Active Noise Control of a Baffled, Small Centrifugal Fan

John K. Boyle

Senior Capstone Report

Department of Physics and Astronomy

Brigham Young University

June 2012

ABSTRACT

This paper presents an experimental verification of the error sensor placement theory developed by Esplin (2012) for control of the blade passage frequency (BPF) tone in the exhaust duct of a notebook computer centrifugal fan. As the theory modeled the fan and duct as a baffled, closed-open, rectangular duct, such a duct was constructed. Small receivers were placed in the side of the duct at locations similar to those of the primary noise source of the fan and the secondary (control) source used in the model. Maps of the pressure difference in the plane of the baffle between the pressure field of the primary source and the field corresponding to the predicted minimum sound power condition were used to guide error sensor placement. Radiated sound power reduction of tones in the baffled, closed-open duct was measured and showed good agreement with predicted values. The model predictions were applied to the real fan and exhaust duct, and sound power reduction was measured. The observed trend of power reduction over frequency follows prediction, and suggests the simplified model of the fan assembly is an effective approximation for ANC purposes.

List of Figures:

Figure 1 - Fan, duct, and heat-sink used in the experiment. 9

Figure 2 - Isometric view of the ideal system. 9

Figure 3 - Pressure difference between uncontrolled and minimum sound power condition upon baffle surface. 11

Figure 4 – Predicted sound power reduction for the baffled duct. 12

Figure 5 – Thirteen microphone rotating boom. 14

Figure 6 – Measured control of a 1 kHz tone in the ideal duct. 15

Figure 7 – Predicted and measured power reduction for the ideal duct. 16

Figure 8 – A comparison of the predicted pressure patterns for 1000 and 1700 Hz 16

Figure 9 – Measured control of the BPF tone at 1008 Hz in the fan assembly. 17

Figure 10 – Effect of non-ideal error sensor location. 18

Introduction

Notebook computer components often generate large amounts of heat. The small centrifugal fan in most notebook computer models dissipates this heat by drawing air through the interior of the computer and across hot components. However, as notebook computers become thinner, the airflow resistance increases and a fan must operate at higher speeds to maintain a given flow volume. The noise generated by a fan also increases with fan speed and becomes more noticeable to a computer user due to the increased levels of tones (Baugh, 2008). Fan designers have adjusted clearances and fan blade shaping to reduce tone generation, which unfortunately increases the broadband noise. It is a tradeoff made in favor of reducing the noise that is most noticeable and most annoying.

Noise reduction methods for notebook computer fan noise fall into two main categories, passive and active. Passive noise reduction techniques involve damping, absorption and isolation, none of which are well suited to notebook computers. Recent developments in active noise reduction techniques offer a potential compact and lightweight solution required for notebook computers. This paper presents an experimental verification of such a method that was developed by Esplin (2012), which is based on the work of Sommerfeldt and Gee (2007).

Background

Active noise control

Sound may be attenuated through the principle of superposition. Two sound waves of equal amplitude and opposite phase will sum to zero where they interact. A noise control system emits this wave of opposite phase. The most basic system would consist of a single frequency noise source and a signal generator driving a loudspeaker, known as the control source, as it results in the control of the noise. (The noise source is known as the primary source.) A user of this system would manually adjust the phase and amplitude of the signal from the signal generator (the control signal) to provide the opposite, cancelling sound wave. However, in a large space such as most rooms, the noise is not likely to be globally cancelled as the waves are not travelling in a single direction. The tone is reflected in many directions from many different surfaces and in order to attenuate the tone throughout the volume, the sound field emitted by the control source would need to spatially match the sound field generated by the noise source, but with opposite phase.

If noise reduction or control is required at a specific location, a microphone, called an error sensor, may be added to the noise control system. The operator would place the error sensor somewhere in the room, observe the measured noise level and adjust the control source until the noise is minimized at the error sensor. Unfortunately, this may cause the noise to be increased in some areas despite attenuating it others.

The problem of reflections and multiple waves is less complex in a more confined system where wave propagation is only two- or one-dimensional. In the one-dimensional case, such as in a tube where the noise is below the cut-off frequency, there is effectively only one wave to be cancelled and a manual noise control system as mentioned before can be effective. However, even in this simple system, if the frequency or amplitude of the noise source were to fluctuate, the system operator would not be able to adjust the cancelling wave's phase and amplitude quickly enough. This is one of the many reasons that *adaptive* noise control systems were developed.

An adaptive noise control system is similar to the manual system just described except that it is capable of responding to changes in the noise source, typically through the use of a digital control system. The controller, often based on an embedded microprocessor, automatically adjusts the control signal(s) to minimize the noise at each error sensor. Another signal, called the reference signal, is input to the controller and represents a signal that is correlated with the signal to be attenuated. In the adaptive active control of fan noise, the most prominent tonal noise is proportional in frequency to the fan rotation speed. Therefore, the fan tachometer signal is often used as a reference signal.

As mentioned before, a limitation of the basic control system is that the noise attenuation may not be global. However, through optimization of control source arrangements, error sensor locations, and adaptive control algorithms, researchers have developed ANC systems that provide noise reduction over large areas and over a wide range of frequencies (though these two qualities are often mutually exclusive). In the context of the previous example, these techniques allow the noise to be attenuated "at the source." (See Martin and Roure, 1998, and Chen, *et al.*,

2010) Another advantage of such methods is that they often rely on control sources placed in the near-field of the primary source and thereby allow for compact control systems.

