This is to certify that the

thesis entitled

THE DEVELOPMENT OF AN ACOUSTIC ENERGY
ABSORBER FOR NOISE CONTROL APPLICATIONS

presented by

GREGORY DAVID HALL

has been accepted towards fulfillment
of the requirements for

MASTERS degree in MECHANICAL ENGINEERING

Clark Radcliffe
Major professor

Date 6/25/93

THE DEVELOPMENT OF AN ACOUSTIC ENERGY ABSORBER FOR NOISE CONTROL APPLICATIONS

By

Gregory David Hall

A THESIS

Submitted to
Michigan State University
in partial fulfillment of the requirements
for the degree of

MASTER OF SCIENCE

Department of Mechanical Engineering

1993

ABSTRACT

THE DEVELOPMENT OF AN ACOUSTIC ENERGY ABSORBER FOR NOISE CONTROL APPLICATIONS

By

Gregory David Hall

Consumer perception of noise plays in important role in the automotive industry. High levels of interior noise have a tiring effect on the driver and are perceived to be an indicator of inferior quality. An Active Acoustic Sink (AAS) serves as a sink for acoustic energy to reduce the level of noise at all points in an enclosed space by absorbing the acoustic energy of incident sound waves. The first part of the research effort was to compensate an audio speaker, using speaker velocity feedback, to improve its bandwidth and reduce its phase shift. The compensated speaker was then used as the control actuator of the AAS to dissipate acoustic energy. Experimental results indicated that the AAS had increased acoustic energy absorption up to 182% over the absorption of an uncontrolled speaker from 65-120 Hz.

ACKNOWLEDGMENTS

I would like to thank Professor Clark Radcliffe of the Department of Mechanical Engineering at Michigan State University for his support and guidance as well as the idea for the acoustic energy absorber. I would also like to thank my wife, Tonya, and my parents for their moral and financial support during my graduate studies.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

NOMENCLATURE

bl	electromagnetic coupling factor
c	speed of sound
$e(t)$	voltage
e_b	back emf
G_{spkr}	transfer function of the speaker
$\mathbf{I}$	acoustic intensity vector
I	acoustic intensity
K_a	absorber gain
K_p	proportional gain
L_{coil}	speaker coil inductance
P_d	desired pressure
P_{ref}	reference pressure
r	input
$\mathbf{v}$	velocity vector
v_d	desired velocity
v_{spkr}	speaker velocity
Z	speaker impedance
ρ_0	density of air
$(\rho_0 c)$	characteristic impedance

INTRODUCTION

Consumer perception of noise has become increasingly important in the automotive industry. A high level of vehicle noise is undesirable due to its tiring effect on the driver and passengers. A high level of interior noise is also perceived to be an indicator of an inferior product.

Noise treatment is obtained by either passive or active solutions. Passive solutions such as sound absorbent materials are ineffective at low frequencies (< 200 Hz) due to the large thickness of material required to absorb the sound waves (Everest, 1989). Active noise cancellation has been applied commercially, but it is limited in that it can only eliminate the offending noise at measurement microphone locations. An Active Acoustic Sink (AAS) serves as a sink for acoustic energy in order to reduce the level of noise at all points in an enclosed space by absorbing the acoustic energy of incident sound waves.

The research problem is to determine whether an electronics driven speaker can be designed to absorb acoustic energy out of an acoustic volume. Acoustic intensity is the quantity used for both the basis of the design and for the subsequent evaluation of the design. Acoustic intensity is easily measured using commercial instrumentation such as the Brüel & Kjær Sound Intensity Probe (Type 3519). The first part of the research effort produced an audio speaker compensated to provide accurate tracking of velocity signals generated by a controller. The second research effort produced a controller using the compensated speaker to dissipate sound.

AN ACOUSTIC ENERGY ABSORBER

The two approaches to reducing the sound pressure level at a point are cancellation and acoustic energy dissipation. Noise cancellation involves measuring an input pressure and then creating an equivalent pressure in the opposing direction. The addition of out of phase "anti-noise" to the disturbance results in a zero response at the disturbance point. An alternate approach to decreasing sound pressure level involves reducing acoustic energy using active control. Techniques using acoustic energy dissipation cannot achieve a zero noise level response but reduce the sound pressure level at all points in an acoustic volume.

The history of acoustic energy absorption dates back to the early 1950's when Olson and May built an "Electronic Sound Absorber" (Olson and May, 1953). In contrast to its name, this device did not actually absorb acoustic energy. It used electronics to maintain a zero pressure level at a microphone measurement location and hence was the first sound cancellation system. Today, "anti-noise" sound cancellation remains the dominant approach to active noise control.

There are two problems with noise cancellation that have stimulated interest in alternate methods of noise control. First, cancellation is limited to applications in which the control actuator is co-located with the disturbance. Second, phase distortion of the anti-phase signal could result in the noise being increased rather than eliminated. Techniques using acoustic energy dissipation are not restricted by the need to know the spatial location of the disturbance.

Acoustic intensity is the quantity used for both the basis of the design and for the subsequent evaluation of the design. It is defined as the average rate at which sound energy is transmitted through a unit area (Beranek, 1986).

$$\mathbf{I} = P \cdot \mathbf{v} \tag{1}$$

The objective of the active acoustic sink is to maximize the component of acoustic intensity vector that is directed into the speaker. An uncontrolled speaker will have some acoustic intensity directed into the speaker because of its mechanical dissipation. A controlled speaker will be designed to have a greater magnitude of acoustic intensity directed into the speaker (Figure 1).

