# A compact active control implementation for axial cooling fan noise

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**A multi-channel active control system has been applied to the reduction of free-field tonal noise from a small axial cooling fan, typical of those used in personal computers and other technology. Emphasis has been placed on the development of a compact, practical system, with loudspeakers and error microphones located close to the fan, and which is capable of achieving significant global attenuations for the first four harmonics of the blade passage frequency. In addition, the reference sensor and loudspeakers used have been selected based upon their relatively low cost. Experimental results are presented for five different actuator combinations and tonal mean-square pressure reductions are calculated in each case. A comparison of the performance of each configuration indicates that consistent global control of the four targeted harmonics may be achieved with three control channels.** 

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## 1. INTRODUCTION

Axial-flow fans are required to prevent the electronic components in computers, overhead and computer projectors, photocopiers, and other related technology from overheating. Cooling fan noise radiation constitutes a potential source of annoyance and disruption of concentration, as background noise levels are often increased. The radiation from these fans consists of a broadband spectrum upon which are superimposed one or more tones. The tonal component, normally the dominant portion of the spectrum, is typically comprised of the blade passage frequency (BPF) and harmonics, and is the result of spatially unsteady loading on the fan blades as they rotate.[1] The variation in loading is usually caused by flow obstructions near the inlet or exhaust of the fan and the nature of these obstructions directly affect the number and levels of the tones.

Because the tonal portion of the fan's spectrum is usually dominant, reduction of this part of the noise is initially more important than attenuation of the broadband portion, especially because of the increased sensitivity of the human ear to tonal noise.[2] Implementation of passive noise control measures is made difficult for two reasons. One standard practice to reduce radiated tonal noise is to create relatively clean inlet and exhaust conditions by placing the fan as far as practically possible from objects that would otherwise create unsteady flow.[3,4] However, this option is often severely limited by spatial constraints imposed by manufacturers' desire to make technology (such as personal computers) as compact as possible. In addition, the absence of a duct in most cooling fan applications prevents absorptive measures from being extremely effective, especially when the relatively low frequency nature of the tonal noise is considered. However, active noise control (ANC) is not subject to these same limitations, and becomes a potential candidate for the reduction of the tonal noise radiated from a cooling fan in a typical installation.

The use of ANC is not without its own limitations and difficulties, however, especially when considering how such a system would be applied to the reduction of cooling fan noise in office equipment. As in the case of passive control, size restrictions play a significant role. However, unlike passive noise reduction techniques, the spatial constraints which ANC is subjected to are mainly related to the space required for control actuators and sensors. The electronic components of the system may be contained on a small board mounted virtually anywhere within the equipment enclosure, but the placement of actuators and sensors requires careful consideration, especially when global control is desired. Practicality dictates that the actuators and sensors be placed either on or in the enclosure, so as to be integrated with the electronic device, yet in many research applications, placement of the error sensors in the far field of the sources is necessary to achieve consistent global control.[5]

A second major hindrance to the extensive application and commercialization of ANC is economics. The potential utility of active control has been successfully demonstrated in numerous research settings, but actual widespread commercial applications are rather sparse because the cost of actuators, sensors, and digital signal processing (DSP) technology is often too great for the product for this noise reduction method to be considered economical. This may possibly be said of a cooling fan application–the addition of ANC technology would increase the cost of cooling the electronic device and drive up its overall cost, which may or may not be worth the additional benefit provided. An active control system designed to reduce cooling fan tonal noise must be, therefore, both compact and economical, yet provide enough attenuation to be deemed worthwhile.

The research presented hereafter builds upon, in part, previous single channel efforts in the area. Quinlan's active control apparatus consisted of a loudspeaker placed adjacent to a cooling fan in a large baffle.[6] The error microphone was

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placed on-axis, equidistant from the centers of the two sources so as to maximize a dipole-type effect, and an optical tachometer was used as a reference sensor. Sound power reductions of 12 and 7 dB were reported for the fundamental and second harmonic of the BPF. Lauchle *et al.* also used a baffled fan and a similar reference sensor, but the control actuator consisted of the fan itself being driven axially by an electrodynamic shaker.[7] The error microphone used was also located roughly on-axis, out of the plane of the baffle. The results reported were similar to those of Quinlan, with reductions in sound power of 13 and 8 dB for the BPF and second harmonic, respectively. Wu's apparatus consisted of a rectangular plywood enclosure; with the fan and loudspeaker mounted on one face.[8] In addition to mounting the fan in a more realistic setting, effort was made to place the error microphone on the surface of the enclosure. Although no sound power data were given, global reductions for the BPF, but little attenuation for the second harmonic, were shown along one plane.

