positioning of the sleeve. The galling problem caused by the sliding contact of the aluminum sleeve was addressed by using a teflon centerpiece for the sliding surface. This is a low-tech version of the plasma-sprayed metal-glass-fluoride lubricant coating used in reference [7]. To avoid additional mass a noncontact optical position sensing technique was used . The sensor is comprised of a infrared light emitting diode (IRLED) and phototransistor as shown in Figure 2. The sensor has a range of +/-1 mm with a noise level below 4μm. Similar sensing applications for larger ranges using light pipes has been discussed in reference [8].

3. VALVE DESCRIPTION

The valve modulates mass flow by regulating a choked orifice area. The 25.4 mm diameter coil-sleeve provides the valving mechanism by adjusting the exposed area in a set of 2mm high ports in the teflon "arm", as shown in Figure 2. Pressurized air passes through the ports to the center of the teflon arm. The coil-sleeve is position by the force that is induced by an electrical current passing through the voice coil, which is suspended in a permanent magnetic field. The current is provided by a voltage controlled current source using a power operational amplifier, which is commanded by an analog PID position control system. The position of the coil-sleeve is fedback by the optical position sensing system. A high force to mass ratio, (about 35 N to 0.007 kg) enables a sufficiently large acceleration and thus position bandwidth to be achieved.

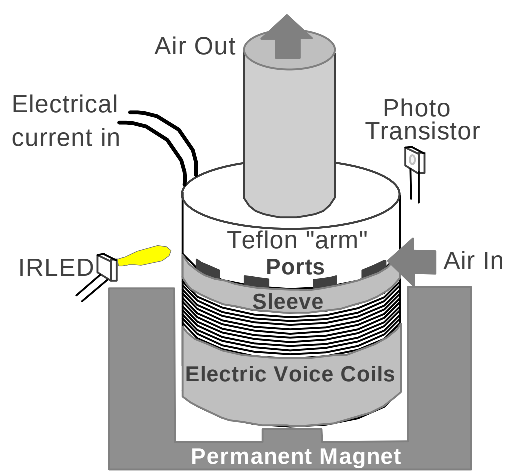

Figure 2 Sleeve Valve Schematic

Figure 3 is a schematic diagram of the completed valve and ejector with a section cut away to reveal the interior of the valve and the air flow path. Air enters the manifold through the supply line inlet area, A_0, (0.5 inch NPT pipe fitting), which is about 3 times larger than the maximum valve orifice area to provide the necessary mass flow without coupling the dynamics to the air supply lines.. The air then passes through the controlled choked orifice area, A_1. A_1 is proportional to the linear position of the sliding sleeve that cover the ports in the center teflon slide. Once the air passes through A_1, it continues through a nearly constant area tube to the ejector exit area A_2. For this valve, A_2 is required to be at least 25% larger than the A_{1_max} in order maintain a linear relationship between coil position and static mass flow.[6] The ejector exit conditions are near the ambient conditions assuming the valve is mounted in the first stage of the compressor .