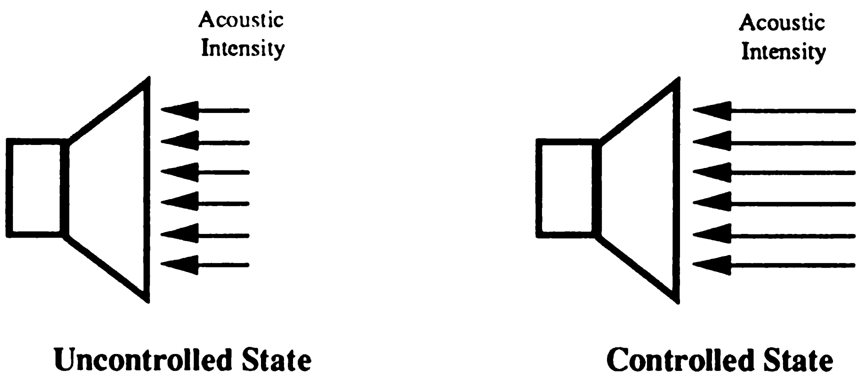

Figure 1. Effect of Controller on Acoustic Intensity

Speaker velocity must be accurately tracked to guarantee acoustic energy flow into the speaker. Pressure cannot be controlled because it is a response, but it can be easily measured using a microphone. The measured pressure can be multiplied by an absorber gain, K_a, to obtain the velocity, which may be controlled using a prototype AAS (Figure 2).

$$v = \mathrm{v}(P) = -K_a P \tag{2}$$

Acoustic intensity (1) may be expressed as the product of the negative of the absorber gain, K_a, and the pressure, P, squared.

$$I = -K_a P^2 \tag{3}$$

If the speaker velocity can accurately track the measured pressure (2), the resulting acoustic intensity (3) will always have a negative sign and thus be directed into the speaker.

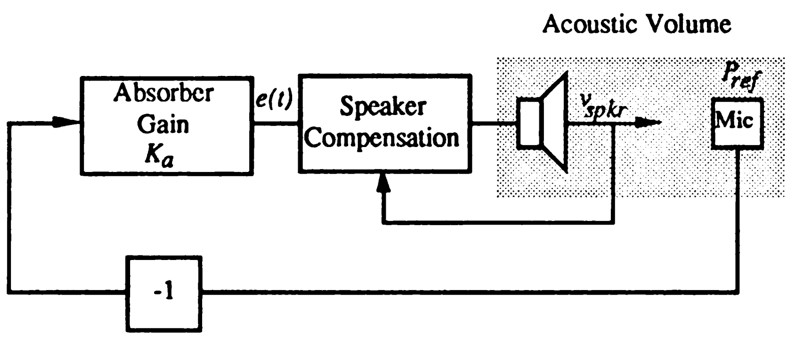

Figure 2. Block Diagram of the Active Acoustic Sink

SPEAKER COMPENSATION

Speaker compensation is needed to allow the use of a limited bandwidth audio speaker as a control actuator for low frequency noise control (Radcliffe and Gogate, 1992). The typical audio speaker is designed to provide a flat pressure frequency response (Figure 3), however, the pressure response is proportional to speaker cone acceleration which provides a non-constant velocity frequency response (Figure 3). The use of an audio speaker in control applications requires a flat velocity frequency response with phase as close to zero as possible. The zero phase condition is what ensures that the sign of the acoustic intensity is negative and thus acoustic energy is directed into the speaker

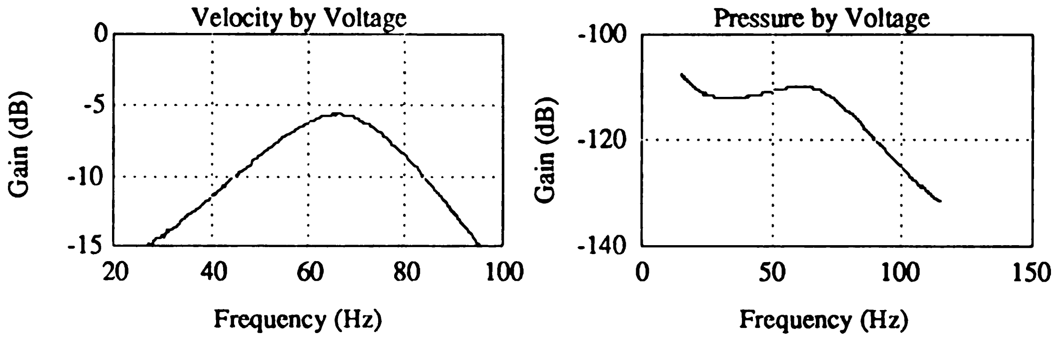

Figure 3. Typical Speaker Velocity/Voltage and Pressure/Voltage Frequency Responses

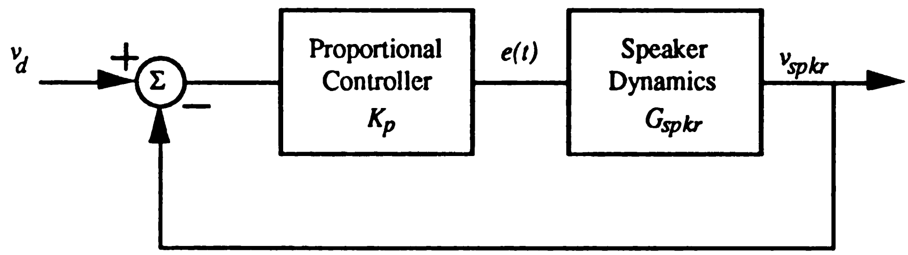

Figure 4. Speaker Velocity Feedback

Velocity feedback compensation (Figure 4) increases the speaker bandwidth and reduces the phase shift. Speaker velocity may be obtained with a dual-coil speaker in which the second set of speaker leads is used to acquire the speaker back emf, e_b, which is proportional to the speaker velocity, v_{spkr}, at low frequencies.

$$e_b = (bl)v_{spkr} \tag{4}$$

The constant of proportionality, *bl*, is the speaker electromagnetic coupling factor.

