AD NUMBER
ADA239595
LIMITATION CHANGES
TO: **Approved for public release; distribution is unlimited.**
FROM: **Distribution authorized to U.S. Gov't. agencies only; Proprietary Information; JUN 1988. Other requests shall be referred to Arnold Engineering Development Center, Arnold AFB, TN.**
AUTHORITY
doaf/hqaedc/afsc ltr dtd 12 Aug 1991

AEDC-TR-88-16 ~~PROPRIETARY~~

C-3

Development of a Dynamic Pressure Response Calibrator

Kevin E. Smith
CSA Engineering, Inc.
Palo Alto, CA 94306-4682

June 1988

Final Report for Period August 1987 – February 1988

Distribution authorized to U. S. Government agencies only; proprietary information; June 1988. Other requests for this document shall be referred to Arnold Engineering Development Center/DOCS, Arnold Air Force Base, TN 37389-5000.

WARNING

This document contains technical data whose export is restricted by the Arms Export Control Act (Title 22, U.S.C., Sec 2751 et seq.) or The Export Administration Act of 1979, as amended. Title 50, U.S.C., App. 2401, et seq. Violations of these export laws are subject to severe criminal penalties. Disseminate in accordance with the provisions of AFR 80-34.

ARNOLD ENGINEERING DEVELOPMENT CENTER
ARNOLD AIR FORCE BASE, TENNESSEE
AIR FORCE SYSTEMS COMMAND
UNITED STATES AIR FORCE

~~PROPRIETARY~~

NOTICES

DESTRUCTION NOTICE

APPROVAL STATEMENT

This report has been reviewed and approved.

Marjorie S. Collier

MARJORIE S. COLLIER
Directorate of Technology
Deputy for Operations

Approved for publication:

FOR THE COMMANDER

Marion L. Laster

MARION L. LASTER
Technical Director
Directorate of Technology
Deputy for Operations

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE

REPORT DOCUMENTATION PAGE

Form Approved OMB No. 0704-0188

1a REPORT SECURITY CLASSIFICATION: Unclassified	1b RESTRICTIVE MARKINGS
2a. SECURITY CLASSIFICATION AUTHORITY	3. DISTRIBUTION/AVAILABILITY OF REPORT: Distribution authorized to U.S. Government agencies only; proprietary information; June 1988. Other requests for this document shall
2b. DECLASSIFICATION/DOWNGRADING SCHEDULE	
4. PERFORMING ORGANIZATION REPORT NUMBER(S): AEDC-TR-88-16	5. MONITORING ORGANIZATION REPORT NUMBER(S)
6a. NAME OF PERFORMING ORGANIZATION: CSA Engineering, Inc. / 6b OFFICE SYMBOL (If applicable)	7a. NAME OF MONITORING ORGANIZATION
6c. ADDRESS (City, State, and ZIP Code): 560 San Antonio Road, Suite 101, Palo Alto, CA 94306-4682	7b ADDRESS (City, State, and ZIP Code)
8a. NAME OF FUNDING/SPONSORING ORGANIZATION: Arnold Engineering Development Center / 8b. OFFICE SYMBOL (If applicable): DOT	9 PROCUREMENT INSTRUMENT IDENTIFICATION NUMBER: F40600-87-C-0008
8c. ADDRESS (City, State, and ZIP Code): Air Force Systems Command, Arnold Air Force Base, TN 37389-5000	10. SOURCE OF FUNDING NUMBERS: PROGRAM ELEMENT NO 65502F; PROJECT NO; TASK NO; WORK UNIT ACCESSION NO

11. TITLE (Include Security Classification): Development of a Dynamic Pressure Calibrator (U)

12. PERSONAL AUTHOR(S): Smith, Kevin E., CSA Engineering, Inc.

13a. TYPE OF REPORT	13b TIME COVERED	14. DATE OF REPORT (Year, Month, Day)	15. PAGE COUNT
Final	FROM 8/87 TO 2/88	June 1988	41

16. SUPPLEMENTARY NOTATION

17. COSATI CODES			18. SUBJECT TERMS (Continue on reverse if necessary and identify by block number)
FIELD	GROUP	SUB-GROUP	pressure sensor calibrator; swept-sine excitation; servo-controller; rocket motor

19 ABSTRACT (Continue on reverse if necessary and identify by block number)

Measurements of dynamic acoustic pressures in turbine and rocket engines are often made by separating the transducing element from the measurement point with tubing. The tubing is used for thermal isolation and/or when there are size constraints. The acoustic properties of the tubes require that the tubing and sensor unit be calibrated together as a system.

A prototype calibration system has been designed and implemented which is capable of providing a controlled input pressure to the tubing and sensor pair. The specifications require input pressure spanning a frequency range of 2-500 Hz, a level of between 1 psi (171 dB re 0.0002 μ-bar) and 0.001 psi (100 dB), with tubing volumes as large as 10 in.^{3}. Input excitation is either swept-sine or random. The entire range of specifications was met for the swept-sine input (except the very lowest pressure setting). The random excitation input never met the specifications for the highest pressure range.

20 DISTRIBUTION/AVAILABILITY OF ABSTRACT: ☐ UNCLASSIFIED/UNLIMITED ☒ SAME AS RPT ☐ DTIC USERS	21 ABSTRACT SECURITY CLASSIFICATION: Unclassified
22a. NAME OF RESPONSIBLE INDIVIDUAL: Carlton L. Garner	22b TELEPHONE (Include Area Code): (615) 454-7813 / 22c OFFICE SYMBOL: DOCS

DD Form 1473, JUN 86 Previous editions are obsolete.

SECURITY CLASSIFICATION OF THIS PAGE

UNCLASSIFIED

3. DISTRIBUTION/AVAILABILITY OF REPORT. Concluded.

be referred to Arnold Engineering Development Center/DOCS, Arnold Air Force Base, TN 37389-5000.

19. ABSTRACT. Concluded.

The system utilizes a choice of driver configurations (high frequency compression driver or shaker and piston). The precise control of the input is achieved by careful design and sizing of the input cavity and placement of the reference sensor and a digital servo-controller. The design was carried out by a novel hybrid modelling technique using finite element models of the acoustic cavity and combining these with an admittance model of the driver.

PREFACE

The work reported herein was conducted by CSA Engineering, Inc., Palo Alto, CA, at the request of the Directorate of Technology, Arnold Engineering Development Center (AEDC), Arnold Air Force Base, Tennessee, under contract number F40600-87-C0008. The Air Force Project Manager was Ms. Marjorie S. Collier; the CSA Engineering, Inc. Project Manager was Mr. Kevin E. Smith. The reproducibles used in the reproduction of this report were supplied by the authors.

CONTENTS

ILLUSTRATIONS

Figure

TABLES

1. INTRODUCTION

Arnold AFS engineers make extensive use of pressure sensor/tubing configurations in its jet and rocket engine development programs. Each testing situation may call for numerous new and unique tubing configurations, therefore, necessitating several calibrations. As such, there is no *typical* configuration and thus, no *a priori* calibration.

Currently, there are no commercial calibration systems capable of calibrating these sensors over the entire wide range of frequencies and pressures that are required. Arnold engineers must make do with partial calibrations, i.e., calibrations covering only a portion of the frequency and pressure range that the sensor will see in service.

2. BACKGROUND

In all phases of development testing, qualification, and into production, jet turbine and rocket engines must be instrumented with dynamic pressure transducers. These sensors are installed at many locations; pump inlet/outlet pressures, radial and axial flow seals, combustion chambers, and nozzles. They are installed for many purposes: feedback sensors for pump shaft speed controllers; thrust measurement; and many "red-line" functions, such as overpressure indicators for solid rocket motors, secondary indicators for turbine over-speed, component failure, and studying or monitoring unsteady combustion.

These sensors are often removed from the measurement point by lengths of tubing, ducts in castings, or hollow rotating shafts. There are two major reasons for these configurations: thermal isolation and space constraints. Placing a precision pressure sensor in an exit nozzle or in a combustion chamber would lead to rapid failure. The high temperatures of combustion and the corrosive nature of these gases (especially in solid-rocket motors) requires that the sensor be physically removed from them. The pressure at the measurement surface must be conveyed to the sensor with a duct of some type. Typically there is little or no mean flow inside the duct and the duct may have a heat sink or exchanger attached to it in order to eliminate conducted heat from reaching the sensor.

In some applications a pressure measurement must be made at a point where it would be difficult or impossible to place a sensor without extensive changes to the motor to accommodate the size of the sensor. Measuring pressure near a radial seal might require complete redesign of the bearing/seal/housing arrangement. But a small tube may be inserted into the housing and extended to the surface where pressure is to be measured. The tube could then be connected to the pressure transducer and the measurement made.

Since the tubing has acoustic properties of its own, these arrangements must be designed and implemented with care. A tube with one closed and one open end will behave like a resonator. If the open end is connected to a small chamber, this chamber is known as a Helmholtz resonator or filter. If the pressure quantity to be measured has a significant component at this filter's notch frequency, this is clearly a poor design. The manner in which the tube is terminated at both ends is also very important. Small changes in diameter at mating joints will produce reflections and will have distinct filtering characteristics. Sharp bends, tees, branches, and other physical changes in the shape or diameter of the tube will result in complex and potentially undesirable characteristics. Tubes that have compliant walls may be analyzed, but detailed knowledge of the material properties of the wall are required. This is generally an undesirable situation. Even a perfectly straight tube will have a fundamental resonance with overtones consisting of all odd harmonics.

Clearly, the tubing, connectors/adaptors, and sensor must be calibrated together as a system. If circumstances force a tubing/sensor combination that has a notch or low-pass filter characteristic, then measurement bandwidth must be restricted to be well below the limiting frequency.

The calibration curve may be used not only as a means of determining sensor characteristics but directly as a gain/phase compensation curve if spectral content of the sensor signal is known. Some sophisticated controllers use this approach to extend the usable bandwidth of a sensor. This is the basis of the random-vibration servo-controller.