Large Centrifugal Fans (Blowers)

Work on centrifugal fan noise has primarily focused on fans larger than the fan used for this study. These fans are often called blowers and are frequently used for industrial ventilation. The primary tone generated in a centrifugal fan is at the blade passage frequency (BPF). It is equal to the number blades that pass by a fixed portion of the fan, such as the cutwater, in one second (the wedge-shaped protrusion in the fan housing that "cuts off" the air rotated by the blades and forces it into the exhaust duct).

Noise reduction techniques described in the literature include adjusting fan parameters such as the distance between blade tips and the cutwater, lining the fan housing with absorptive material, and placing resonators at the cutwater (Neise, 1982). Reductions at the blade passage frequency in both the inlet and exhaust ducts of the fan were shown through the placement of active control actuators placed in the fan cutwater (Koopmann, Neise, and Chen, 1988). A more recent study (Wu & Bai, 2001) demonstrated adaptive control of the BPF tone in a fan exhaust duct using control sources placed in the duct, and also control of the BPF tone for an un-ducted fan with the control sources mounted in the cutwater. In both cases, control was first tested on a synthetic tone, with the fan turned off. Control was adversely affected with the fan in operation, which the authors suggested was a result of air flow perturbing the propagation path from the noise source to the control speakers, especially at higher flow speeds.

Axial Fans

Small axial fans are common in information technology products, such as the computer "towers" of most desktop personal computers. Research on active control of axial fan noise (Gee and Sommerfeldt, 2003, 2004, Sommerfeldt and Gee 2007) demonstrated extreme near-field placement of error sensors and control actuators that gave consistent global reductions in tonal components of axial fan noise.

Their method is as follows: to predict effective error sensor locations, an expression for the radiated sound power of the system in question is derived through source coupling following Nelson *et al.* (1987). This expression is then minimized by taking the derivative with respect to the real and imaginary parts of the complex control source strength(s). Each resulting expression is set equal to zero and the roots give the complex source strength(s) that causes the minimum sound power radiation condition. As an example, for two point sources in a free field that are closely spaced with respect to a wavelength, the secondary source strength corresponding to the minimum sound power radiated would be nearly equal in amplitude and opposite in phase to that of the primary source. This result is intuitive even without the use of a source-coupling sound power expression, but as source configuration and boundary conditions become more complex, the sound power expression becomes useful.

If the noise source had a fixed complex source strength (fixed amplitude and phase), the predicted control actuator source strengths could be used to adjust each control actuator relative to the noise source and thereby cause the minimized sound power condition. Manual adjustment is generally not feasible for control of fan noise, and an adaptive control system is required.

When this method is applied to a given system, it is generally the placement of error sensors that will determine the effectiveness of control. Therefore, the predicted source strength of the control source(s) relative to the primary source is used to calculate the corresponding acoustic pressure magnitude in the plane of the sources. Through examination of the pressure magnitude difference between the field of the noise source and that of the predicted minimum sound power condition, regions of greatest pressure magnitude difference are seen that Sommerfeldt and Gee (2007) graphically used to determine error sensor placement. When error sensors were placed in non-ideal locations, the control system became significantly less effective.

Shafer, Gee, and Sommerfeldt (2010) used near-field pressure measurements to show that the predicted pressure reduction patterns are in fact produced when error sensors are placed upon the predicted relative nulls. They also demonstrated that the predicted ideal error sensor locations are much wider in practice and that because the primary and control sources are not true monopoles, some non-ideal error sensor locations could lead to effective global control of the BPF tone and its first four harmonics.

Small Centrifugal Fans

Only one study in the published literature demonstrates the application of active noise control to notebook computer fan noise. Cordourier-Maruri and Orduña-Bustamante (2009) used the speakers of a notebook computer as control sources and microphones placed at the user's ear and at the fan exhaust as the error sensor and reference sensor, respectively. They did not achieve any reduction of the tonal noise at the blade passage frequency of the fan, citing the spectral width of the blade passage frequency as being "out of range of effective operation of the [control] system."

Recently, Esplin (2012), developed an acoustic model of the exhaust duct of a notebook computer centrifugal fan, following the methods of Sommerfeldt and Gee (2007). No other similar models are found in the literature. As this present study is an experimental verification of the model predictions, the model is described in the following section.

Theory

This section is a summary of the acoustic model of the exhaust duct of a small centrifugal fan typical of a notebook computer as developed by Esplin (2012), and its application to control of the BPF tone of the fan in question.

Sources and Exhaust Duct Radiation Impedance

A typical notebook computer fan, duct and heat-sink of dimensions 1x4.8x10cm, seen in Figure 1, are modeled as a baffled, closed-open duct of these same dimensions, as shown schematically in Figure 2.