This investigation comprises both a description of a practical application of ANC to cooling fan noise and a study of the effect of control actuator configuration on the global reduction of multiple harmonics of the BPF. Specifically, the reference sensor and control actuators chosen for the research were selected on the basis of cost and widespread availability, not necessarily for their quality. In addition, the actuators and sensors were located close to the fan to make the configuration as compact as possible. These components were utilized in an adaptive multi-channel feedforward control system to investigate the effect of the control actuator arrangement on the average global attenuation possible.

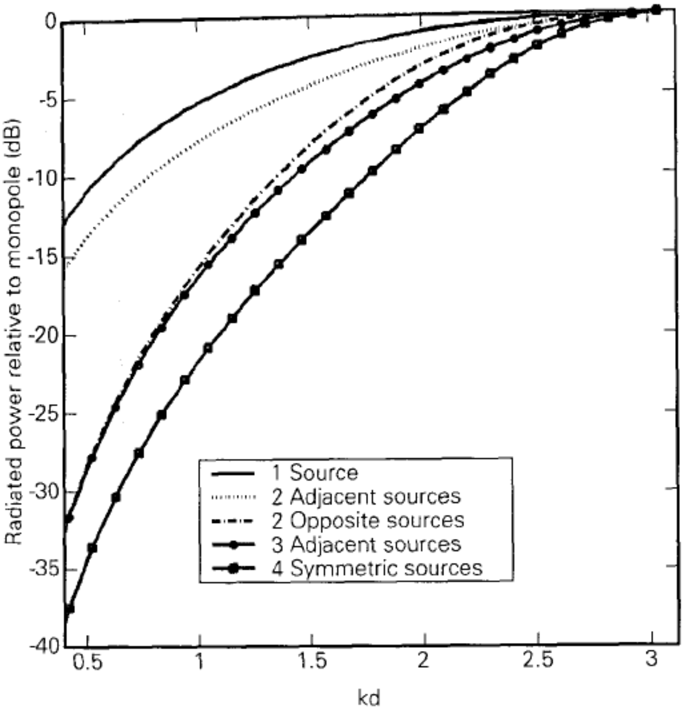


*Fig. 1– The effect of the addition of optimized control point sources on the radiated power of a primary point source for various configurations.*

## 2. SUMMARY OF CONTROL CONFIGURATION MODELING

There are two main challenges associated with the practical implementation of the active control of cooling fan noise, namely, the determination of (1) an appropriate number of and locations for control sources; and (2) near field error sensor locations which result in consistent global attenuations of multiple harmonics of the BPF. An investigation of the latter problem, that of near field error sensor locations, is treated in Refs. 9-11 and is not repeated here. While the theoretical determination of appropriate control source configurations is also described in Ref. 9, further development and discussion is merited because actuator arrangements not considered in that reference are examined below.

To maximize the reduction of the sound energy radiated to the far field under free-field conditions, the primary and control sources must be close enough, relative to a wavelength, to couple.[5] Physically, this coupling corresponds to the minimization of the real part of the radiation impedance, the radiation resistance, seen by the sources such that they are acoustically less efficient and less power is radiated to the far field. Appropriate control source configurations have been investigated by modeling both the control sources and the fan (the primary source) as baffled point monopole sources and then determining the optimal control source strengths which minimize the total radiated power, after the method developed by Nelson *et al.*[12] The radiated power relative to the primary monopole for various optimized control source configurations is displayed in Fig. 1, as a function of the non-dimensional $kd$, where $k$ is the acoustic wavenumber and $d$ is the source separation distance. Given four control sources placed at 90 degree increments around the primary source and at a distance $d$ from the primary source, the configurations represented are the various combinations of the four sources. For example, "two adjacent sources" may represent the sources at 90 and 180 degrees, while "two opposite sources" would mean the sources at 0 and 180 degrees or 90 and 270 degrees.

Because the fan and control actuators have been modeled as ideal point sources, the plot of the radiated power gives the maximum attenuation that may be achieved with these control source arrangements. While these levels of reduction are not expected with a real fan and non-ideal actuators, the levels of the various curves should provide some sense of relative performance expected from the various configurations over the range of $kd$ which is applicable, roughly 0.4 to 1.7. In general, more attenuation can be expected as channel count and source symmetry are increased. However, there is very little difference between the radiated power of the two opposite and three adjacent source configurations; the addition of the third source at most provides about 2 dB additional attenuation.