Many calibration techniques have evolved which cover various types of pressure sensors, pressure ranges, bandwidth, accuracy, speed of calibration, cost of the calibration equipment, and skill level of the operator. The combination of performance factors called for in this project place the calibrator outside the bounds of typical and most specialized calibrators. For instance, the required bandwidth of 2–500 Hz is easily achieved with many devices, e.g., a piston phone. But the ability to achieve this bandwidth at 1 psi (171 dB re .0002 μ-Bar) with a random pressure field and good phase accuracy is beyond the capability of standard devices.

Many calibrators rely on the geometry, i.e., analytical knowledge of the acoustic properties of the calibrator to produce known (but not controlled) acoustic fields. Explicit in most of these devices is knowledge of the (unchanging) geometry of the sensor under calibration. The sensor is usually assumed to have an input impedance that is very large compared to the source impedance. None of these assumptions hold in this instance. The geometry of the tubing inlet/coupler is variable by definition and the input impedance is not necessarily large if the tube has acoustic resonances in or near the measurement bandwidth.

3. PHASE I TECHNICAL OBJECTIVES

The objective of the Phase I work was to determine if a device could be constructed that could calibrate tubing/sensor combinations of specified type and range over a specified range of frequencies and pressure levels. The tubing specifications were

- Tubing inside diameter up to 1/4" (with 1/4" being typical).
- Tubing lengths to 36" (with 24" being typical).
- Tubing volumes of up to 10 in^3 (1.2 in^3 being typical).

Pressure and frequency specifications were

- Calibration range from 0.001 psi (111 dB SPL) to 1.0 psi (171 dB SPL) with input amplitude flat to ± 0.5 dB being preferred with ± 1.0 dB being acceptable.
- 2–500 Hz sine or random excitation (no specification on random level).

Output from the system is to be an *equalization* curve (magnitude and phase) relating pressure response at the sensor-under-test to the reference sensor at the tubing inlet.

If such a system is possible, detail a plan to implement this system into an integrated package suitable for use by calibration engineers (Phase II).

4. APPROACH & METHOD

The basic concept used in this project is shown schematically in Figure 1. The servo-controller monitors the reference sensor signal (pressure applied to the tube inlet). The reference sensor is placed as close as is possible to the tube inlet. The spectral content of this signal is compared to a target spectrum which has been entered into the servo-controller's memory. If the two spectrums do not match, the servo-controller will construct a waveform based on the difference between the measured and target spectrum. This waveform is then input to a power amplifier connected to the driver. In this way, a pre-selected spectrum may be maintained in the calibrator cavity. The target spectrum will, for calibration, usually be flat over the frequency range of interest.

The key elements which set performance limits are

Driver The driver has the greatest effect on bandwidth and ultimate pressure that can be achieved. The mass and stiffness of the armature set the bandwidth and the stroke sets the lower frequency limit.

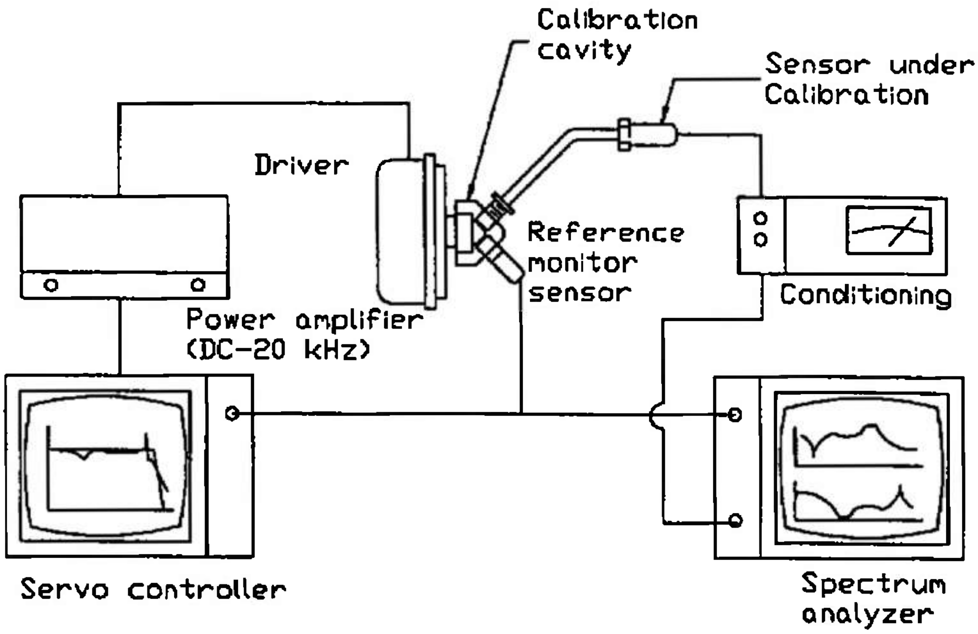

Figure 1. Dynamic Pressure Calibrator.

Cavity The shape and volume of the cavity interacts with both the driver and the tube. The reference sensor must be carefully placed in the cavity in order to accurately sense the input to the tube.

Servo-Controller The servo-controller must rapidly respond to the changing system response and provide an updated drive signal. The servo-controller must have sufficient dynamic range to compensate for peak responses at resonances and minimum response at anti-resonances.

The methodology used is shown in Figure 2. Specified quantities (e.g., tube length and volume), results from tests of candidate components, and an initial cavity design are fed into the finite element (FE) model. The results of the FE analysis indicate performance and changes in shape and size for the assumed cavity design. The analysis cycle iterates until a proper system is achieved. A performance prediction is made and the system is implemented and carried through to full testing. At the end of the testing comparisons are made with the predictions. Discrepancies between the two indicate improvements in either analysis, component selection, and/or testing.

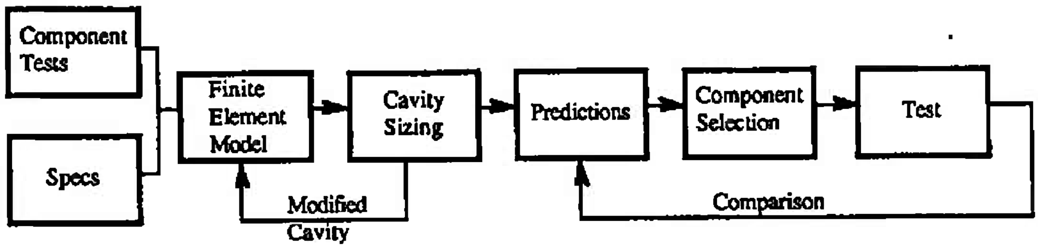

Figure 2. Project methodology.

4.1 Driver Requirements

The large-volume, high-pressure, and low-frequency requirements dictate that the calibration driver–cavity pair must provide a large volume rate. The high-frequency requirement dictates a high-volume acceleration. Designing a driver–cavity pair that meets all of these requirements is clearly complex.

An ideal driver for this problem would consist of an electrodynamic shaker coupled to a piston. This would allow the generation of large stroke at low frequency. The shaker has a large effective area over which the electromagnetic field can generate a force on the armature. The armature has a rugged flexure support to handle and guide the excitation. Electrodynamic shakers are available in a somewhat limited load–frequency range.

The main drawback to the shaker as a driver is a lack of high-frequency response. At high-frequencies, most of the electromagnetic force generated by the magnetic field and armature-current, goes into accelerating the relatively massive armature. There is little force left over to drive the system attached to the shaker head. The upper frequency limit of the shaker is set by the mass of the shaker armature and flexures and the current handling ability of the shaker (or current delivering capability of the amplifier).

A simpler setup involves the use of an electromagnetic compression driver. The compression driver has a fairly rugged suspension but typically has a small voice-coil (compression drivers usually are intended for high-frequency applications). Modern compression drivers are efficient (typically 2-3 times as efficient as a shaker) and have good power (cooling) handling. A wide range of compression drivers are commercially available.

4.2 Cavity Design

The calibration cavity must effectively couple the pressure generated by the driver to the tube inlet. It must also provide access for the reference sensor. The design of the cavity is critical in that the reference sensor must accurately measure the tube inlet pressure *and* provide the feedback signal to the servo-controller. If the reference sensor is placed at or near an anti-resonance of the combined system (tube and cavity) then it will not be capable of providing a suitable control signal to the servo-controller. (In control system language such a system is said to be *unobservable*). Therefore, the cavity must be sized and shaped in such a way as to make the reference sensor position capable of always providing a usable control signal.

4.3 Analysis

The number of components (and their interrelatedness) required for this pressure calibrator makes a prototyping approach unattractive. What is needed (and is the approach taken) is a model of the complete system capable of producing performance predictions of sufficient accuracy that a prototype can be built with a good chance of success. The model need not predict the exact behavior of the system (particularly at the very extremes of volume, pressure, or frequency) but indicate whether a specific configuration will work over the target performance range.

To produce realistic predictions each of the system's components must be modeled and combined into an overall model. Components which require analysis include

- Driver—Calibration and/or mass, stiffness, and damping.
- Cavity—Wall compliance (if any) and acoustic properties.
- Tube—Wall compliance (if any) and acoustic properties.

(Note that *performance*, as used here, relates to open-loop behavior. All analytical characterizations assume open-loop behavior.)

Standard finite element (FE) modeling techniques and programs may be adapted to analyze acoustical systems. Analogies between stiffness and bulk modulus, structural density and fluid density, and boundary conditions (applied acceleration and applied pressure) allow design of acoustical enclosures using structural analysis tools.[1]

[1] "MSC/NASTRAN Handbook for Dynamic Analysis," MacNeal-Schwendler Corporation

4.3.1 Finite Element Model

The mechanical analogy for modeling acoustical systems can be carried to a high level of complexity. For instance, an acoustical cavity interacting with a mechanical structure having flexible walls can be modeled, albeit with considerable effort. At the simplest level an acoustical cavity with rigid walls and openings into an infinite (non-reflecting) medium is quite simple. The rigid walls have zero-displacements in the FE model and the openings are free boundaries (equivalent to zero-pressure).