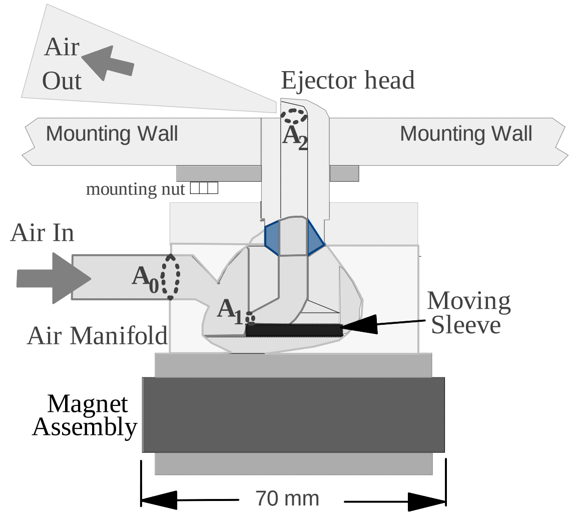

Figure 3 Valve Cutaway Schematic

4. VALVE MODEL

4.1 Static Flow Model

A static flow model was used to predict the static flow characteristics of the valve[6] and will be compared to experimental data in section 5.1. The model is represented schematically in Figure 4. T, P, ρ, and w indicate the temperature, pressure, density, and mass flow rate respectively. The points of interest in the flow are: "1" which denotes the upstream supply conditions, "2" denotes the lumped volume between the supply and the sink (exit), and "3" denotes the downstream exit conditions. Areas A_1 and A_2. are as previously described. The important item to note is the relative size of flow areas A_1 and A_2. As discussed in reference [6], A_2 must larger than A_1 to provide a linear relationship between the orifice area and the mass flow. This maintains a single choking point in the valve flow path. The following static flow modeling assumptions are made:

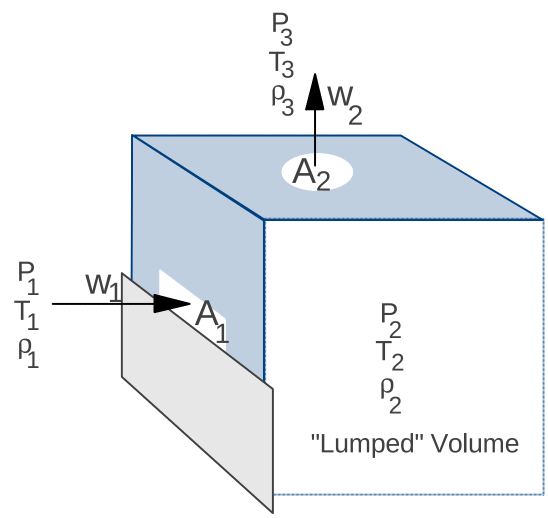

Figure 4 Schematic of Valve Model Geometry

1) A_0 is large enough to provide suffice mass flow to the largest orifice area, A_1.
2) The system is adiabatic, (no heat exchange) and isentropic.
3) The working fluid, air, is calorically perfect, (ideal gas with constant specific heats).

The following equations define the idealized static mass flow, (w_1=w_2), relationship for the valve defined by the geometry shown in Figure 4.

$$PRF_1 = \left[\left(\frac{P_2}{P_1} \right)^{\frac{2}{g}} - \left(\frac{P_2}{P_1} \right)^{\frac{1+g}{g}} \right]^{\frac{1}{2}} \tag{1}$$

$$w_1 = A_1 P_1 \left(\frac{g_c}{R\, T_1} \right)^{\frac{1}{2}} \left(\frac{2g}{(g-1)} \right) * PRF_1 \tag{2}$$

with

PRF_1	= Pressure Ratio Function (nondimensional)
w_1	= mass flow through orifice A_1, (lbm/sec)
A_1	= orifice area, (ft^2)
P_1	= upstream supply pressure, (psfg)
T_1	= upstream supply temperature, (deg R)
P_2	= pressure downstream of orifice, (psfg)
R	= ideal gas constant for air = 53.3 lb_f f / (lbm deg R)
g_c	= English units conversion constant = 32.17 lb_f f / (lbm s^2)
g	= specific heat ratio for air = 1.4

Further details are available in reference [6]. A comparison of the above model equations to experimental data is discussed in section 5.1.

4.2 Dynamic Model

The fluid dynamics are assumed to be small under 600 Hertz, so the linear motor position dynamics dominate the dynamic response. Figure 5 is a block diagram of closed-loop positioning dynamics. The major sections are the second order model for the mechanical voice coil, the electrical voice coil model which is coupled to the voltage controlled current source (VCCS), the first order model for the optical position sensor, and the model of the analog PID control system which includes a derivative filter. The voice coil displacement, x, multiplied by a constant, yields the orifice area of the valve, A_1 in equation 2. Equation 2 can then be used to calculated the valve mass flow given the upstream and downstream conditions. The closed-loop response of the voice coil position model is compared to experimental date in a section 5.2