## 3. EXPERIMENTAL SETUP

### A. Research fan and enclosure

The fan selected for the investigation was an 80 mm (3.15 in), DC voltage Mechatronics fan with seven blades and four support struts located on the exhaust side of the fan. The BPF of the fan was maintained at 370 Hz throughout the research. The fan was mounted on an inside face of a 241 x 457 x 406 mm (9.5 x 18 x 16 in) aluminum enclosure and such that the air flowed out of the enclosure. A 25.4 mm (1 in) wide rectangular piece of aluminum was placed on the inlet of the fan to cause spatially unsteady flow into the fan and simulate a real-life obstruction. The A-weighted on-axis power spectrum of the fan, measured at a distance of 1.3 m, is displayed in Fig. 2. The global attenuation of the four harmonics of the BPF shown, whose levels are significantly above the broadband noise, is the desired end result of the investigation.

### B. Reference Sensor

An adaptive feedforward control system requires a reference signal that is correlated with the noise to be attenuated. Because the tonal noise from a fan consists of the BPF and harmonics, an appropriate non-acoustic reference sensor is one for which a pulse is generated at each passage of a fan blade. A non-acoustic reference sensor is advantageous because the need for compensation for the feedback from the acoustic radiation from the control actuators to the sensor in the control algorithm is eliminated, resulting in a more simply implemented adaptive controller. Previous investigations in this area employed the use of an optical tachometer with reflective tape on the fan blades to sense the passage of each blade and obtain a reference signal containing the BPF and a number of harmonics.[6,7] However, this would be both an unwieldy and relatively expensive sensor in a practical commercial application. For purposes of this research, a more suitable sensor has been chosen, that of an inexpensive infrared emitter-detector pair. When mounted on opposite sides of the fan blades and oriented axially, simple voltage divider circuitry permits the measurement of a pseudo-square waveform whose frequency is equal to the BPF.[11] Even with an inexpensive phototransistor detector with a rise time on the order of several microseconds and fairly noisy circuitry, several harmonics of the BPF are present with sufficient amplitude to be above the electronic noise. Therefore, as a reference sensor, the emitter-detector pair is compact and inexpensive, yet is suitable for feedforward control of multiple harmonics of the BPF. Because both a DC offset and more harmonics of the BPF than were desired to control were present, the reference signal was band pass filtered, with the cutoff frequencies set at 300 Hz and 1.5 kHz. The resulting power spectrum from the filtered signal is shown in Fig. 3. Finally, the signal from the phototransistor was amplified to a level convenient for analog-to-digital conversion.

### C. Actuator considerations

The control actuators selected for the experiment were Radio Shack 28 mm diameter mylar cone loudspeakers and were chosen based upon their small size and relatively low cost. The loudspeakers were mounted at 90 degree angles around the fan and the distance between the center of the fan and the center of each loudspeaker was 60 mm. This arrangement is shown in Fig. 4. PVC caps with an inside diameter of 31.8 mm (1.25 in) were mounted on the inside of the enclosure, encasing the back of each loudspeaker. This was done principally to prevent loudspeaker radiation inside the enclosure, which after reflecting off the inner surfaces and exiting through the hole for the fan exhaust, would be sensed by the error microphones and could potentially degrade controller performance. The average on-axis autospectral response of the four loudspeakers to a white noise input is shown in Fig. 5. The loudspeakers' resonant frequency occurs at approximately 800 Hz and by 370 Hz has rolled off almost 20 dB below the nominally flat response which exists above 1200 Hz.