The closed loop transfer function of the speaker velocity feedback loop is obtained from the speaker velocity feedback block diagram (Figure 4) where G_{spkr} represents the transfer function relating the speaker velocity to the drive voltage, *e(t)*.

$$\frac{v_{spkr}}{v_d} = \frac{K_p G_{spkr}}{1 + K_p G_{spkr}} \tag{5}$$

The speaker velocity transfer function (5) indicates that we would optimally like to drive the proportional gain, K_p, as high as possible to achieve unity. A unity gain would represent a perfect tracking of the desired velocity, v_d, by the speaker. The gain and phase margins of the open loop response must be determined to find the maximum value of the proportional gain, K_p, that yields a stable response.

The open loop frequency response of the speaker velocity feedback system (Figure 5) exhibits magnitude varying over 20 dB and a phase shift varying almost 180 degrees. Closed loop speaker velocity feedback response (Figure 6) with a proportional gain, K_p, equal to 10.0 provides a magnitude response that varies less than 4 dB and phase distortion less than 30 degrees for a bandwidth of 10-810 Hz. This proportional gain setting was close to the threshold of instability and was chosen to be the largest possible stable gain.

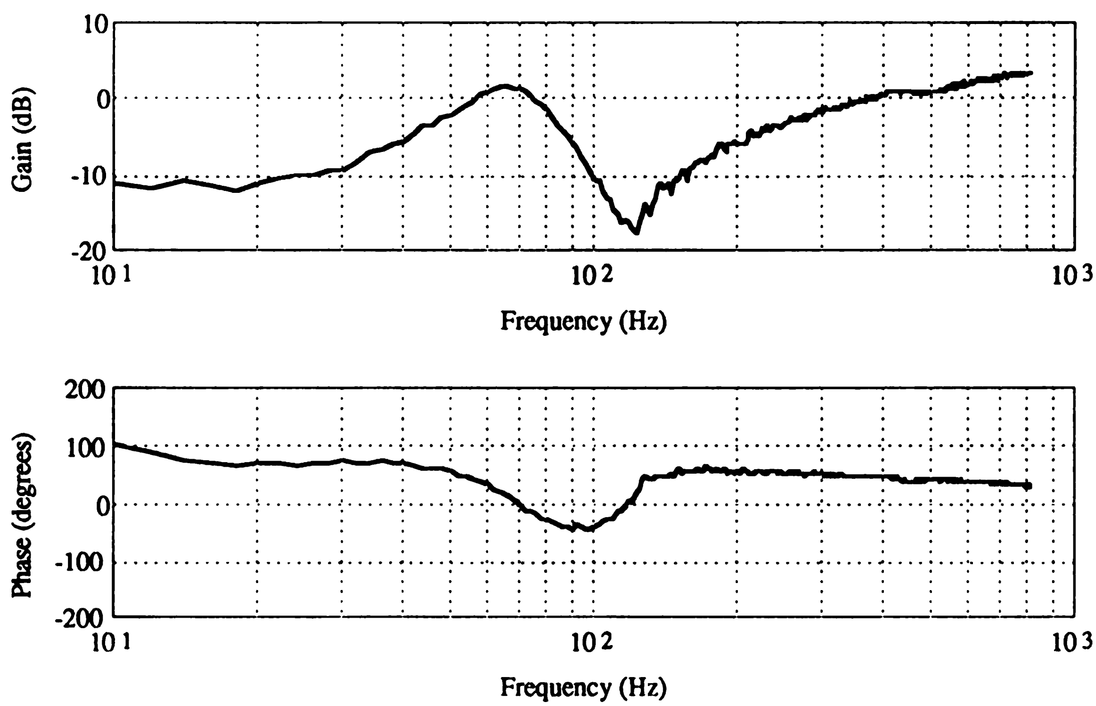

Figure 5. Open Loop Frequency Response of Speaker Velocity Feedback System

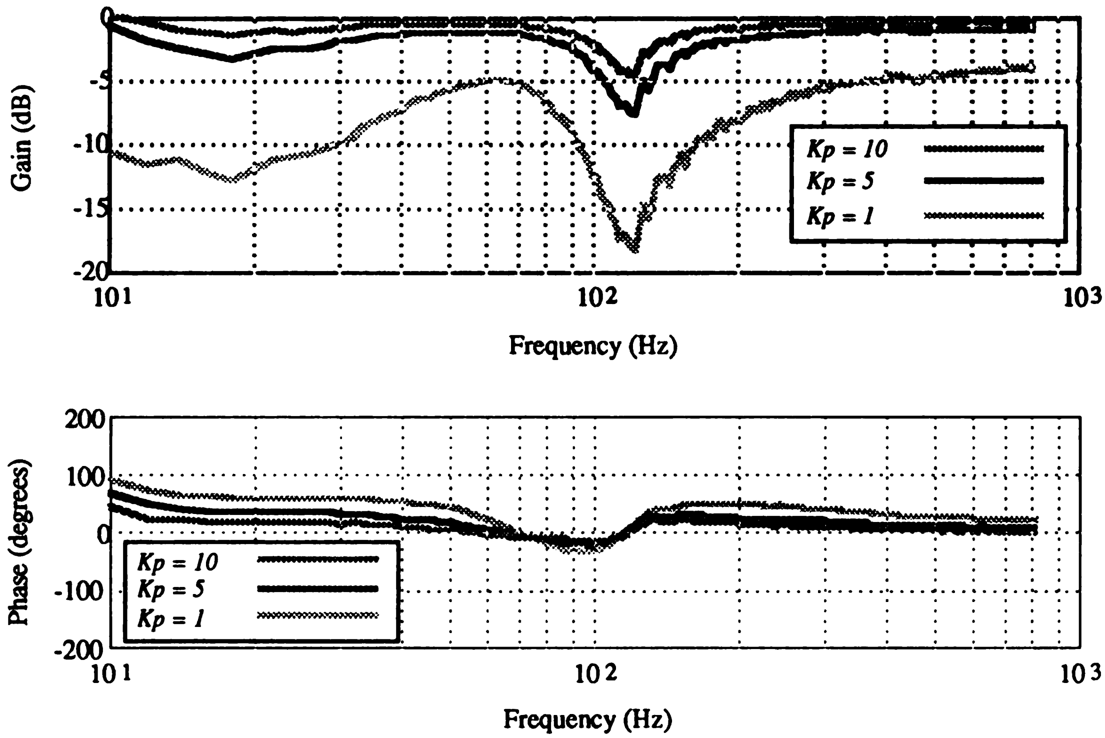

Figure 6. Closed Loop Frequency Response of Speaker Velocity Feedback System

Table 1. Results of Speaker Compensation (20-100 Hz)

Compensation	Gain Variation (dB)	Phase Variation (degrees)
Uncompensated Speaker	10	95
Compensated Kp=1	7	90
Compensated Kp=5	3	60
Compensated Kp=10	1	30

ACTIVE ACOUSTIC ABSORBER IMPLEMENTATION

The active acoustic sink drives speaker surface velocity, v_{spkr}, so that it tracks measured pressure, P_{ref}, resulting in local acoustic intensity always being directed into the speaker. This process would appear to be open loop, however, the acoustics of the space provide a transfer function between speaker velocity and local measured pressure. A typical feedback control design using an open loop Bode diagram was performed because the system could be represented as closed loop.