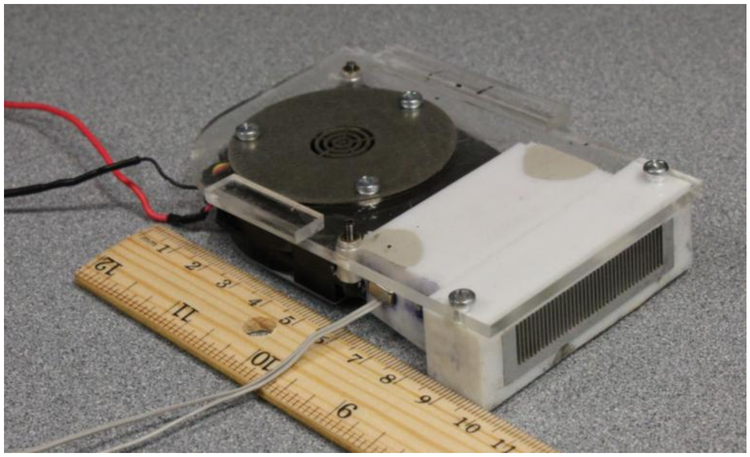

Figure 1 - Fan, duct, and heat-sink used in the experiment and also as the basis for the model. Fan assembly dimensions are nominally 1x4.8x10 cm.

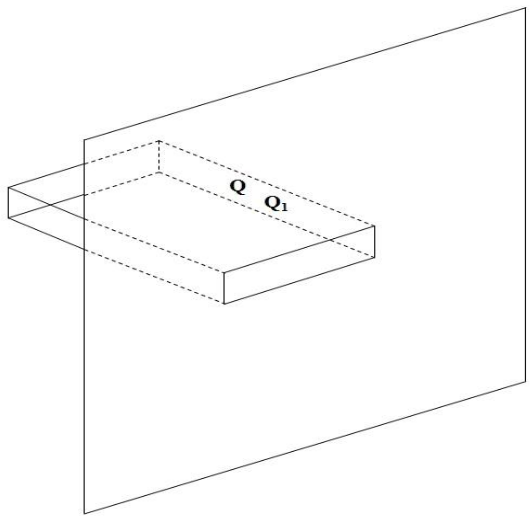

Figure 2 - Isometric view of the ideal system with a primary source Q and control source Q_1. Duct interior dimensions are 1x4.8x10 cm.

Fan inlets and any corresponding effects due to the interior field of the notebook computer are ignored. Point sources were used to model the primary and control source because, at standard operating voltages, the BPF tone of the fan used in this study is near 1 kHz and therefore all sources are much smaller than a wavelength. The primary source was placed upon one interior side of the duct, at a distance from the opening that corresponds to the location of the fan cutwater, the effective source of the BPF tone in the fan. The single control source was also modeled as a point source, located 5 mm downstream from the primary source, a location chosen due to the difficulty of placing a secondary source within the fan housing. In the experiment, this location is in the duct between the fan and heat-sink.

Although in the model the primary source is placed in the side of the duct, the cutwater in the fan is located several millimeters away from the side of the fan. This difference in source placement between the fan and the model is a small fraction of a wavelength; thus the effect will be minimal.

A modal coupling radiation impedance boundary condition developed by Kemp, Campbell, and Amir (2001), was applied at the duct opening after preliminary calculations showed that the use of the radiation impedance of a baffled rectangular piston gave unphysical results.

Damping

Absorption losses were included in the model by making the wavenumber in the source expressions complex, i.e. $k^* = k(1 + j\alpha_s)$, where k^* is the complex wavenumber, k is the wavenumber and α_s the damping term. This leads to a term in the expression for acoustic pressure of $e^{-\alpha_s kx}$. Preliminary measurements of the frequency response of the simple closed-open duct constructed for the experiment and of the fan assembly indicated that a value of $\alpha_s = 0.05$ was appropriate to represent the damping.

Error Sensor Placement Guides

The expression for sound power radiated from the duct opening was derived and then minimized by taking the partial derivative with respect to the real and imaginary parts of the complex control actuator source strength, setting each expression equal to zero and solving for the complex source strength. After this analysis, the inclusion of the predicted complex control actuator source strength in the expression for acoustic pressure gives the predicted pressure field which results from minimizing the radiated sound power.

Acoustic pressure along the surface of the baffle in this minimized sound power condition was calculated and then compared to the case of the uncontrolled noise source. The difference between the two was calculated and the plotted values are used to guide error sensor placement. The theory behind this is as follows: the adaptive control system adjusts amplitude and phase of the control source to drive squared pressure at the error sensor to zero.

The regions of greatest acoustic pressure reduction in the minimized sound power condition are shown in Figure 3 .

This pressure pattern represents the unique solution for minimized sound power for this specific system, and again, is a result of adjusting the complex source strength of the secondary source. With an error sensor placed in the predicted region of greatest acoustic pressure reduction, the ANC system will seek to minimize squared pressure there by adjusting the control source phase and amplitude, and thereby yield the minimized sound power condition. The theoretically ideal error sensor locations are found along the relative null seen in Figure 3.

Cunefare and Koopman (1991) showed that in following the method of Nelson *et al.* (1987), the minimum sound power condition for a given source configuration is due to an increase in the combined source complexity and thereby decreased sound power radiation, i.e. a dipole-like source radiates more like a quadrupole when control sources are added. In this case, the minimized sound power condition is brought about by the creation of dipole-like pressure behavior at the end of the duct, as seen within the black rectangle in Figure 3.