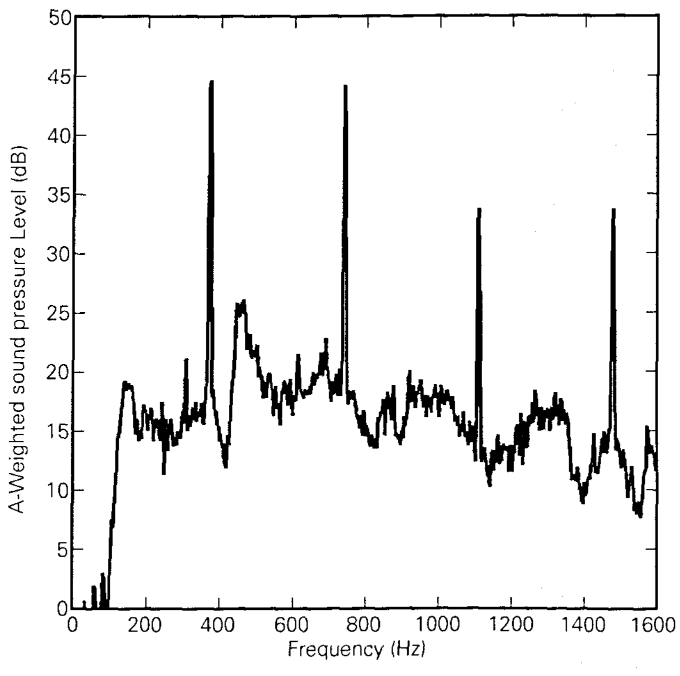


*Fig. 2– On-axis fan spectrum. The measurement was made at 1.3 m with the fan mounted in the enclosure and the flow obstruction in place.*