Because of the intricacies involved in using the FE technique in this manner a number of increasingly complex models were used to verify procedures. All of these models possessed closed-form solutions that could be used to verify the results. For instance, the "organ pipe" problem (one closed end, one open end) has a simple solution for its natural frequencies

$$f_n = \frac{(2n-1)c}{4L}, \quad \text{for } n = 1, 2, 3, \cdots \tag{1}$$

where

c = speed of sound in the fluid

L = length of the tube

n = mode number

The steps in the verification process were to:

1. Calculate the admittance function (pressure–velocity) for an infinite tube (one end is a piston) where the piston is forced to execute a given sinusoidal velocity. This problem has a closed-form solution. Using this admittance, calculate the piston displacement at 2 Hz for an RMS pressure of 1 psi. This is a worst case estimate of driver displacement. Then calculate the piston acceleration at 500 Hz for an RMS pressure of 1 psi. This places an upper bound on allowable piston mass (knowing the maximum force the driver can generate).
2. Model, with the finite element technique, a "long" tube with a piston at one end and the other end closed. The tube should be long enough so that its admittance function approaches the infinite tube case. This model has most of the features of the ultimate model and allows a check against the closed-form problem (a very desirable comparison).
3. The last problem is to model the piston as a mass and spring, where the spring represents the suspension of the driver. Once this model is working it can be modified to include various tubing configurations. By including the mass of the voice-coil explicitly, direct estimates of the drive force can be made. At this point a number of favorable trade-offs can be made. For instance, if the driver can deliver sufficient force, the voice-coil can be made heavier. This

is desirable since the higher mass will reduce the dynamic range of the drive admittance. This will lessen the dynamic range the servo-controller will need to equalize. In essence, by shifting some of the burden to the driver, the servo-controller can be unloaded.

The verification process proved to be somewhat tedious but ultimately gave good results that allowed the more complex models to be constructed. For example, for a simple closed-closed tube the predicted and theoretical natural frequencies are shown in Table 1.

Mode	Theory	Predicted
1	216.8	217.6
2	428.3	435.2

Table 1. Accuracy of finite element model.

A computer program was written which automatically generated the cavity and tube FE data. Many different configurations were analyzed and verified.

4.3.2 Example Finite Element Model

A detailed look at a FE model is given here. The model (shown in Figure 3) is of the *typical* configuration (described above). The 24"-long, 1/4"-diameter tube terminates into a closed 0.1-in^3 volume at one end and a 1.25"×2" long cavity at the other end.

In the FE model each structural grid point is reduced to having a single degree-of-freedom (DOF). The analog is displacement to pressure; a structural point may have different displacements in different directions but pressure is a scalar field. This reduces the model size considerably. Spatial resolution is governed by the wavespeed of the fluid medium. As mentioned above an open surface (zero pressure differential) corresponds to a fixed displacement and a zero particle displacement in acoustics (fixed boundary) is a free FE boundary. Material density of the fluid is replaced with the reciprocal of its bulk modulus and the elastic stiffness of the fluid is replaced with the reciprocal of the density.

The 601 grid point model is analyzed using standard eigensolvers with results requiring careful interpretation. For instance, to arrive at particle acceleration (as a result) stress must be calculated. (Double time derivative quantities are area products.)

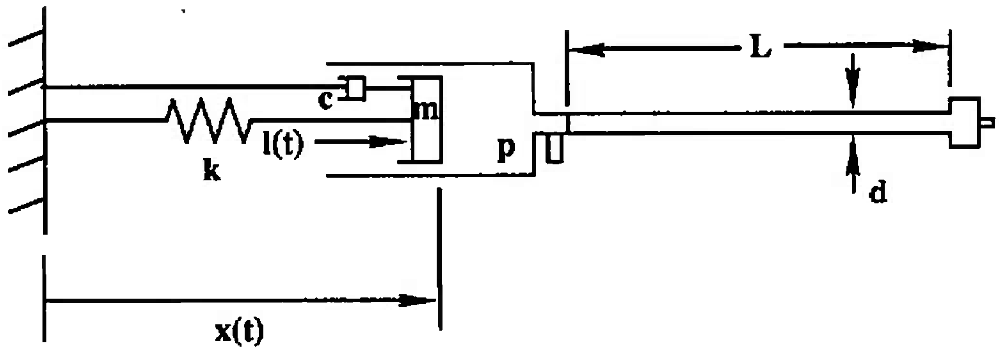

Figure 3. Verification model—24" tube with 1.25"×2.0" cavity.

4.3.3 Extension to a Rigid Moveable Wall

The case of interest is a rigid tube with a rigid piston connected to ground through a spring. The FE model becomes considerably more difficult when we attempt to couple actual structural degrees-of-freedom (DOF's) to the analog acoustical DOF's: fictitious intermediate connection DOF's are required. In addition, there are no closed-form solutions available to verify the technique.

Instead of pushing the FE model to model every aspect of the system we attempted to combine the FE model of the acoustic cavity with another modeling technique known as admittance modeling.[2]

4.4 Admittance Model (combined method)

Figure 4 shows schematically how the admittance model is set up. The acoustic cavity is represented solely as a pressure quantity. As such, the modal data of the finite element model of the isolated cavity is used to calculate a pressure on the piston (or at any other point).

We present a short derivation of the combined model here. Summing forces on the piston

$$m\ddot{x} = -kx - c\dot{x} - ps + l \tag{2}$$

[2]Admittance modeling is a technique pioneered largely by CSA. The technique is described in detail in "Introduction to Admittance Modeling," CSA Engineering Report No. 87-06-01, June 1987., Smith, K.E. and Kienholz, D.A.

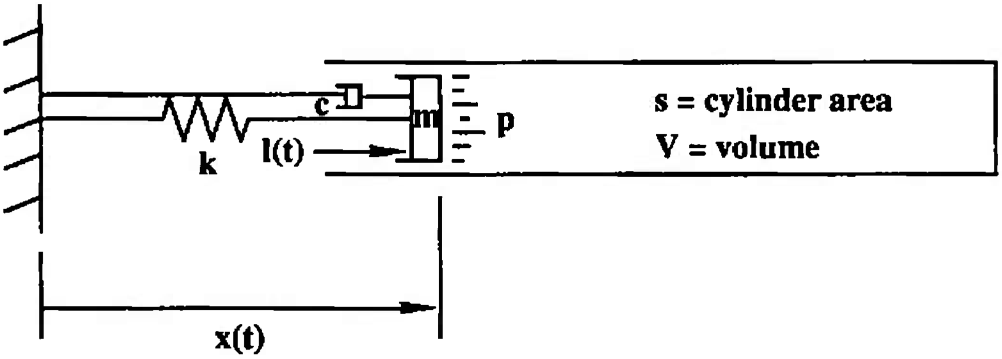

Figure 4. Admittance model of cavity and driver.

Taking the Fourier transform of this equation yields

$$-m\omega^2 X = -kX + c\omega X - Ps + L \tag{3}$$

and dividing by L gives

$$-m\omega^2\frac{X}{L} = -k\frac{X}{L} + c\frac{X}{L} - s\frac{P}{L} + 1 \tag{4}$$

Now expand Eq. 4 using the identity

$$\frac{P}{L} = \frac{P}{X}\frac{X}{L} \tag{5}$$

Since pressure p is caused entirely by x, the ratio P/X is a frequency response function (FRF), which can be calculated entirely from normal mode properties.

Now we define the impedance of the piston as

$$z \triangleq -m\omega^2 + k - c\omega \tag{6}$$

and

$$H_1 \triangleq \frac{P}{X} \tag{7}$$

and

$$H_2 = \frac{1}{-s\omega^2}H_1 \tag{8}$$

Rearranging and solving for P/L we get

$$\frac{P}{L} = \frac{-s\omega^2 H_2}{z - s^2\omega^2 H_2} \tag{9}$$

That is, the pressure P, generated at the piston surface due to a load L on the piston looks like a force divider. It has units of $1/in^2$.

This expression is very simple and is very easy to solve in CSA's admittance modeling software. The only unusual component to this equation is H_2 which is computed from the normal modes of the acoustic cavity.[3]

5. COMPONENT TESTS

The testing aspects of this work can be broken into component testing and overall system testing.

5.1 Component Testing

Dynamic tests were performed on a number of components. Most of the testing was performed to provide directly usable information for the performance models.

5.1.1 Compression Driver

Compression drivers are typically used in conjunction with an exponential-horn. These drivers produce a high-pressure in the throat of the horn and the horn then converts this pressure into a plane-wave at the mouth. It is this ability to produce a high-pressure that spurred interest in using them in this project. Along with their pressure developing character, compression drivers have very low-weight voice-coils and diaphragms. This makes them ideal for high-frequency applications. However, at low frequencies these drivers develop small displacements—due to their stiff suspensions. No data were available that indicate how these drivers would perform in this application.

In order to determine if a compression driver would be suitable, a series of tests were performed on a representative driver. An Emilar EC 320A driver (see Figure 5) was chosen for the tests. This driver will dissipate 120 W and has a 3-inch diameter diaphragm.[4]

The driver cavity was capped (natural volume of ≈ 1.8 in^3) and a noncontacting displacement transducer was installed to sense displacement of the diaphragm. A current-sensing circuit was used to measure input current to the driver. The input

[3]It is interesting to note that the cavity is modeled as a free-free structure. Thus, there is a rigid body mode. This rigid body mode is extremely important. It relates pressure change due to an isentropic compression. This bit of thermodynamics actually dominates the response of this system at low frequencies and pressures.

[4]Application engineers at Emilar felt that there was a good chance that this driver would generate the pressures we were interested in.

Figure 5. Emilar compression driver: driver (top) and diaphragm (bottom).

signal was band-limited white-noise—2–500 Hz. The resulting frequency response function was flat in the range of interest with a value of 1.5×10^{-3}-inches^2 per ampere. A peak-current of 20 amperes (our laboratory Crown DC-300A) would produce 0.04 inches peak displacement. This translates into a 0.28-in^3 volume change or a maximum volume of 3.92 in^3 into which this driver could achieve a 1 psi pressure. This is not sufficient to meet the 10-in^3 specification at lower frequency. However, at higher frequency the displacement limit is not an issue—acceleration is. The compression driver was a good candidate for the higher frequency regime.