The open loop transfer function was obtained by breaking the measured pressure, P_{ref}, feedback path and inserting the reference voltage, r (Figure 7). In this configuration, an absorber gain, K_a, of 2.0 yielded a gain margin of 7 dB with infinite phase margin (Figure 8). This was the maximum value of absorber gain, K_a, that met the usual minimum requirement of 6-8 dB gain margin (Ogata, 1986).

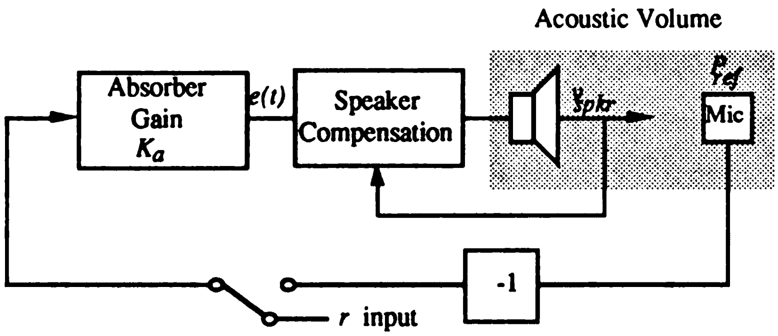

Figure 7. Open Loop System

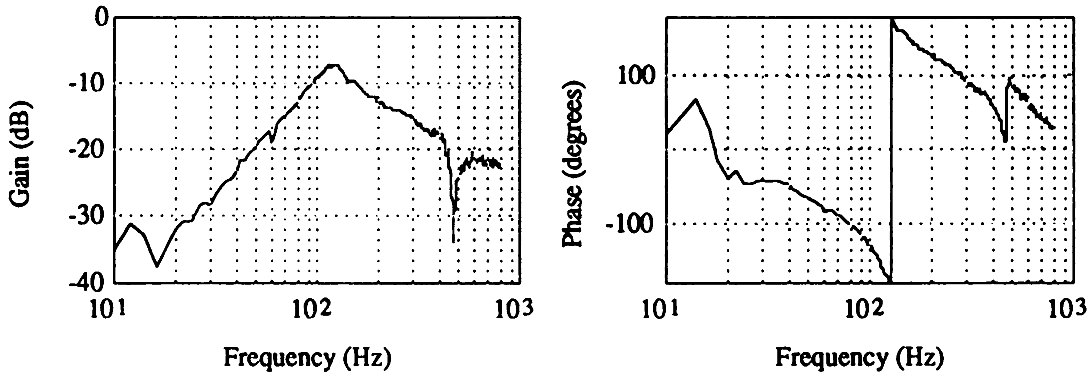

Figure 8. Open Loop Response (K_a=2.0)

Open loop frequency responses of pressure, P_{ref}, by speaker velocity, v_{spkr}, were generated for both the case where speaker velocity was measured from speaker back emf,

e_b, (Figure 9) and the case where the speaker velocity was measured using a Brüel & Kjær (Type 3544) Laser Velocity Transducer (Figure 10). Both responses provide similar responses up to approximately 120 Hz. At this frequency, the speaker back emf, e_b, measured from the speaker coil leads does not accurately represent the speaker cone velocity, v_{spkr}. This error is due to the assumption that the inductance of the speaker coil, L_{coil}, is negligible for low frequencies (Colloms, 1985). For low frequencies, the electric time constant determined by speaker inductance is much smaller than the mechanical time constant. This allows speaker inductance to be neglected and thus enables the speaker velocity to be obtained at low frequencies using the speaker coils (Radcliffe and Gogate, 1992). At higher frequencies, the inductance causes the speaker coils to be an inaccurate velocity sensor. At frequencies above 150 Hz, large fluctuations in the frequency response (Figure 10) indicate that the Laser Velocity Transducer may also be providing inaccurate velocity measurements.