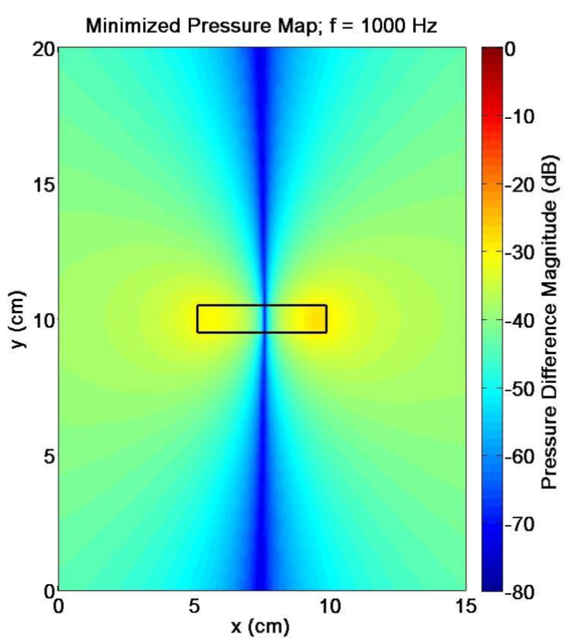

Figure 3 - Pressure difference between uncontrolled and minimum sound power condition upon baffle surface. The black rectangle is the duct opening. The duct extends into the page.

Predicted Sound Power Reduction

The predicted reduction of the sound power radiated from the baffled duct opening is shown in Figure 4. The peak at 775 Hz is due to the first mode of the duct; the peak seen near 2000 Hz is for the second mode.

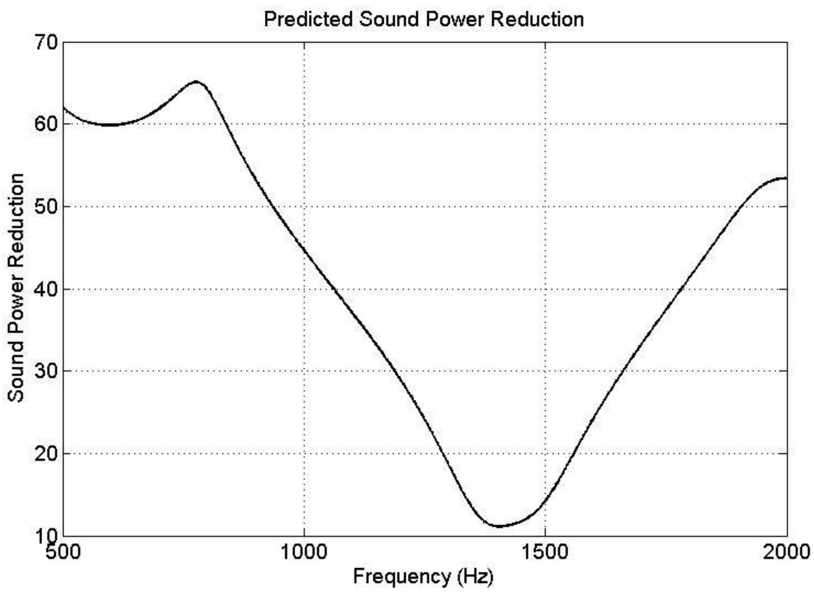

Figure 4 – Predicted sound power reduction for the baffled duct.

Predicted sound power reduction is least near 1500 Hz. At that frequency and for the given source configuration, the uncontrolled primary-source pressure field exhibits dipole-like radiation at the duct opening. As a dipole-like pressure pattern at the duct opening radiates less sound power than a plane-wave pattern, the control source is less able to affect sound power radiation of an already inefficient radiator.

The frequency at which this dipole-like behavior occurs is dependent on the primary source location along the length of the duct. As the primary source is moved away from the opening, this frequency increases, and it decreases as the source is moved toward the duct opening. Accordingly, sound power reduction could be improved at certain frequencies by varying the location of the primary source along the length of the duct. In this case, reduction of the BPF tone is less effective near 1500 Hz, but this could be improved if the primary noise source could be located closer to the duct opening.

An additional finding of note in this analysis is that varying the control source location, while the primary source location is fixed, causes no change in the predicted power reduction curve. This potentially suggests that, at least within the range of frequencies examined (500-2000 Hz), the control source could be located anywhere along the length of the duct without affecting reduction.

EXPERIMENT

Experimental Apparatus

Ideal Duct and Fan Assembly

A closed-open rectangular duct was constructed of medium-density fiberboard with internal dimensions of 1x4.8x10 cm. Two Sonion 20x22 0935 hearing aid receivers of dimensions 9x7x4 mm were flush-mounted in the side of the duct, one as the primary source at 4.25 cm from the opening, another as the control source at 3.75 cm (see Fig. 2).

The second phase of the experiment involved control of the BPF tone of the fan assembly shown in Figure 1. The fan is a SEPA HYB60A05A. The control source was another Sonion hearing aid receiver, flush mounted in the duct portion.

ANC System

The ANC system is based on the multichannel filtered-x algorithm (Widrow and Stearns, 1985). The fan emits two tachometer pulses per revolution that were multiplied by the number of fan blades, thirteen, and divided by two, the number of pulses per revolution, to give the BPF tone reference signal for the ANC system. Krohn-hite 8-pole filters were used to condition the reference and control signals. The error sensor was an electret microphone of diameter 6mm.