5.1.2 Electrodynamic Shaker

The low-frequency regime is essentially displacement limited. Electrodynamic shakers typically can develop high displacements at low frequencies but not high accelerations at high frequencies. A shaker would be paired with a suitable piston, cylinder, and pushrod.

CSA's VTS (Vibration Test Systems) shaker was used for analysis and test. The shaker has a low-impedance copper armature with a linear bearing aligning one end with a beryllium-copper-viscoelastic flexure at the other end. The shaker is conservatively rated as generating 100 lbs force. The standard armature weighs 3/4 lb. A lightweight aluminum armature is available (1/2 lb.) but this armature has a higher impedance. Thus, with the same amplifier, this armature will not produce a higher peak acceleration at high frequencies. Custom wound low impedance armatures and matched amplifiers are available.

Hand-calculations produced upper-bounds values for peak displacement and acceleration that a driving "piston" connected to the shaker would need to achieve. Basically, the maximum stroke of the piston is such that a flexure-type seal between the piston and cylinder wall is unacceptable. (In addition, the dynamic stiffness of a seal would require an even higher peak force from the driver.) The relative displacement along the axis of the cylinder is so large that the sealing mechanism must follow the piston.

The Airpot actuator is a low leakage, low mass, low friction matched piston-cylinder pair. These actuators consist of a graphite piston in a glass cylinder. The sealing consists of extremely close tolerance fitting of the piston and cylinder—losses < 0.05in^3/sec/psi are typical. Integrating this loss over one cycle at 2 Hz (peak leakage frequency) shows negligible increase in needed peak displacement. Peak friction force might be one-eighth pound or less.

The Airpot actuator comes in a variety of sizes and with a variety of pushrods and connectors. The actuator chosen was 1.256" in diameter with a 3"-long cylinder. This actuator is rated for a maximum working pressure of 100 psi. This Airpot was modified to have a larger opening at the top of the cylinder. This opening was based on the FE model where the cavity length was equal to about 3 diameters of the reference sensor diameter.

6. INSTRUMENTATION

There are three main components making up the instrumentation set for the calibrator

1. The reference pressure sensor and sensor-under-test.
2. The signal conditioning for these sensors.
3. The servo-controller.

The reference sensor and its conditioner must possess the required dynamic range, sensitivity, and noise characteristics of a true reference. In addition, the signal created by the sensor must be measured with considerable accuracy. In this case the servo-controller serves as both the spectral/pressure control and the final signal measuring device.

6.1 Reference Sensor Requirements

The calibrator incorporates a reference sensor as the reference pressure device and as the sensing device in the feedback path to the servo-controller. The dynamic range requirement of the calibrator extends from 0.001 psi (110 dB re 0.0002 μ-bar) to 1.0 psi (171 dB). Realistically, the reference sensor should possess a noise floor of about 70 dB (40 dB signal-to-noise ratio is considered adequate for calibration of pressure devices). Some amount of head-room is also desirable. Therefore, a reference sensor must possess a dynamic range in excess of 100 dB *and* have a noise floor at 70 dB or lower. We were not able to find a single device that could cover this entire range.

For instance, the Endevco 8550M1 piezoresistive pressure sensor's working range is imposed on the target range in Figure 6. The device has adequate dynamic range, sensitivity (133 mV/psi at 10 VDC excitation), and head-room but its noise floor is rated at 80 dB (in fact it is actually lower than this but Endevco rates the noise spec very conservatively). This device comes closest to covering the entire dynamic range but just misses (Kulite and PCB devices were examined but neither company offers a device with the performance of the Endevco). The conclusion is that a single reference device can not be used to cover the entire range of calibration pressures. However, since the higher pressure range is of most interest, the Endevco device is an excellent reference for most of the range of interest. This means that a minimum of swapping references will be required. The lowest pressure range must be covered by a microphone device.

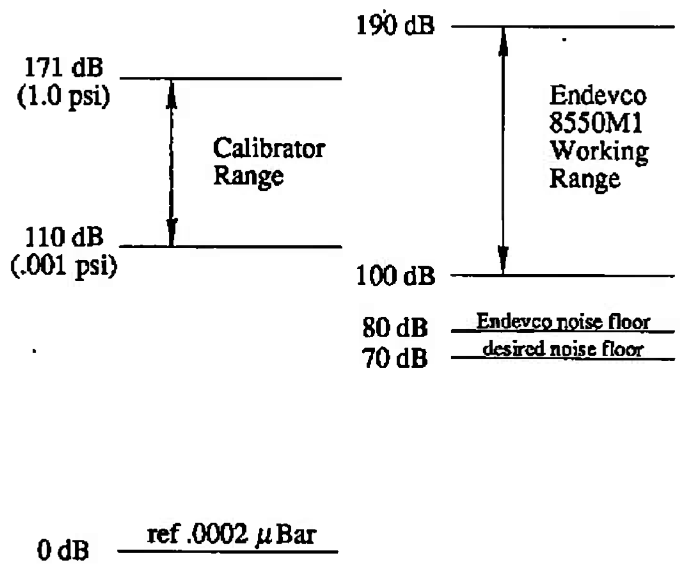

Figure 6. Calibration dynamic range.

A PCB model 103A02 microphone was obtained to fill in the lower pressure calibration range (approximately 110-130 dB). The device consists of an Invar diaphragm connected to a piezoceramic bender. The bender configuration produces extremely high sensitivity (2800 mV/psi) and the low-noise integrated amplifier produces a low-impedance output signal.

Unfortunately, when the microphone was installed into a calibration setup (a side-by-side comparison with the two Endevco sensors) the PCB did not perform to its calibration. A frequency response test showed no signs of distortion but the calibration factor was no where near the specification. A replacement was not available in time for the full-up system tests and so there are no results for the very lowest pressure range.

The strain gage signal conditioners used in this project (Validyne SG71) are not low-noise type (20 mV output at 10 VDC excitation). The highest quality low-noise strain gage amplifiers are good for approximately 2-μV RMS noise RTI in the frequency range of interest. Such an amplifier combined with careful shielding and grounding would produce much lower noise floors than what was measured in this project.

6.2 Servo-Controller

Two basic configurations of servo-controller were considered: 1) sweep generator with a dynamic compressor, and 2) a digital system. The sweep/compressor configuration has many attractive features

- very high dynamic range (>140 dB is possible)
- very high sensitivity
- relatively low cost

The drawbacks are considerable

- compressors require *long* warmup times before they are stable (3 hours is usually a minimum)
- require frequent recalibration
- very limited test reporting capabilities

Digital servo-controllers have average dynamic range (>60 dB), good sensitivity, are very stable, and very flexible. They have the advantages of computer based instrumentation, such as, flexible data communications and data output, easily stored and restored test configurations, and infrequent required calibrations. Their main drawback is their expense.

The digital controller was used in this project for several reasons

- They possessed adequate dynamic range and sensitivity.
- Rapid test setup and test modification (very handy in prototype and development work).
- Good archiving and documentation of test results.

This project required both swept-sine and random control of the input. There are very few single controllers that combine both features. The GenRad 2511 and 2514 controllers are the best known controllers that meet these requirements.

These controllers are very expensive and not frequently carried by instrument rental companies. CSA was able to rent a GenRad 2514 from a local environmental testing company that we occasionally work with. The expense of the controller limited us to a few days of testing in our lab.

6.3 Calibrator Hardware

Figure 10 shows the setup of the shaker–piston driver. The Airpot is intended for use as an actuator, i.e., air pressure enters through a hose barb at the top thus generating a force in the pushrod. A machinist shaved (very carefully since the cylinder *is* glass) the hose barb and a portion of the bonded phenolic top. The existing air hole in the phenolic was then bored out to the proper diameter (1/2"). Several sets of different "legs" were made the stand the Airpot off from the lower support plate. The lower triangular standoffs pick up some modified threads of pre-existing bolts used to hold the shaker together. The pushrod has a double-ended ball joint. This turned out to be important because the top plate imbedded in the phenolic does not register perpendicularly to the cylinder. Thus the pushrod is slightly slanted with respect to the shaker centerline. The reference sensor is inserted into a tapped acrylic block. This block, along with the remaining phenolic was sized to make up the joining cavity between the piston bore and the tube. The tube is extruded 1/4"-ID acrylic. The tube is set into an acrylic block which mates with the reference sensor block. The upper plate and long screws serve as a clamp to seal the two blocks. The upper end of the tube two acrylic blocks serve to hold the sensor-under-test and as the terminating volume (0.1 in^3). Figure 7 shows essentially the same configuration in a schematic cross-section.

Figure 5 shows the compression driver setup. It is mechanically very simple. The suspension, diaphragm, and cavity were already complete. All that was required was to cap the cavity with a plate containing a centered mating hole. The entire reference sensor block and tube assembly was simply transferred over.

A 10-in^3 volume (shown in Figure 13) was made in essentially the same way as the *typical* tube. This tube had a 1" inside diameter and was 12-3/4" long.

7. PREDICTIONS

With the results of the component tests in hand we can construct a prediction model. This model will give predictions of actual system performance to the degree that the actual system has been modeled. We have neglected

- flexibility of the tube walls.
- leakage in the driver (if any) and friction in the driver.
- dispersion sources in the fluid, i.e, there is no compressible flow. This clearly is untrue at acoustic pressures of several psi.

Using Eq. 9 we can compute the performance of candidate systems very quickly. Figure 8 shows a predicted response for the *typical* configuration with (24"-long, 1/4"-diameter tube, 1.25"×2.0" cavity. The piston is the Airpot S325P. The piston and pushrod weigh 30 grams.

The dotted curve shows that this configuration can meet the highest pressure requirement over a significant part of the frequency range.