A frequency response (Figure 11) representing the transfer function relating the velocities measured by the Laser Velocity Transducer and speaker back emf, e_b, clearly illustrates the influence of the inductance of the speaker coil, L_{coil}, on the velocity measurement. The open loop speaker response (Figure 5) indicates that there is a zero appearing in the open loop speaker transfer function at 120 Hz. This zero is attributed to the resonance due to the speaker coil inductance. The large phase change in the velocity measurement technique frequency response (Figure 11) at 120 Hz restricts controller operation to below that frequency. An improved velocity sensor would improve the tracking of the control signal velocity and thus improve performance.

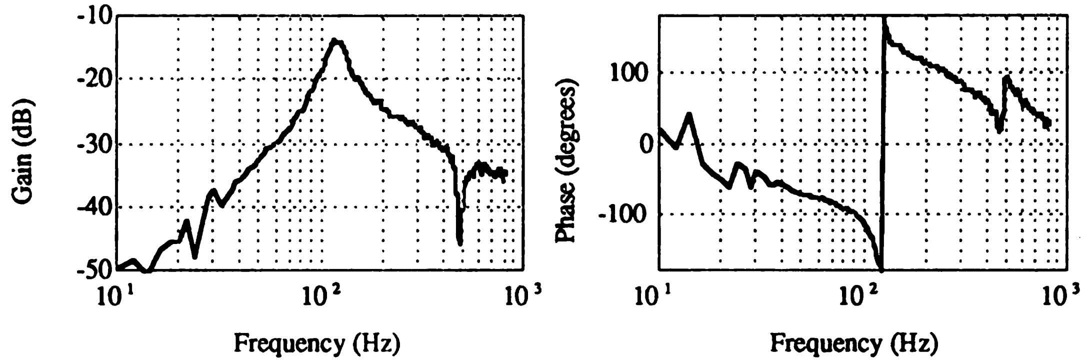

Figure 9. Frequency Response with Velocity Measured from Speaker Back EMF

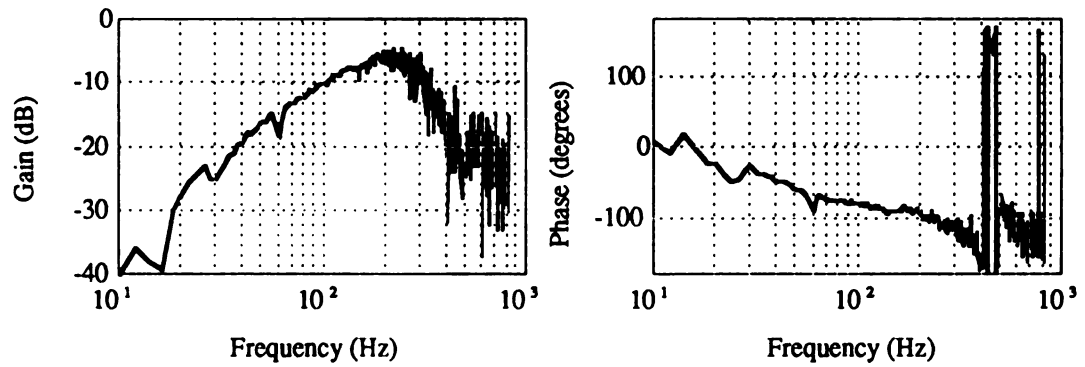

Figure 10. Frequency Response as Measured by Laser Velocity Transducer

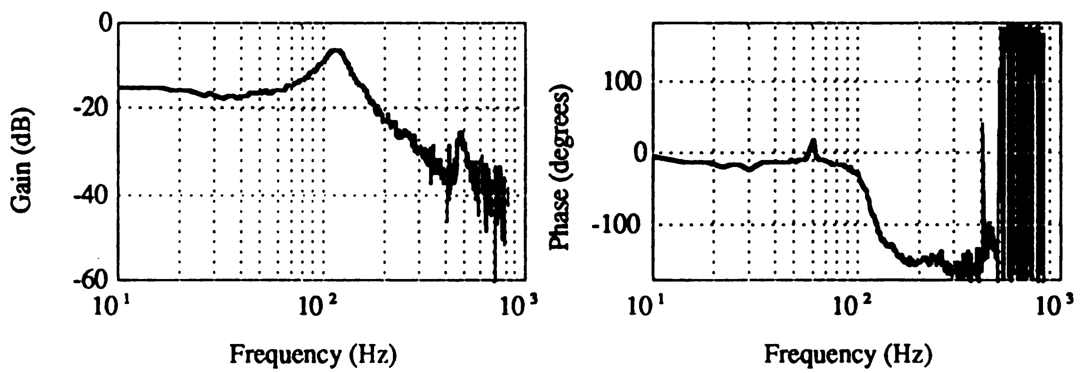

Figure 11. Frequency Response of Laser Velocity Transducer/Speaker Back EMF

EXPERIMENTAL RESULTS

Acoustic energy absorption was measured to determine the effectiveness of the active acoustic sink. A random noise signal was generated and amplified to a sound pressure level (SPL) of 120 dB. Acoustic intensity was measured at the front of the speaker for both the controlled and uncontrolled states (Figure 12).