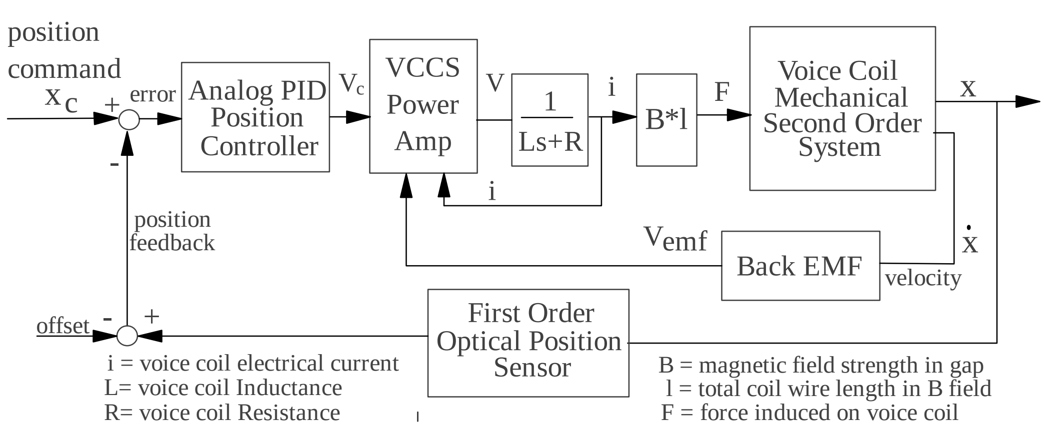

Figure 5 Block Diagram of the Model for the Closed-Loop Voice Coil Positioning System

5. EXPERIMENTAL RESULTS

5.1 Static Flow Measurements

The valve was calibrated for static flow in the Flow Calibration Lab at NASA Lewis Research Center. The mass flow was measured using a calibrated orifice upstream of the valve at several different supply pressures. Figure 6 is a schematic diagram of the flow calibration setup. The supply pressure was measured just upstream of the valve inlet area A_0. The mass flow was measured with the value in it full closed position to get a bypass or leakage mass flow rate. This leakage mass flow was used to estimate a leakage flow area in order to properly compare the calibration data to the static flow