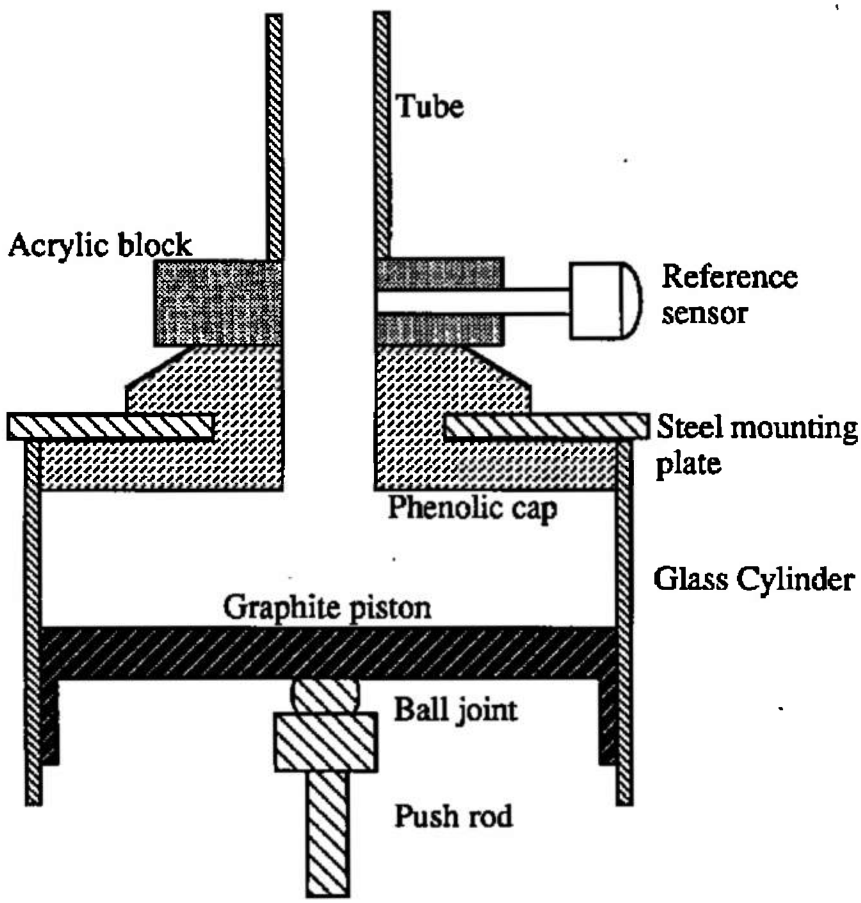

Figure 7. Cavity schematic

7.1 Effects of Cavity Configuration

The combined method also allows us to predict pressure response at any particular point in the cavity or tube. Equation 9 is modified to include cross admittances between the piston and the point of interest. With this capability we are able to "probe" around a candidate cavity design and find the best spot to place the reference sensor. It also allows us to determine if the reference sensor placement we have chosen permits accurate measurement of the actual input to the tube.

Using the configuration above, FRF's were computed between (see Figure 9) the piston (point A) and reference sensor location (B) and between the piston and the first grid point of the tube (C). The spacing between the grid points was 70% of the diameter of the reference sensor's outer case. (This is typical of the actual spacing.) The RMS of the difference of the magnitudes of each FRF ($|H_{AB}| - |H_{AC}|$) was computed and compared to the RMS of the magnitude of $|H_{AC}|$. The ratio was

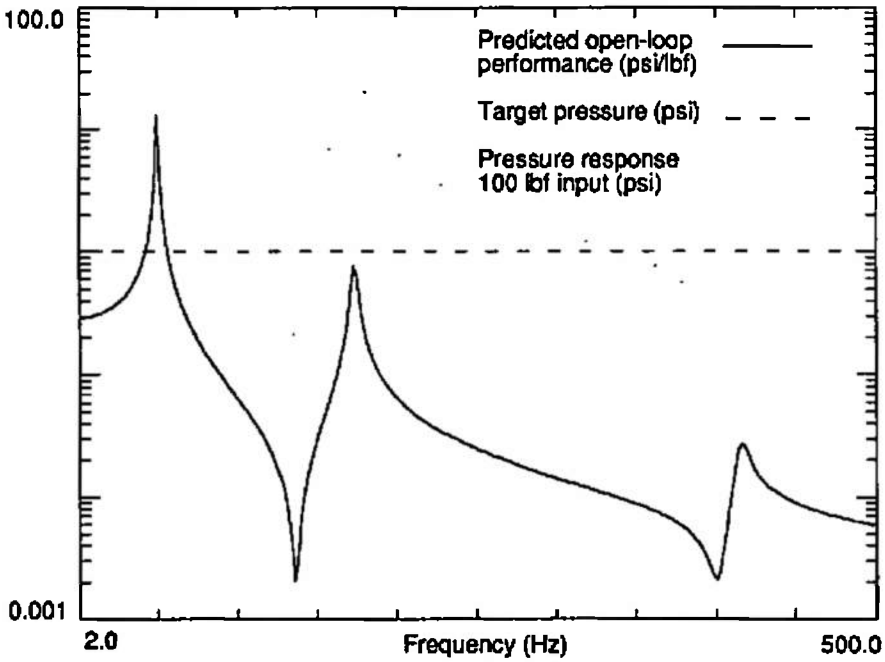

Figure 8. Open-loop performance for the *typical* configuration.

2.1×10^{-6}. This difference is considered to be so small as to be negligible. The worst phase error over this range (occurring at the second resonance) was 0.012°.

If the tube length were considerably shorter this error would rise. Obviously, if the sensor size and/or the spacing between the sensor and the start of the tube is large compared to tube length, the error introduced might be appreciable.

Some studies indicated that the larger the cavity (to some degree) the smaller the error computed above.

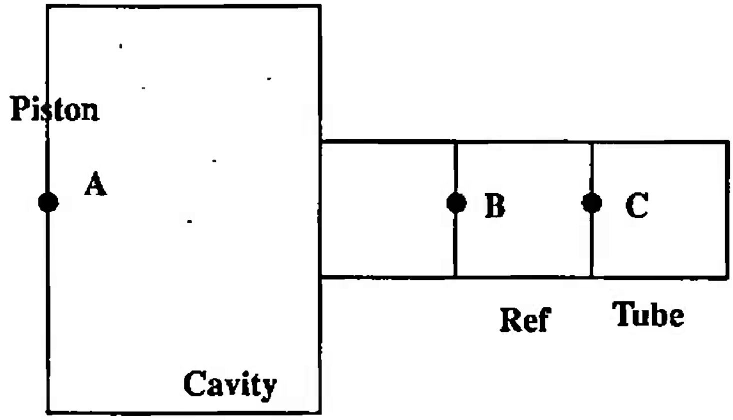

Figure 9. Model for computing reference sensor placement error.

8. RESULTS

Figure 10 shows the setup of the shaker-piston driver and the *typical* configuration. The reference sensor is the Endevco 8510B (just above the piston) and the sensor-under-test is another Endevco 8510B. A Kistler 5-lb load cell is placed between the shaker head and the pushrod (see Figure 11). The signal was monitored and recorded during testing.[5]

Figure 12 shows the *typical* configuration set up on the compression driver.

Figure 13 shows a tube with a 10-in^3 volume.

Figure 14 shows the reference sensor mounted in its block with the large tube mounted on top.

A large number of individual tests were required to determine performance of the system for the various extremes in pressure, frequency, and volume. Table 2 lists configuration and results for the most important tests.

In short, using the shaker–Airpot combination achieved the swept-sine high pressure, largest volume specification over most of the frequency range within the specified maximum pressure variance. The compression driver was able to meet the pressure specification into the *typical* configuration over most of the frequency range. By using both drivers the entire frequency range may be covered for the largest volume tube. By using two reference sensors (the strain gage type and a microphone)

[5]To verify calibrations of the sensors two sensors were placed side-by-side in a closed cavity atop the piston. The force into the piston was measured for a sinusoidal input at a given frequency. This system was modeled using the admittance technique and compared with the test. Using the measured force as the input it was verified that the Endevco sensors were in spec but the PCB microphone was not.

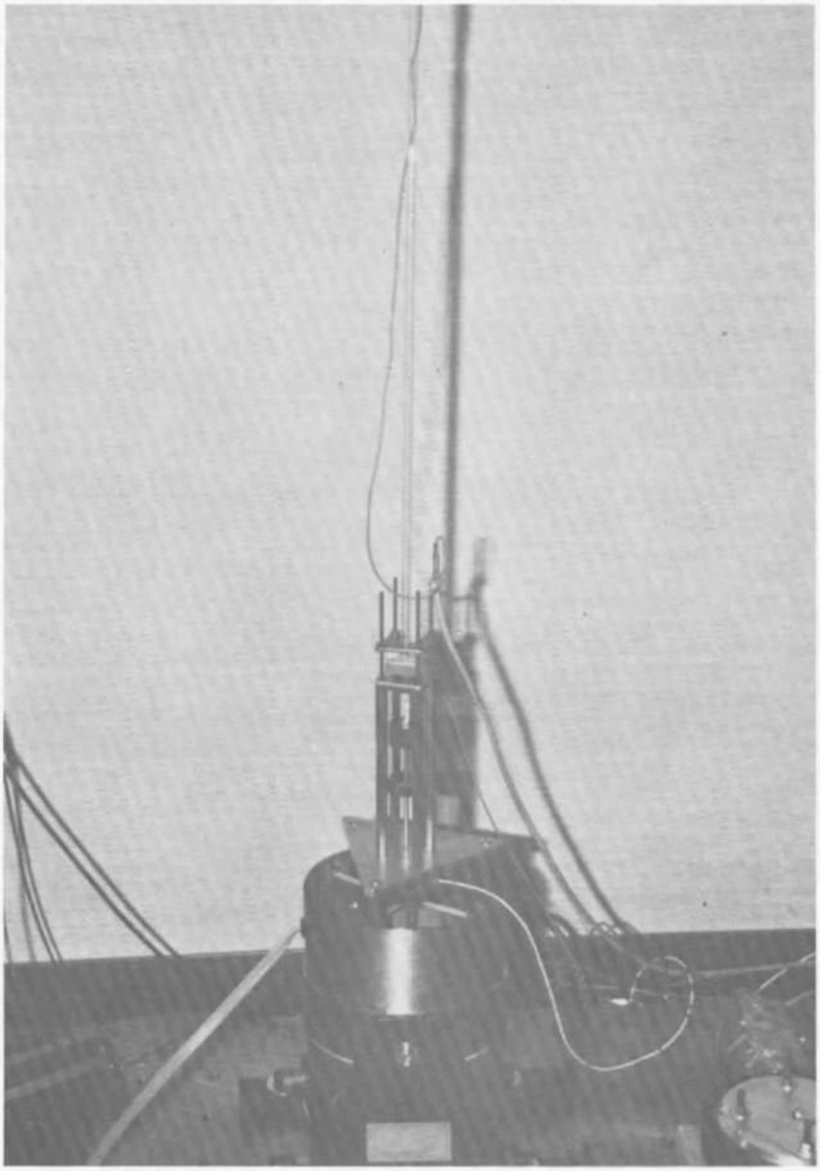

Figure 10. *Typical* configuration test setup with shaker and piston.