Rotating Microphone Boom and Baffle

The radiated sound power from the duct or fan was measured with a 0.6 m radius semi-circular microphone boom with thirteen, half-inch Gras 40AE free-field microphones, spaced at equal angles for the calculation of sound power from squared pressure. The boom was controlled with a LabVIEW VI and rotated by a stepper motor in 15 degree increments. A 2.2 meter square baffle was in-place for the measurements. See Figure 5.

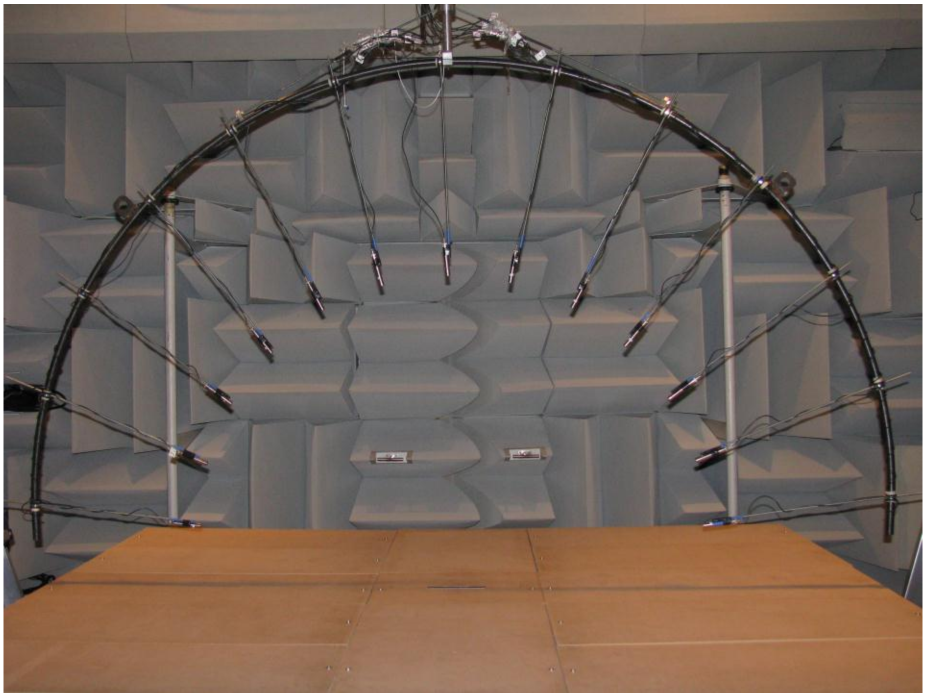

Figure 5 – Thirteen microphone rotating boom used to make sound power measurements. The baffle is 2.2 x 2.2 m.

Experimental Methods

The ideal duct and the fan assembly were each mounted in the baffle, opening upward toward the microphone boom. For the ideal duct, the primary source was driven with a single tone to simulate the BPF tone from 1000-1700 Hz in 100 Hz increments. The fan was operated to give a BPF tone of 1000 and 1100 Hz. As explained previously, maps of the predicted pressure reduction for a given frequency were used as error sensor placement maps (Figure 3).

Sound power reduction was measured in two steps. First, the ideal duct or fan assembly was placed in the baffle and the microphone boom measured the field of the noise source alone. Second, the ANC system was then turned on and another measurement was made to measure the controlled field. Each set of measurements was processed and compared to compute the sound power difference and to create a plot of the sound pressure level as a 3D surface.

Results and Discussion

Ideal Duct Driven with Tone at BPF

The primary source in the ideal duct was driven at frequencies between 1000-1700 Hz in 100 Hz increments, and sound power reduction was measured. A typical measured value at 1 kHz is shown in Figure 6 where an average global reduction of 37 dB was achieved.

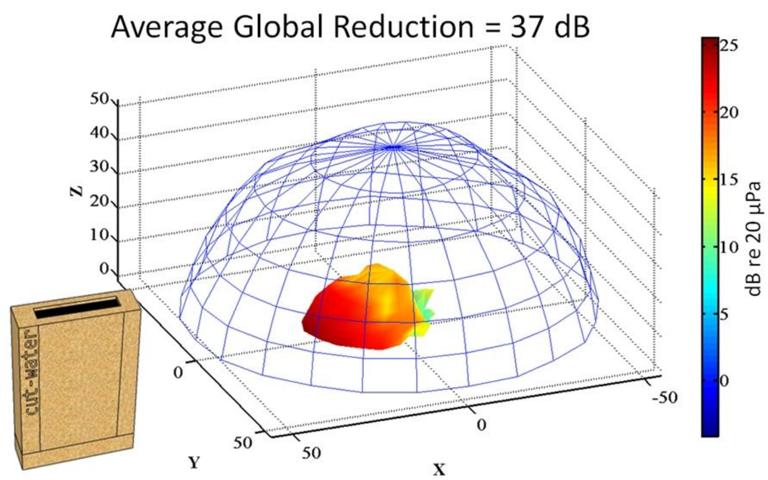

Figure 6 – Measured control of a 1 kHz tone in the ideal duct. Each point of the mesh represents the sound pressure level recorded as the boom was rotated. The inner surface is the interpolated surface from the sound pressure level of the controlled field. The duct at the side is to show the orientation of the duct in this measurement.