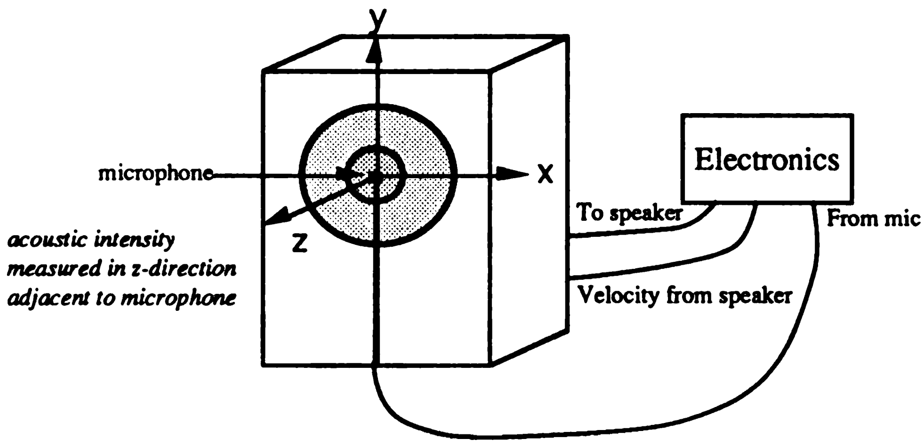

Figure 12. Experimental Set Up

Acoustic intensity measurements (Figure 13) taken with a Brüel & Kjær Sound Intensity Probe (Type 3519) clearly indicate that there is an increase in the absorption of acoustic energy over the range of measurement. The uncontrolled speaker had no connections applied, hence, there was no coil current and only mechanical dissipation. The increase in acoustic energy absorbed ranged from 9% over 70-80 Hz and 182% over 110-120 Hz (Table 2). The performance of the controller began to decrease above the bandwidth limitation (120 Hz) imposed by using speaker back emf as the speaker velocity sensor.

Acoustic intensity measurement problems were encountered below 65 Hz. The acoustic intensity measurements obtained below this frequency appeared to be unreliable. This was confirmed by measuring the acoustic intensity from a speaker emitting random noise below 65 Hz. The results showed only small amounts of acoustic intensity fluctuating in direction instead of large values of acoustic intensity directed out of the speaker. Subsequent measurements produced expected results at frequencies greater than 65 Hz. This behavior occurred both in the laboratory and when the tests were conducted outside the building in an unenclosed space. The measurement problem raises questions about the accuracy of the measurements made slightly above 65 Hz. If the acoustic intensity cannot be measured with any accuracy below 65 Hz, it is likely that the measurements over the 70-80 or 80-90 Hz bands may also contain some error.

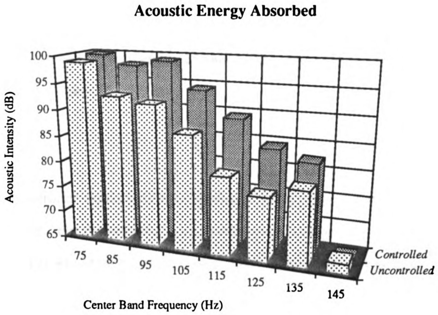

Figure 13. Acoustic Energy Absorbed By AAS

Table 2. Acoustic Energy Absorbed By AAS Compared To Uncontrolled State

Frequency Band	Change
70-80	+9%
80-90	+78%
90-100	+124%
100-110	+124%
110-120	+182%
120-130	+124%
130-140	+41%
140-150	-20%

Pressure by velocity transfer functions were obtained to provide an additional means to validate the effectiveness of the AAS. Pressure by velocity equals the impedance of the speaker, Z.

$$\frac{P}{v_{spkr}} = Z \tag{6}$$

A successful controller would indicate a change in impedance between the controlled and uncontrolled state. For the velocity to accurately track the pressure, the impedance of the controlled speaker would have to equal one. This corresponds to a pressure by velocity transfer function with 0 dB gain and 0 degree phase over the bandwidth of the controller. A comparison of the pressure by speaker velocity transfer function for the uncontrolled (Figure 14) and the controlled speaker (Figure 15) validates the prediction that the AAS controls the impedance of the speaker. The pressure by velocity transfer function for the controlled speaker (Figure 15) has gain between 1 and -3 dB and phase between 0 and 45 degrees over a 65-120 Hz bandwidth. The pressure by speaker velocity transfer function (Figure 14) for the uncontrolled speaker closely resembles the drive velocity by speaker velocity transfer function (Figure 16) obtained for the uncompensated audio speaker.