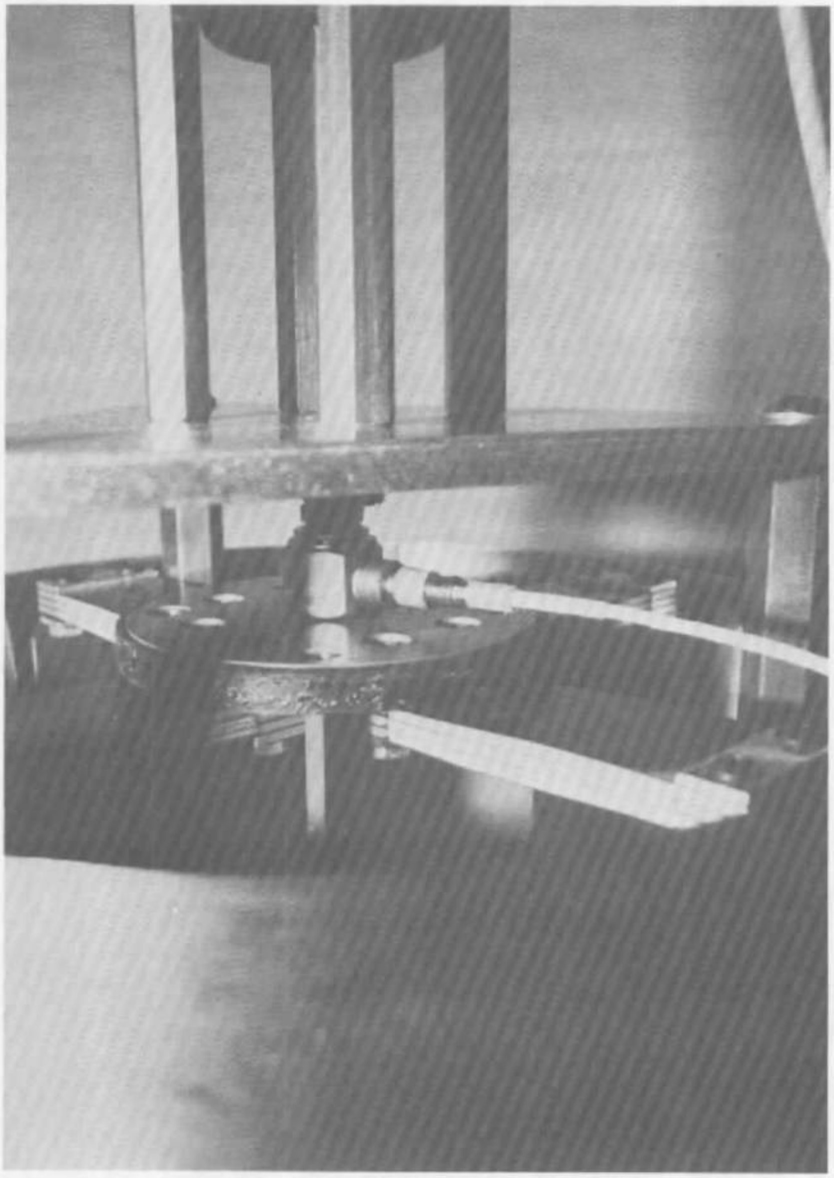

Figure 11. Closeup: loadcell between shaker head and pushrod.

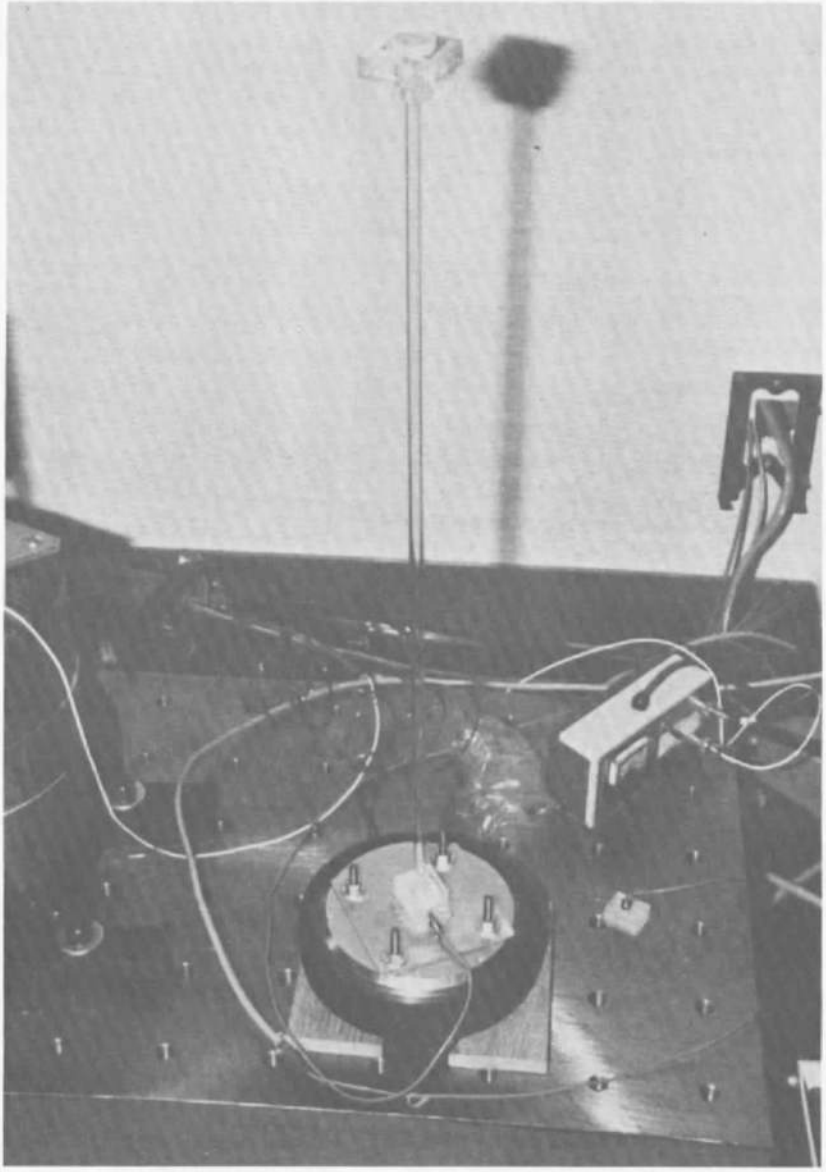

Figure 12. *Typical* configuration test setup with compression driver.

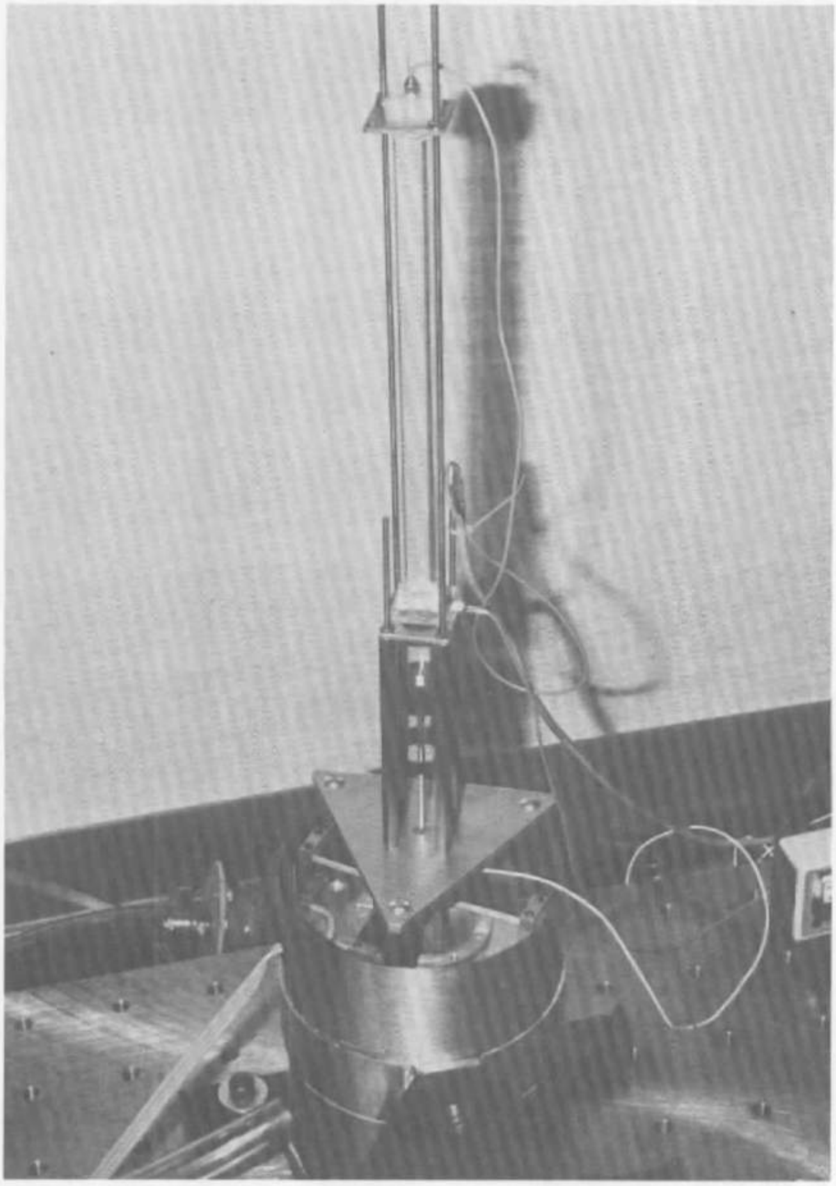

Figure 13. Large volume (10 in^3) with shaker and piston.