The mesh surface is created from mapping the uncontrolled sound pressure level at the frequency of interest, at each point in the boom rotation. The radius of the mesh is proportional to the sound pressure level in that direction. The colored surface is the interpolated mesh of sound pressure levels of the controlled field, where both the radius and color correspond to the value. Power reduction is the average difference between the two surfaces. Control is fairly uniform and therefore global. Measured and predicted values as a function of frequency are shown in Figure 7. Predicted reduction is lowest near 1500 Hz as explained in the Theory section.

The spatial average introduced by the finite width error sensor microphone is likely the cause of the difference between measurement and prediction seen at 1600 and 1700 Hz (Figure 7 and Figure 8). The error sensor microphone was of diameter 6mm. Allowing for a small error of about 2mm in error sensor placement, the average value of reduction across the 10 mm that span the null gives an indication of the "depth" and width of the predicted null. At 1000 Hz, the average value is -48 dB, and for 1700 Hz, -36 dB, e.g. at 1000 Hz the null is deeper (there is greater predicted reduction along it) and wider. Thus, if the error sensor were placed slightly off of the nodal line of 1000 Hz, there would be a lesser effect on the reduction attained than that at 1700 Hz.

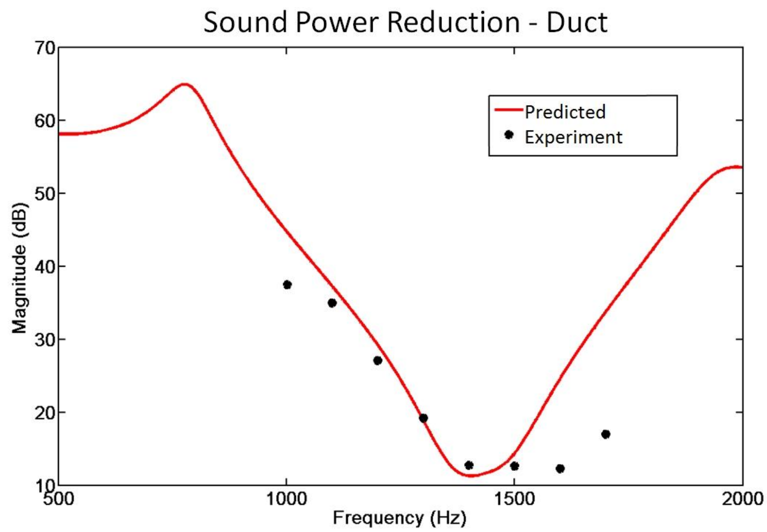

Figure 7 – Predicted and measured power reduction for the ideal duct.

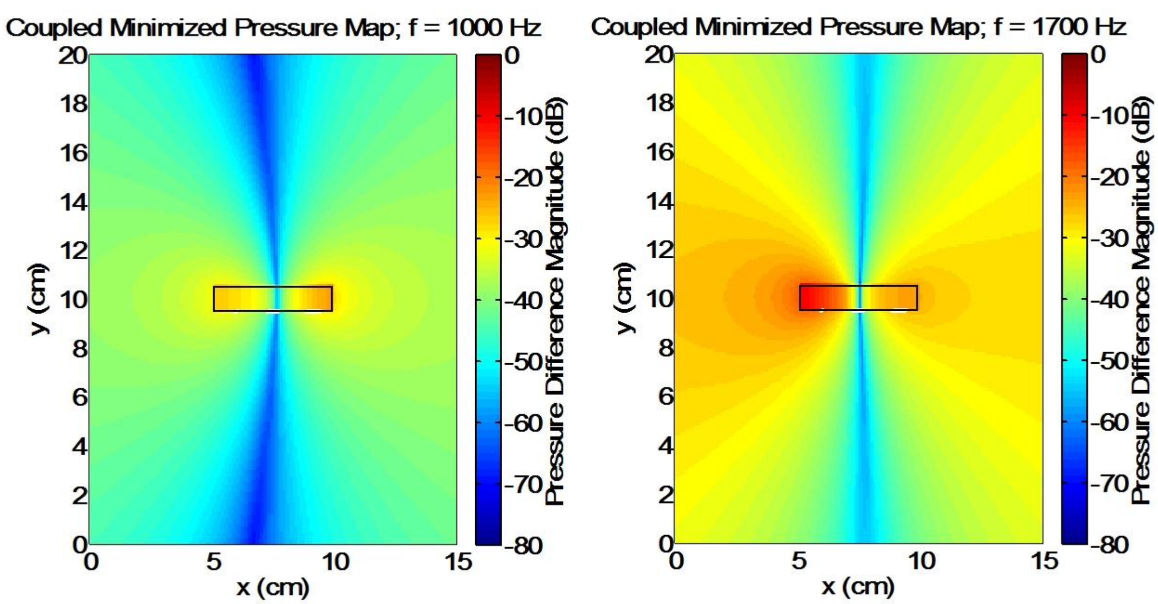

Figure 8 – A comparison of the predicted pressure patterns for 1000 and 1700 Hz. The relative null of 1000 Hz is wider and deeper than for 1700 Hz, which makes the effect of inexact error sensor placement smaller than at 1700 Hz.