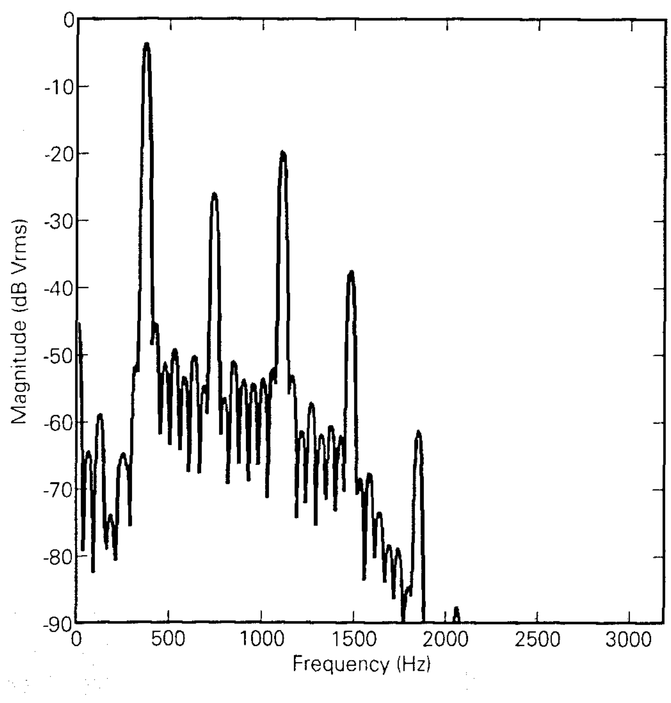


*Fig. 3– Band pass filtered and amplified reference signal.*

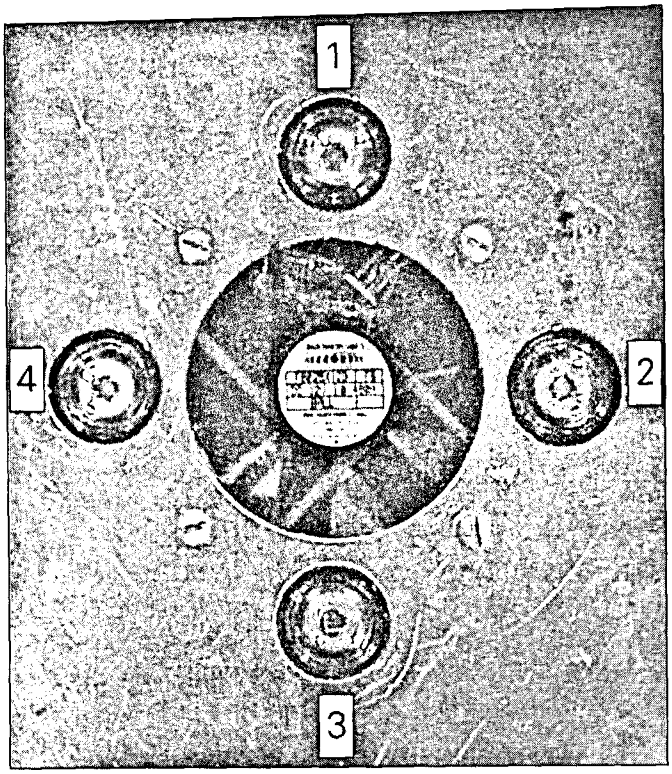


*Fig. 4– Photograph of the arrangement of the loudspeakers around the fan.*

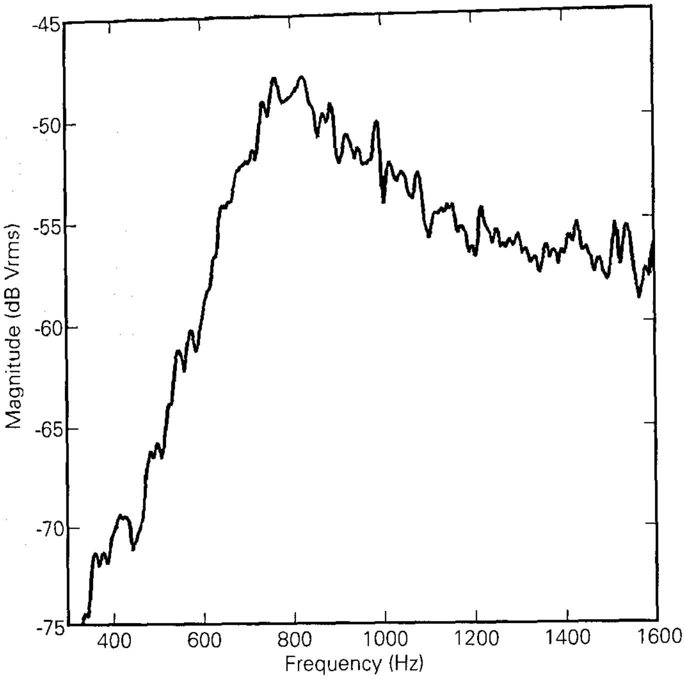


*Fig. 5 – Loudspeakers' average on-axis autospectral response to white noise input.*

## D. Error microphones

The error sensors used in this investigation consisted of Larson Davis Type-I microphones, varying from 6.35 mm and 12.7 mm condenser to 12.7 mm ICP microphones. While these microphones would never be used in a cost effective commercial application, because of the moderate frequency range of interest of small cooling fans (e.g. 200-2000 Hz), they could be replaced with inexpensive electret-type microphones. The microphones were located on the surface of the fan's enclosure, in various positions for each configuration, but always within 75-90 mm of the sources. To maintain hardware simplicity by avoiding the use of windscreens, the microphones were placed at a sufficient distance from the fan's exhaust to prevent mean flow from compromising the signal-to-noise ratio of the error signal. Fortunately, the required distance was less than 25 mm from the edge of the fan.

## E. Adaptive controller

The adaptive controller used to generate the control signals was a single reference, multiple output filtered-x LMS algorithm, which was implemented on a Spectrum 96000 floating point DSP board. The controller sampling frequency was typically set between 3.5 and 3.8 kHz for the various tests performed and 16-20 coefficients were used for the adaptive FIR control filters. Though the ability to use the passive online system identification scheme described in Ref. 13 was demonstrated as part of the investigation, fairly static propagation paths permitted the system identification to be performed offline. The resultant reduction in real-time algorithm computations was necessary for proper controller operation during four channel tests. Sixteen coefficients were typically used for the system identification filters.

As part of the digital-to-analog conversion process, the signals generated by the controller are passed through a single-pole low pass filter with a cutoff frequency of approximately 1.5 kHz. However, this nominal filtering is insufficient to prevent the generation of high frequency tones which are the result of aliasing the BPF and harmonics about the Nyquist frequency. An example of this phenomenon is shown in Fig. 6. The poor response of the loudspeakers at 370 Hz results in a control signal with an amplitude approximately 40 dB higher than the other harmonics, and the 6 dB per octave roll-off of the low pass filter cannot attenuate the significant aliased tone at about 3.1 kHz enough to prevent it from being quite audible without additional low pass filtering. Another unfortunate consequence of the loudspeakers' response at the BPF is that some error microphone locations result in control signals with sufficiently large amplitudes at the BPF to cause significant loudspeaker distortion. For such error microphone locations, the introduction of this nonlinearity into the control system often leads to controller instability.