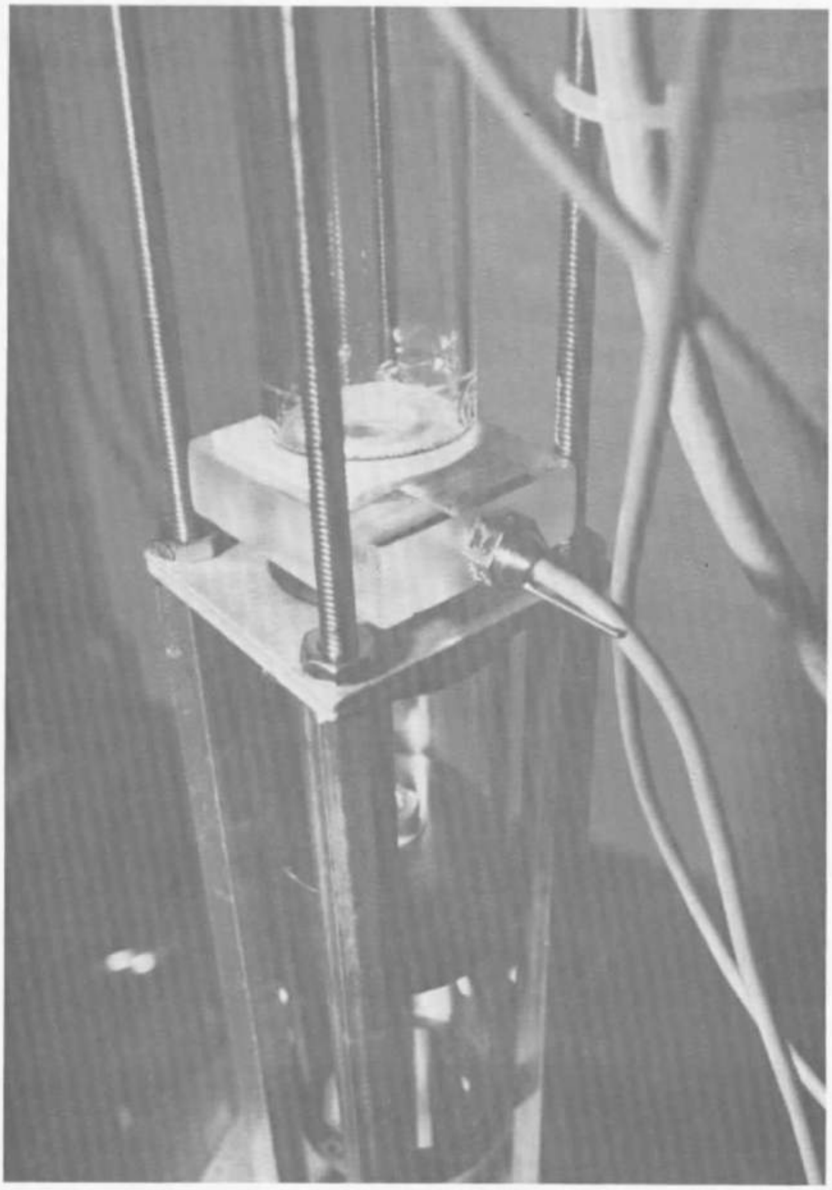

Figure 14. Reference sensor setup.

Swept-sine / Shaker

Pressure (psi)	Configuration	Frequency range (Hz)	Average error	Maximum error
1.	*typical*	0.5–420 *	< .5 dB	.7 dB
1.	10 in^3	2–400	< .2 dB	.4 dB
0.005	*typical*	2–500	< .2 dB	.35 dB

*required 3 separate sweeps to cover full frequency range.

Random / Shaker

Pressure RMS (psi)	Configuration	Frequency range (Hz)	Average error	Maximum error
1.	*typical*	2–500	failed to EQ	6.2 dB
0.2	*typical*	2–500	1.7 dB	3.5 dB

Swept-sine / Compression Driver

Pressure (psi)	Configuration	Frequency range (Hz)	Average error	Maximum error
1.	*typical*	5–1000 **	< .2 dB	.4 dB
0.005	*typical*	2–1000	< .2 dB	.32 dB

** 5 Hz limit was difficult to achieve, 1000 Hz limit was easily achieved.

Random / Compression Driver

Pressure RMS (psi)	Configuration	Frequency range (Hz)	Average error	Maximum error
0.2	*typical*	10–500	failed to EQ	2.5 dB

Table 2. Summary of major full system test results. Average error is calculated by averaging the rectified error. That is, at each sample point (512 equally spaced points in log time) the absolute value of the difference between the actual reference pressure and the target pressure is used. The maximum error is the largest single point error that occurred over the control bandwidth.

the entire pressure range may be covered. (This is somewhat speculative since the lowest pressure range was never actually achieved.)

The random input case was never successfully run at the highest pressure (for any configuration).

8.1 Typical Calibration Curves

A series of representative calibration curves are presented in this section. Some of the curves are best case, some are simply typical.

Figure 15 is a best case result using the compression driver into the *typical* configuration. Only one gain setting was required for the sweep (shown in the bottom). The reference sensor pressure was remarkably flat with the biggest variations occurring at acoustic resonances. The average error in this sweep was 0.1 dB and the largest error was 0.7 dB (occurring at the lowest resonance). The two horizontal lines (called guards) are set at ± 1 dB of the target pressure. Figure 16 is a plot of phase between the reference sensor and sensor-under-test (note the slightly different frequency scale).

Figure 17 shows a typical result from the random input tests. This is a high pressure run (0.002 psi^2/Hz, 1 psi RMS). The reference pressure curve (bottom) shows the common features of these tests. The lower frequency gains are too high and the high frequency gains too low, with the middle decade being quite stable and with little variance.

8.2 Observations

Running the closed-loop tests using the shaker as the driver required that that the operator either

- monitor drive voltage into the power amplifier and carefully (and slowly) increase or decrease gain to keep input voltage in a usable range, or
- carefully prefix the amplifier gain for a given frequency range and use several sweeps to cover the entire frequency range.

The first choice is nerve wracking and clearly undesirable. The second choice is simply slow. Ideally, a compression-expansion circuit would be inserted between the servo-controller and the power amplifier to modify overall gain to keep signal-to-noise figures optimal.

Figure 18 shows the amplifier gain settings used for various input pressure calibration sweeps for the shaker configuration. Another way to interpret this graph is that it shows optimal gain setting for a single sweep as response pressure changes.

Clearly, the dynamic range of input voltage and gain product is too high for a single setting. (Most sweeps used the full dynamic range of the servo-controller drive signal at a fixed amplifier setting.)

The driver diaphragm weight is so low that the driver looks like a pure stiffness in the frequency range of interest. Because of this only one gain setting on the amplifier was required for the compression driver sweeps.

In random mode the servo-controller gradually brings drive level up to the specified level. The controller attempts to stabilize and equalize before going to full drive level. At the very high level of 1 psi RMS (0.0002 psi^2/Hz) the controller was not able to equalize. Even long equalization times below full drive level were not sufficient. Figure 19 shows a spectrum of drive voltage for the 2–500 Hz range. The spectrum shows that a very large dynamic range in drive voltage (control signal) is required. In fact the low frequency region is at the noise floor level of the controller output amplifier. This indicates that the controller is essentially incapable of controlling the driver at low frequencies (all of its authority is being used at higher frequencies).

At the 0.005–psi pressure level the piston displacement needed to generate this pressure was extremely small. This meant that the amplifier gain was extremely small. Therefore, cavity volume was doubled from the volume used in the high-pressure tests. This was done by inserting longer "legs" between the triangular support plate and the cylinder top. (The S325P Airpot cylinder was longer than necessary in anticipation of this situation.) Doubling the cavity volume improved the overall system stability considerably. (Having a control with excessively high authority often leads to overshoot or even runaway.)