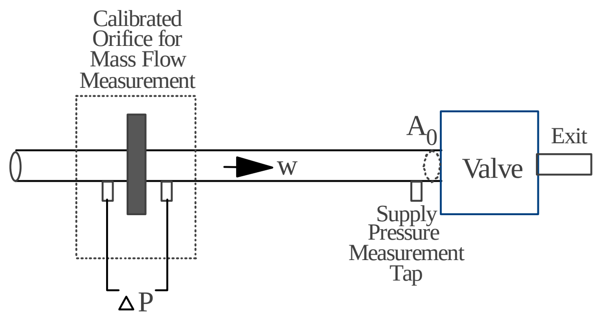

Figure 6 Schematic of Static Flow Calibration

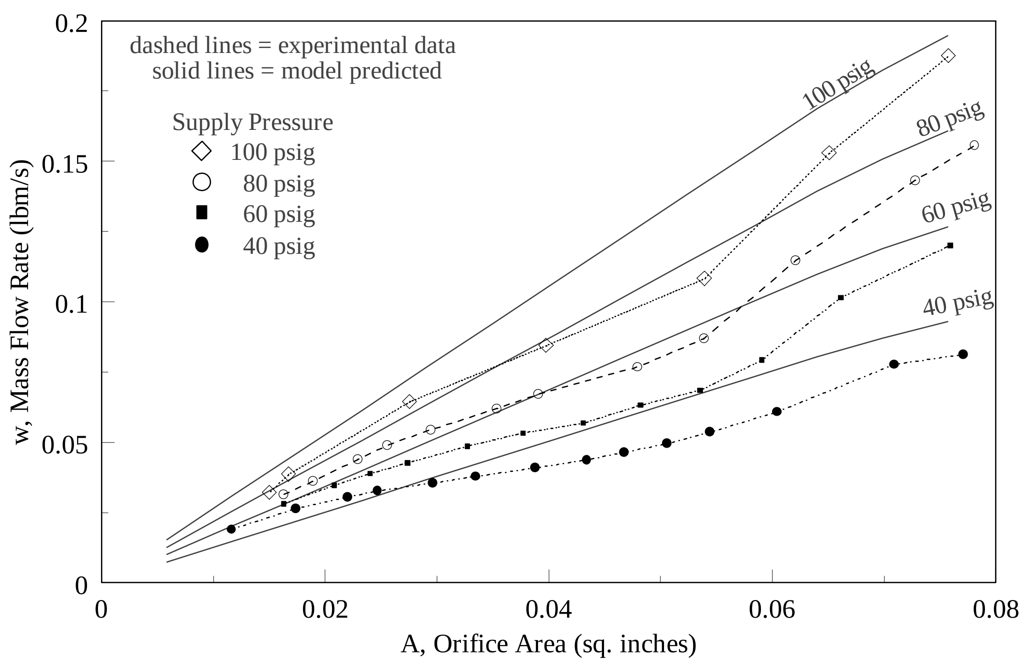

Figure 7 Comparison of Experimental Static Mass Flow Data to Model

model which does not include a leakage model. When the valve is fully closed, it is assumed that P_2=P_3 in equation (1) and equation (2) is used to estimate a leakage area, A_{1_leak} for a measured amount of leakage flow, w_{1_leak}. A_{1_leak} and w_{1_leak} are used as the origin for the model estimated data. Thus the model area ratio and the model predicted mass flow are offset by A_{1_leak} and w_{1_leak} respectively. Figure 7 compares the actual flow calibration data to the model predicted data for supply pressures of 40, 60, 80, and 100 psig, with atmospheric exit conditions and a fixed exit area of 0.116 inches^2, (diameter = 0.385 inches). The static flow is not perfectly linear as all traces have mass flow dip around an orifice area of 0.05 in.^{2}. The cause of this small nonlinearity has not yet been identified. No discharge coefficient was used in the model equation (2). Including a discharge coefficient of 0.9 improves the matching of the model to the experimental data, but the nonlinear dip is still an issue to be resolve. For forced response testing using sinusoidal sweeps, this nonlinearity would introduce additional harmonics at frequency that are integer multiples of the fundamental signal frequency. The small magnitude of these additional harmonics should not be a major concern for compressor forced response testing, but additional test will be made on the valve to isolate the cause of the small dip shown in Figure 7.

5.2 Dynamic Model

The frequency response of the linear model shown in Figure 5 is compared to an experimentally obtained frequency response in figure 8. An two channel analyzer was used to perform sinusoidal sweeps on the closed-loop position system of the valve at various magnitudes. At maximum displacement, (+/- 1mm), the VCCS encounters built-in 3 amp current limits at around 400 Hertz at 100 psig and around 450 Hertz at 80 psig. The circuit still functions properly, but some high frequency harmonics are introduced. The closed-loop frequency response of the valve does not vary significantly for the various supply pressures as long as the current limits are avoided. Additional electrical current is required as the supply pressure increases. The experiment frequency response plotted in Figure 8 is at 50% of the maximum displacement (+/- 0.5mm) with a supply pressure of 80 psig.