Control of BPF Tone in the Fan

Figure 9 shows a typical measured reduction of the BPF tone of the fan assembly, specifically at 1008 Hz. For this figure, the control was implemented for three separate trials, and the results were averaged to give average reduction. The average spatially-averaged value of reduction over these three measurements is 17.1 dB. As in Figure 6, reduction is fairly uniform over the entire hemisphere. The rough controlled surface is a result of the tone being reduced to the level of the broadband flow noise, where it has a more random directivity than the uncontrolled field (the outer mesh). At 1100 Hz, the measured sound power reduction, again averaged over three measurements, was 16.7 dB. At both frequencies, the uncontrolled BPF tone level as measured by the error sensor and the boom microphones is about 13 dB above the broadband flow noise. Also in each case, the tone at the BPF, as measured by the error sensor, was reduced by about 35 dB.

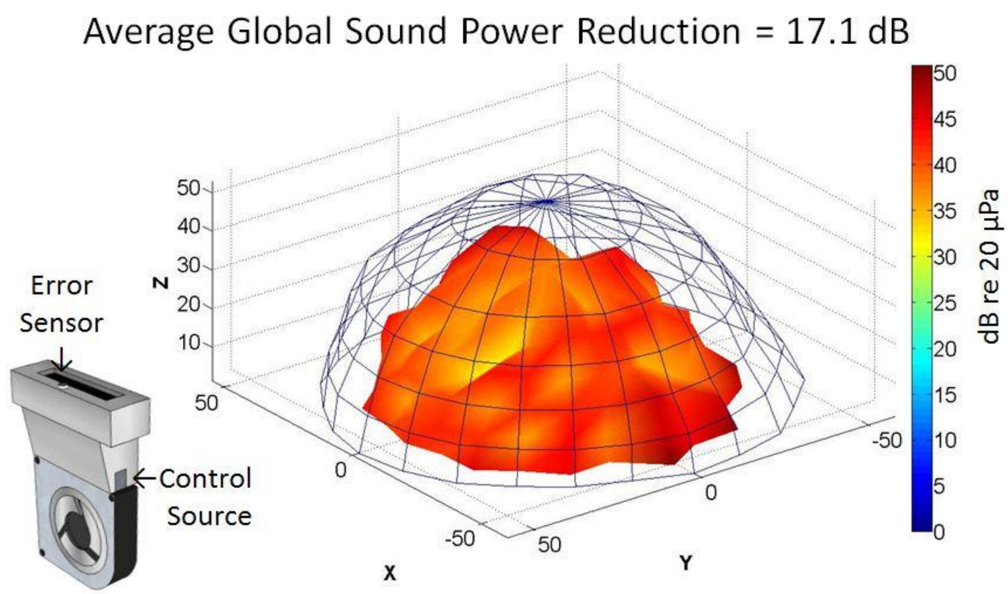

Figure 9 – Measured control of the BPF tone at 1008 Hz in the fan assembly. The fan model on the left is to show the fan orientation in the measurement.

Sound power reduction for control of the BPF tone in the fan assembly is about 23 dB less than predicted (Figure 4). Reduction is likely limited as the BPF tone is only about 13 dB above the broadband noise of the fan. Another potential cause of this discrepancy are the differences between the model and experiment, such as the distributed nature of the BPF tone generation mechanisms in the fan, as well as the fact that the cutwater (which represents the effective noise source) is not located exactly along the side of the fan housing as it is in the model and the ideal duct. Finally, the model does not account for the fan inlets.

It is possible that because of these factors, the ideal error sensor locations vary from those predicted by the current model. However, this variation cannot be estimated with the current model and despite this source of error, the model nonetheless appears to accurately predict the trends of proper error sensor locations.

Ideal Error Sensor Location

To test the effectiveness of predicted error sensor locations, the error sensor was placed to the side of the ideal duct opening. A loss of 10 dB of sound power reduction was measured, seen in Figure 10 (compare with Figure 6). This effect was not investigated in the fan assembly.

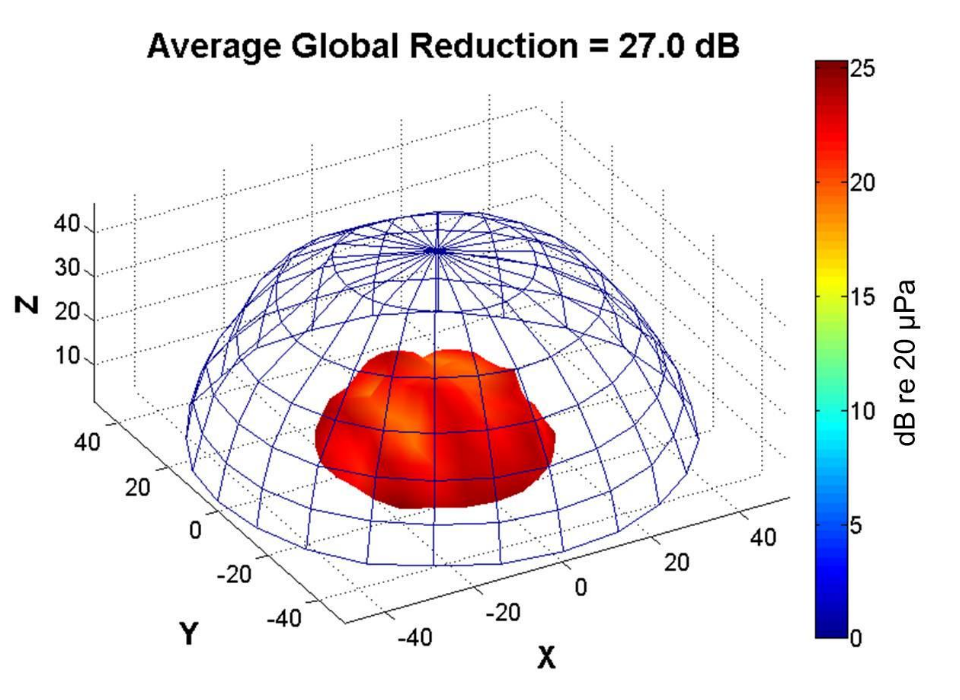

Figure 10 – Effect of non-ideal error sensor location. Measured control of 1 kHz tone in the ideal duct, with the error sensor placed in a non-ideal location at the side of the duct opening rather than immediately above the middle of it (where the nodal line lies) as in previous results (Figure 6).