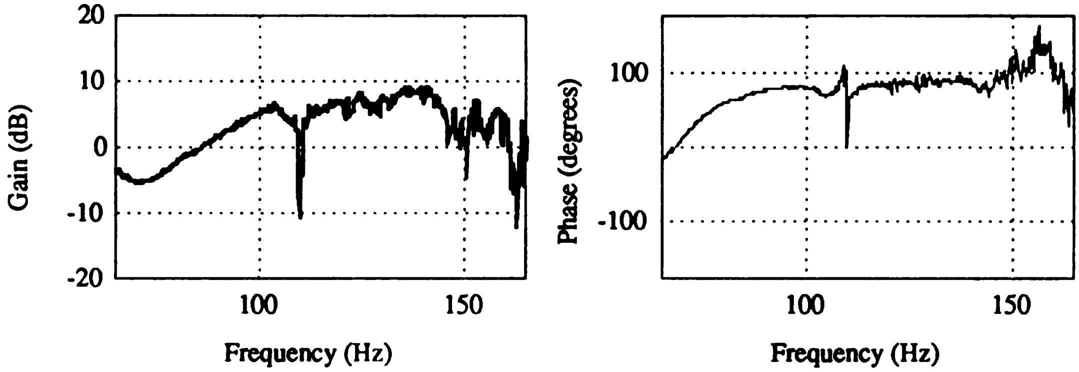

Figure 14. Uncontrolled System Pressure / Velocity Frequency Response

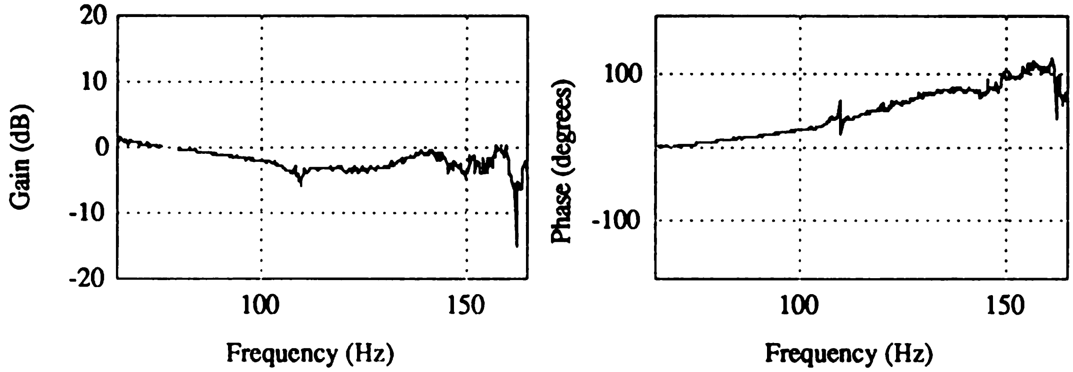

Figure 15. Controlled System Pressure / Velocity Frequency Response

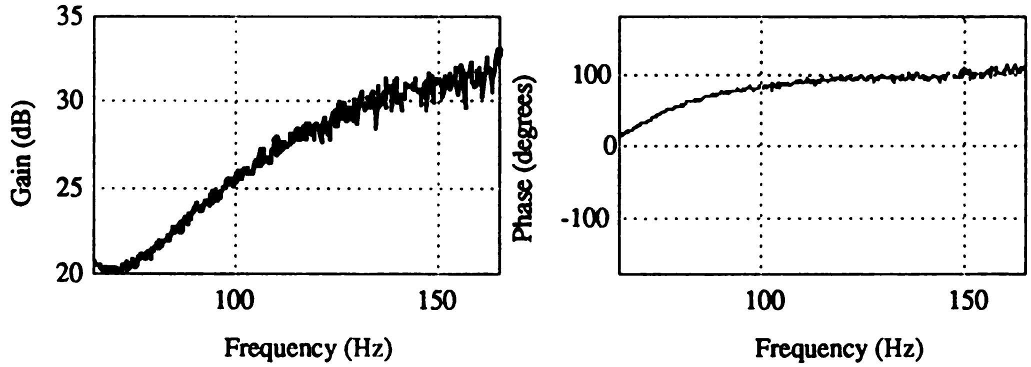

Figure 16. Speaker Frequency Response

Sound pressure level divided by acoustic intensity is a final method to confirm the effectiveness of the AAS. Simplification of the ratio shows that this expression represents the impedance of the speaker.

$$\frac{SPL}{I}=\frac{P^2}{P\cdot v}=\frac{P}{v}=Z \tag{7}$$

The ratio of sound pressure level to acoustic intensity was calculated over a 65-100 Hz bandwidth. A polynomial was fit to the measured data for both the uncontrolled and controlled states. The best fit polynomial for the uncontrolled system (Figure 17) shows in increasing value of SPL divided by acoustic intensity which tracks an increase of the uncontrolled pressure by velocity frequency response (Figure 14). Similarly, the best fit polynomial for the controlled system (Figure 18) shows a decreasing value of SPL divided by acoustic intensity which tracks a decrease of the controlled pressure by velocity frequency response (Figure 15).