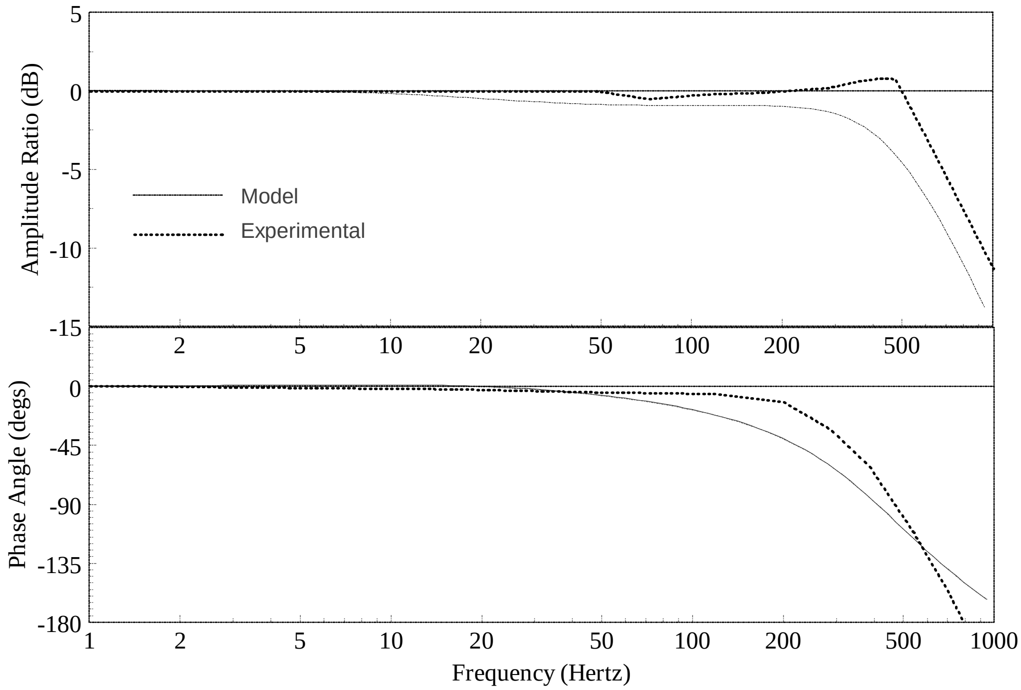

Figure 8 Comparison of Experimentally Obtained Closed Loop Position Frequency Response to Model

Figure 8 shows that the experimental phase angle at 450 Hertz is approximately -45 degrees, but falling rapidly. Also there is a small dip in the experimental amplitude response near the natural frequency of the open loop system. There is a wide discrepancy in the phase angle when compared to the model phase angle, which falls off more slowly. The model is actually 9th order, but most of the dynamics are at higher frequencies. Some additional modifications to the model will be required to obtain a closer comparison to the experimental data.

6. ADDITIONAL WORK

As mentioned above, the current model does not accurately represent the closed-loop system, but it is believed that more data and some minor changes will resolve this difficulty. As discussed early, the nonlinear mass flow dip in Figure 7 needs to be examined in more detail.

Certain desired measurement of the valves performance were not made because suitable facilities were not available to make the measurements. The momentum deficit caused by the ejector head being in the flow and an actual measurement mass flow versus frequency was not done because a suitable facility could not be scheduled in a timely manner. A survey of the ejector flow pattern and the actual measurement of the mass flow of the air out of the valve as a function of frequency will be performed once suitable facilities are located and scheduled.

The valve work that was presented here was initially meant to be a prototype or demonstration of what type of bandwidths are achievable. There are several key areas that could significantly improve the performance of the valve, particularly a study of the heat dissipation of the voice coil and the use of digital electronics and Pulse Width Modulation (PWM). In bench top experiments, coil position bandwidths of 1000 Hertz were obtain by allowing electrical current levels beyond the current 3 amp thermal safety limit for uncooled 28 gauge enamel coated wire. Since the valve is typically used with a DC flow bias, some cooling airflow is always available and higher current levels are possible. Digital electronics would allow for a more robust closed-loop control system (better amplitude and phase margins), when compared to the fixed structure of the analog PID controller. A microcontroller would also allow for automatic position and flow calibration. PWM would reduce the need for a relatively large heat sink to cool the power operational amplifier used to provide the necessary current. All of these modifications would significantly enhance the usefulness of the valve while actually lowering the cost of the unit.

7. CONCLUSIONS

The design of a 0-450 Hertz Bandwidth, voice coil actuated, proportional sleeve valve was presented. The valve was designed to modulate air mass flow for use in compressor forced response testing. The valve design specifications were presented and some of the trade-offs in the design were discussed. Experimental data of the static mass flow and dynamic position bandwidth were compared to experimental data. A nonlinearity in the area to mass flow relationship was shown and this nonlinearity needs further examination. The dynamics frequency response showed that the valve satisfied the positioning bandwidth requirement, but the dynamic model needs to be tuned to better represent the physical system.