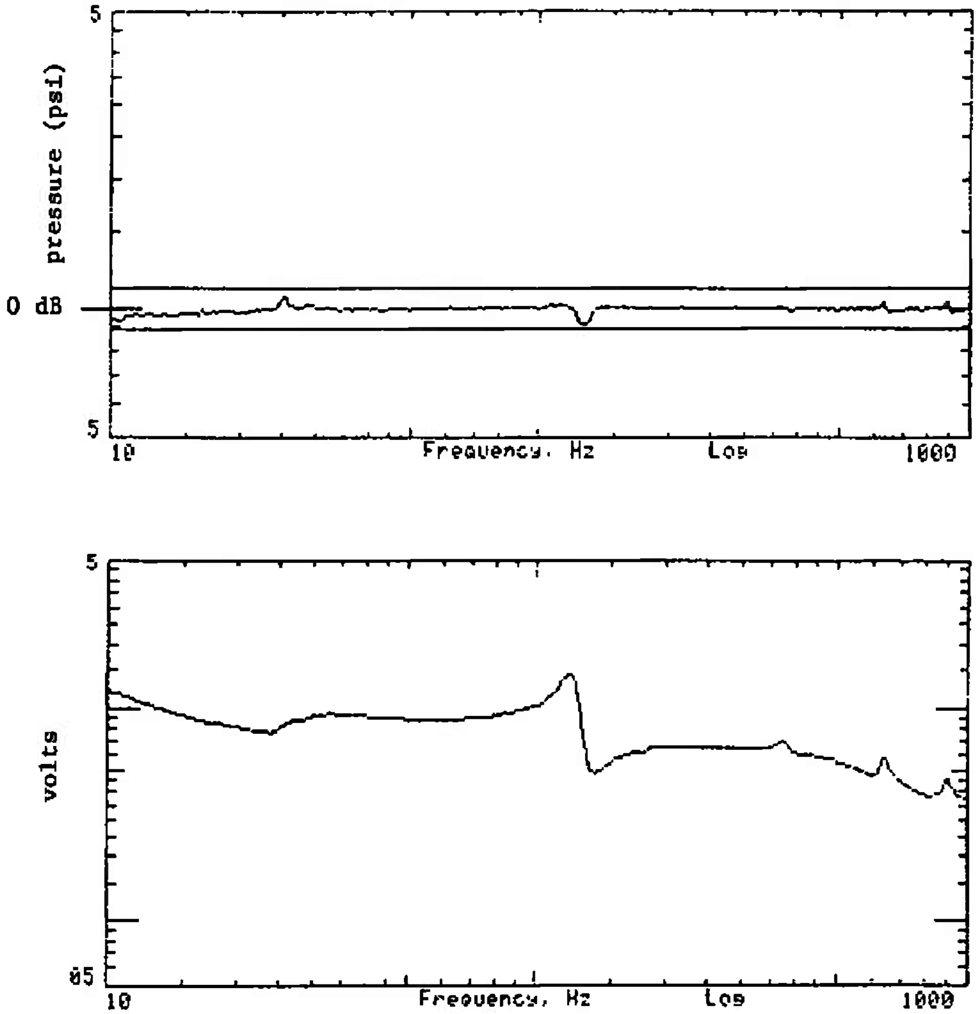

Figure 15. Sine sweep 10–1000 Hz at 1psi peak, *typical* configuration, compression driver. Top: reference pressure (psi). Bottom: drive voltage (volts).

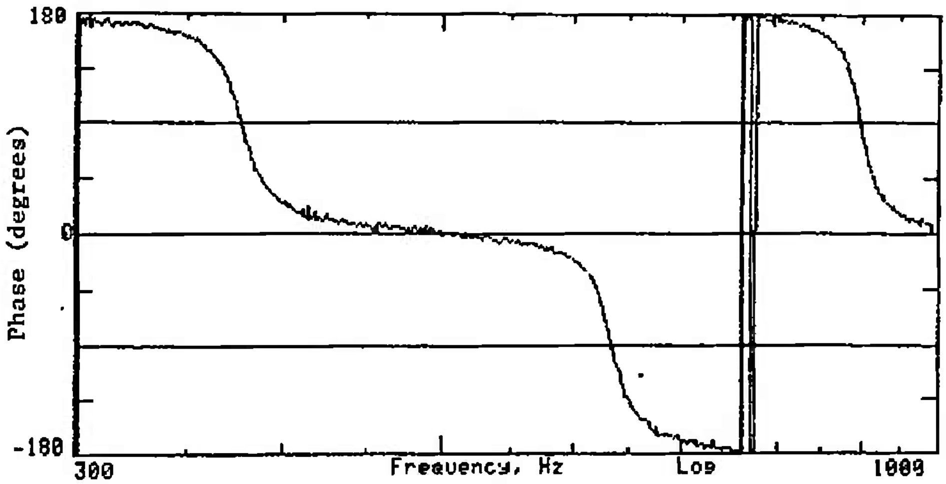

Figure 16. Phase: Reference to Sensor—Sine sweep 300–1000 Hz at 1psi peak, *typical* configuration, compression driver.

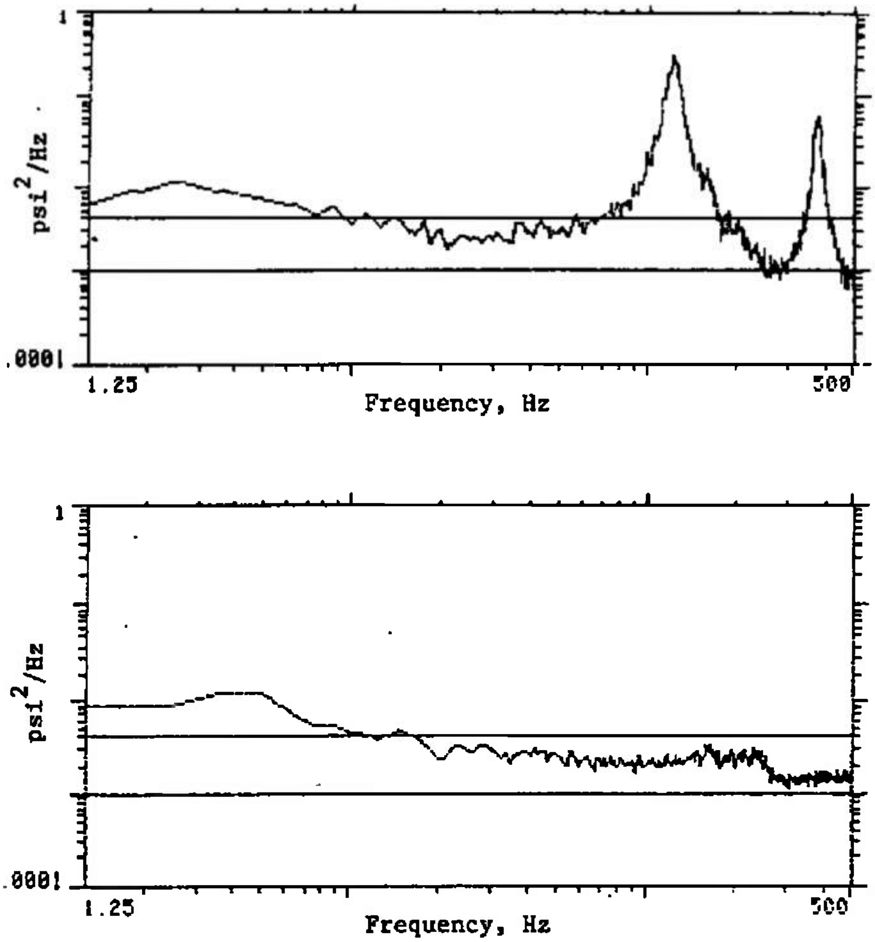

Figure 17. Random input, 2–500 Hz at 1psi RMS, *typical* configuration, shaker driver. Top: pressure sensor-under-test (psi^2/Hz). Bottom: reference pressure.

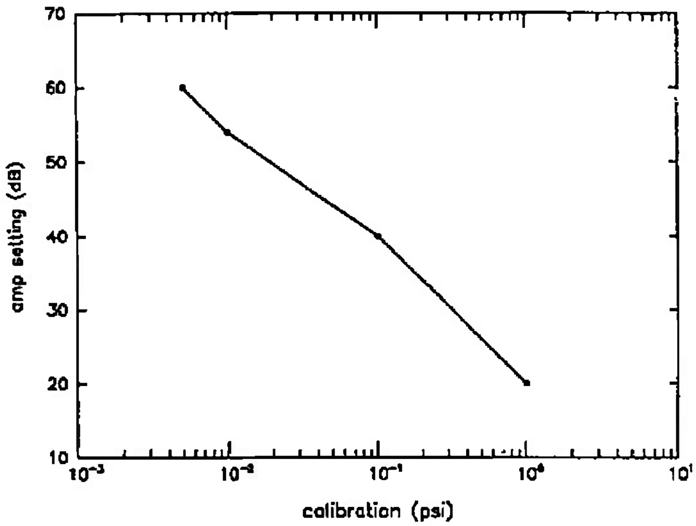

Figure 18. Optimum amplifier setting for a given calibration pressure.

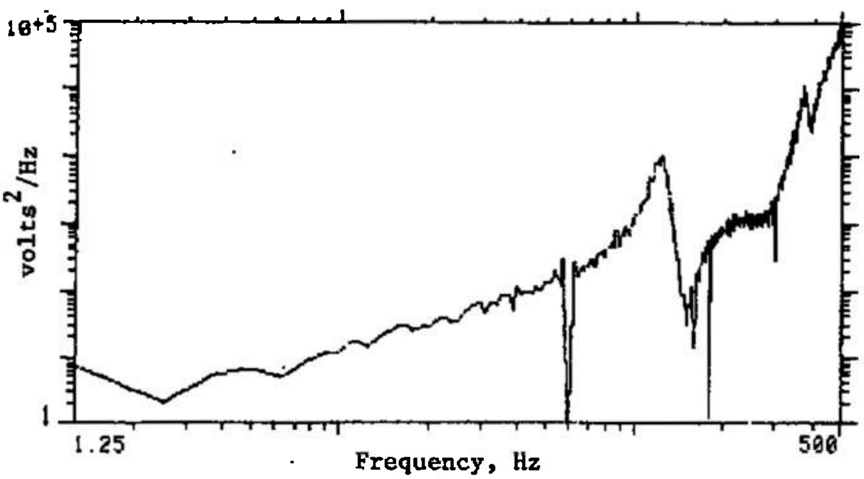

Figure 19. Random excitation drive voltage spectrum.

9. CONCLUSIONS & SUMMARY

A hybrid technique which allows modeling of combined acoustic-mechanical systems has been developed and used to size the calibrator. Using measured component properties this technique accurately predicted the total system performance and enabled evaluation of the effects of reference sensor placement.

The calibrator is capable of achieving all of the swept-sine specifications by using two driver and two reference sensor combinations.[6] The calibrator was actually capable of greatly exceeding the requirements for certain configurations. The random input specifications were never met for the entire frequency range at high pressure. The servo-controller could not achieve equalization; this stems from inadequate dynamic range of the control.

Two different driver configurations were investigated. The shaker driver works best in the low-frequency, high-volume region. The compression driver has excellent high-frequency characteristics.

The approach has proven capable of providing an accurate and controlled pressure over a wide range of configurations and should be capable of being extended to an even wider range of applications.

[6]The only caveat being that the lowest pressure range was never actually tested.