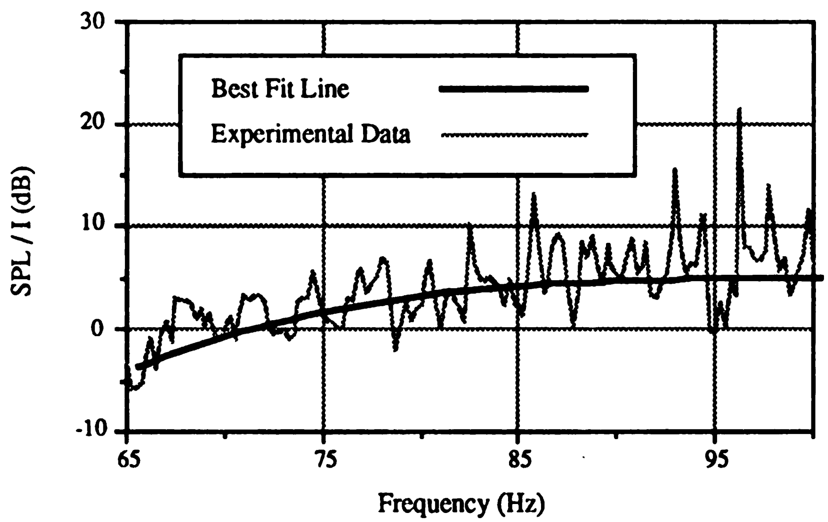

Figure 17. SPL / I for Uncontrolled System

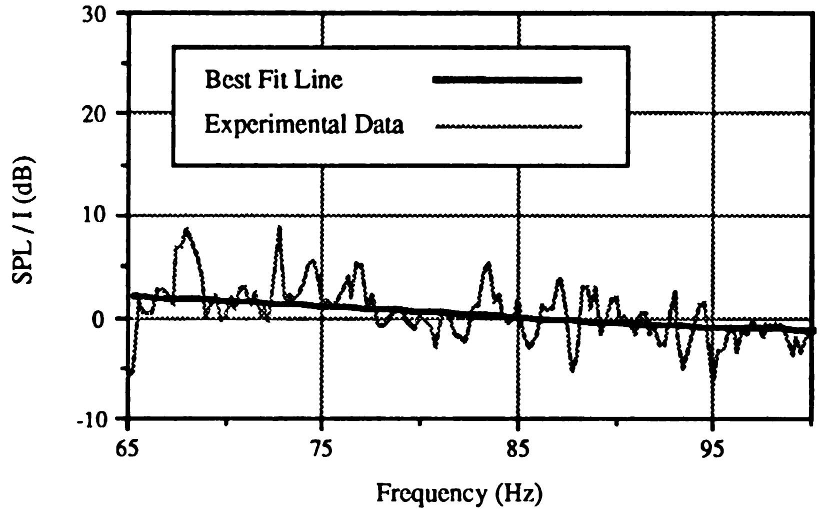

Figure 18. SPL / I for Controlled System

CONCLUSIONS

A prototype Active Acoustic Sink was constructed to determine if an electronically driven speaker can be built to absorb acoustic energy. A speaker was successfully compensated using speaker velocity feedback. The compensated speaker had 1 dB of gain variance and only 30 degrees of phase shift over a 20-120 Hz bandwidth. Through a comparison of speaker velocity and speaker back emf, it was determined that the speaker back emf, e_b, does not provide an accurate representation of the speaker cone velocity above 120 Hz. The compensated speaker was used as a control actuator for the AAS, and the effectiveness of the AAS was determined by measuring the acoustic intensity at the face of the speaker. The impedance of the controlled speaker was measured and found to be close to the desired value of one. This value of the impedance ensured that the velocity was accurately tracking the measured pressure. The results indicate that there is an increase (up to 182%) of acoustic energy absorption using the AAS over a 65-120 Hz bandwidth.

LIST OF REFERENCES

LIST OF REFERENCES

Colloms, M. *High Performance Loudspeakers*. New York: John Wiley & Sons, 1985.

Everest, F.A. *The Master Handbook of Acoustics*. Blue Ridge Summit, PA: Tab Books, 1989.

Olson, H.F. and E.G. May. "Electronic Sound Absorber". The Journal of the Acoustical Society of America, Volume 25, Number 6, November 1953.

Ogata, Katsuhiko. *Modern Control Engineering*. New Delhi: Prentice Hall of India, 1986.

Radcliffe, C.J., and S.D. Gogate. "Identification and Modeling of Speaker Dynamics for Acoustic Control Applications". ASME WAM Symposium on Active Control of Noise and Vibration, 1992.

APPENDIX A

APPENDIX A - MATERIALS REQUIRED AND WIRING SCHEMATIC

The following materials were required for the construction of the AAS:

Speaker Compensation

(1) 400 W Resistor
(1) 10 kW Resistor
(6) 1 kW Resistors
(4) 22 μF Capacitors
(4) 741 Operational Amplifiers
(1) Realistic 40W Stereo Power Booster
(1) 12" Realistic Subwoofer w/ enclosure

Feedback Controller

(3) 1 kΩ Resistors
(1) 2 kΩ Resistors
(2) 22 μF Capacitors
(2) 741 Operational Amplifiers
(1) Electret Tie Pin Microphone

Miscellaneous

Breadboard
Micronta Regulated 12 V Power Supply
Misc. BNC Connectors
Misc. Wires

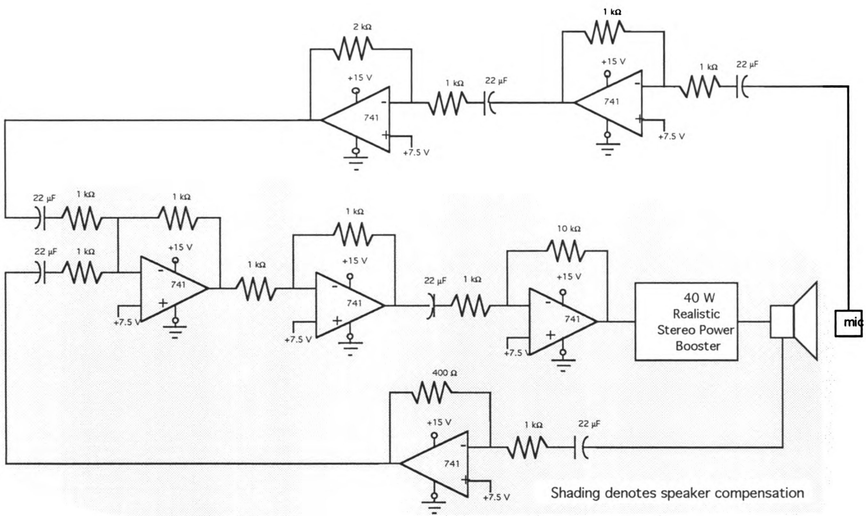

Figure A1. Wiring Schematic of Active Acoustic Sink