8. ACKNOWLEDGMENTS

The authors would like to thank George Readus, Eric Faykus and Mike Krasowski,(NASA), Dan Spina (Calspan), Mark McDaniels (Empirical Sound), and Roland Berndt (MIT: Gas Turbine Lab) for their contribution and assistance towards the development of this valve.

9. APPENDIX

A brief market search in the 4th quarter of 1993 revealed no commercially available high bandwidth valve for this application. NASA Lewis was interested in this type of valve to further the fluid dynamic research of high speed compression system. Besides the onsite contractor work presented in this paper, NASA Lewis funded a university grant with MIT's Gas Turbine Lab. At MIT, Roland Berndt designed a similar valve for a slightly different application using a variable reluctance linear motor and closed-loop positioning system supplied by Moog, Inc.[9] As part of this university grant, MIT is providing 12 valves for forced response testing of a single stage compressor at NASA Lewis Research Center.[10]

Recently it was discovered that an acoustic driver manufactured by Ling Electronics performs similarly to the valve discussed in this paper.[11] The Ling device was designed for acoustic excitation and was used at NASA Lewis to stimulate the harmonics in a circular jet for shear layer experiments.[12] With slight modifications, (including size reduction and closed-loop position control), it is believed that the Ling device could also be used for mass flow excitation. This information is provide for completeness.

10. REFERENCES

1) Hendricks G.J., Gysling D.L., "A Theoretical Study of Sensor-Actuator Schemes for Rotating Stall Control", AIAA 92-3486, presented at the 28th Joint Propulsion Conference, July 6, 1992.
2) Garnier V.H., Epstein A.H., Greitzer E.M., "Rotating Waves as a Stall Inception Indication in Axial Compressors", Journal of Turbomachinery, April 1991, Vol. 113.
3) Owen A.K., "Analysis of Rig Test Data for an Axial Centrifugal Compressor in the 12 Kg/sec Class", presented at the AGARD 82nd PEP, Montreal, Canada, Oct. 4, 1993.
4) Paduano, J., "Active Control of Rotating Stall in Axial Compressors", Gas Turbine Lab Report #208, MIT, March, 1992.
5) Day I.J., "Active Suppression of Rotating Stall and Surge in Axial Compressors, ASME 91-GT-87, presented at the International Gas Turbine and Aeroengine Congress, June 3, 1991.
6) Mattern D., Paxson D., "Static Flow Characteristics of a Mass Flow Injecting Valve", NASA Technical Paper in review, January, 1995.
7) Webb, J.A. Jr., Mehmed O., Lorenzo C.F., "Single-Stage Electrohydraulic Servosystem for Actuating an Airflow Valve with Frequencies to 500 Hertz", NASA TP 1678, 1980, p9.
8) Chin, P. "A New Optical Position Sensor", Tchoupitoulas Research, Sensors Magazine, August, 1993, p39.
9) Berndt R.G., "Actuation for Rotating Stall Control of High Speed Axial Compressors", Master's Thesis, MIT, Feb. 1995.
10) Berndt R.G., Weigl H.J., Paduano J.D., "Experimental Techniques for Actuation, Sensiong, and Identification of Rotating Stall Dynamics in High-Speed Compressors", presented at the 1995 AeroSense Conference, SPIE's International Symposium on Aerospace/Defense Sensing & Control and Dual-Use Photonics, April, 1995.
11) Ling Model EPT-1094 Electro-Pneumatic Transducer Sales Brochure, Ling Electronics, 4890 E. LaPalma Ave., Anaheim, CA 92806 USA, (714)779-1900.
12) Raman G., Rice E., "Axisymmetric Jet Forced by Fundamental and Subharmonic Tones", AIAA Journal, Vol. 29, No. 7, July, 1991, p 1114, presented at the AIAA 2nd Shear Flow Conference, Tempe, AZ, 1989.