Conclusion

The methods developed by Sommerfeldt and Gee (2007) for the active control of axial fan noise that were applied by Esplin (2012) to a small centrifugal fan exhaust duct typical of a notebook computer, have been experimentally verified. Global reduction of the BPF tone was shown for a short closed-open duct and a fan assembly of the same dimensions. The placement of the error sensor in a non-ideal location caused a loss in measured reduction of the BPF tone. The predicted trend of sound power reduction over frequency is observed and suggests that the model developed by Esplin (2012) is an effective approximation, for ANC purposes, of the fan assembly.

Further work would involve integrating this model into a model of a complete notebook computer system. The radiation paths from the notebook computer interior are much more complex than the exhaust duct but a similar approach could provide accurate error sensor locations in the extreme near-field of the fan inlets. Also, the model could potentially be used for active noise control of HVAC systems and other ducted noise sources.

References

Baugh, E. (**2008**). "Acoustic limitations in notebook thermal design." Electronics Packaging Technology Conference 2008 10th, 9-12 Dec. 2008, pp. 725-730.

Chen, W., Pu, H., Qiu, X. (**2010**). "A compound secondary source for active noise radiation control," Appl. Acous., **71,** pp. 101-106.

Cordourier-Maruri, H. A., Orduña-Bustamante, F. (**2009**). "Active control of periodic fan noise in laptops: spectral width requirements in a delayed buffer, implementation," J. Appl. Res. Technol. **7**, pp. 124-135.

Cunefare, K. A., Koopmann, G. H. (**1991**). "Global optimum active noise control: Surface and far-field effects," J. Acoust. Soc. Am. **90**, pp. 365-373.

Esplin, J. (**2012**). "Active noise control of a centrifugal fan mounted in a mock laptop enclosure," (Master's Thesis). Brigham Young University.

Gee, K. L., Sommerfeldt, S. D. (**2003**). "A compact active control implementation for axial cooling fan noise," Noise Control Eng. J. **51**, pp. 325–334.

Gee, K. L., Sommerfeldt, S. D. (**2004**). "Application of theoretical modeling to multichannel active control of cooling fan noise," J. Acoust. Soc. Am. **115**, pp. 228–236.

Kemp, J. A., Campbell, D. M., and Amir, N. (**2001**). "Multimodal radiation impedance of a rectangular duct terminated in an infinite baffle," Acustica, **87**, pp. 11-15.

Koopmann, G. H., Neise, W., Chen, W. (**1988**). "Active noise control to reduce the blade tone noise of centrifugal fans," ASME, Trans., J. Vib. Acous. Stress and Reliability in Design **110**, pp. 377-383.

Neise, W. (**1982**). “Review of noise reduction methods for centrifugal fans,” J. Eng. for Indus. **104**, pp. 151-161.

Nelson, P. A., Curtis, A. R. D., Elliot, S. J., Bullmore, A. J. (**1987**). “The minimum power output of free field point sources and the active control of sound,” J. Sound Vib. **116**, pp. 397–414.

Martin, T., Roure, A. (**1998**). ‘‘Active noise control of acoustic sources using spherical harmonics expansion and a genetic algorithm: Simulation and experiment,’’ J. Sound Vib. **212**, 511–523.

Shafer, B. M., Gee, K. L., Sommerfeldt, S. D. (**2010**). “Verification of a near-field error sensor placement method in active control of compact noise sources,” J. Acoust. Soc. Am. **127**, pp. EL66-EL72.

Sommerfeldt, S. D., Gee, K. L. (**2007**) “Multi-channel active control system and method for the reduction of tonal noise from an axial fan,” U.S. Patent No. 7,272,234 (September 18, 2007).

Widrow, B., Stearns, S.D. (**1985**). <u>Adaptive Signal Processing</u>, Prentice Hall, NJ, pp. 288-294

Wu, J.-D., Bai, M.-R. (**2001**). “Application of feedforward adaptive active-noise control for reducing blade passing noise in centrifugal fans,” J. Sound Vib. **239**, pp. 1051-1062.

Acknowledgements

I want to thank Dr. Scott Sommerfeldt and Dr. Kent Gee for inviting me to participate in this project, and for being generous with their time, attention, and resources. Their guidance and teaching made this work possible. I am also indebted to James Esplin for his advice and guidance on experiments. He has been an invaluable teacher to me throughout this project.

This work would not have been possible without the love and support of my dear wife. She has been patient and understanding on the long days of experiments. She and my daughter make life wonderful and happy.

Finally, I thank my God. He has "[poured] down knowledge from heaven" upon my head. Any contribution I have made to the body of science is through His grace.