## F. Global performance monitoring

Although ANC is often capable of significantly reducing the acoustical radiation at error sensors or in a particular direction, these attenuations do not necessarily result in a net global noise reduction. Because the determination of configurations which result in consistent global reduction of the BPF and harmonics was one of the purposes of the research, some means for monitoring that reduction was needed. Thirteen Larson Davis 12.7 mm Type-I ICP microphones were attached to a 1.52 m (5 ft) radius semicircular boom at equal angle

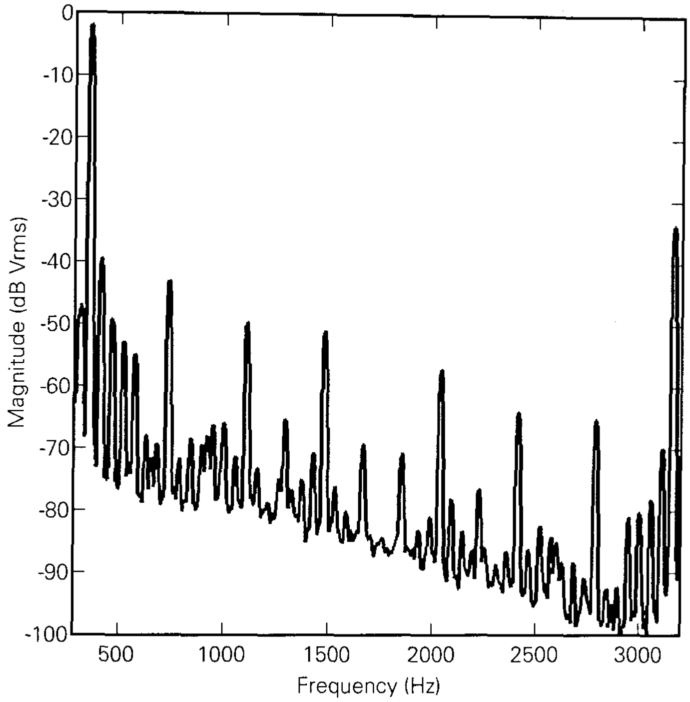


*Fig. 6– Example control signal power spectrum after digital-to-analog conversion and passage through a single pole low pass filter with cutoff frequency set at about 1.5 kHz.*

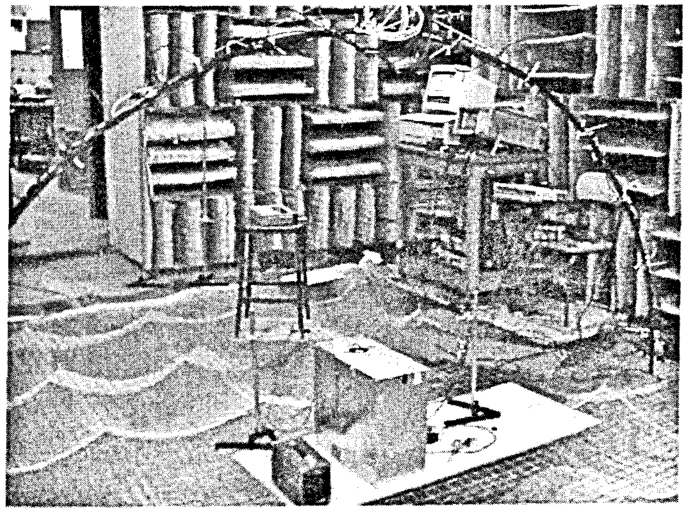

*Fig. 7– Semicircular measurement boom. Also seen are the fan enclosure and other test equipment.*

increments. The boom was then rotated in fifteen degree increments with an Oriental Motors stepper motor and power drive controlled by an automated LabVIEW-based program such that the pressure field was measured at a total of 145 locations around the enclosure. The measurement boom, fan enclosure, and other equipment can be seen in Fig. 7. Autospectral data from the microphones were acquired using a multi-channel Agilent dynamic signal analyzer and the global mean-square pressure (MSP) reductions were calculated and sound field plotted using Matlab.

## 4. RESULTS AND DISCUSSION

Results from five different control actuator configurations are shown in this section, beginning with a measurement of the fan's radiation at the four harmonics of the BPF without control. The loudspeakers are labeled numerically by beginning at loudspeaker number 1 (LS-1) directly above the fan and proceeding clockwise (see markings in Fig. 4). Using a right-handed coordinate system, the positive $x$ axis is toward LS-2 and the positive $y$ axis is toward LS-1, with positive $z$ representing the on-axis radiation of the fan above the enclosure. This is the orientation used for the axis labels in all graphical results in this section. The loudspeaker configurations tested are: one, two adjacent (e.g. LS-2 and LS-3), two opposite (e.g. LS-1 and LS-3), three, and four speakers. A representative example of the tonal reductions is displayed for each configuration and is followed by a comparison of the average performance of the various configurations. For all trials shown, the number of error microphones and loudspeakers were equal, and each loudspeaker was driven by a separate control channel. For this reason, the various configurations are referred to as 1 x 1, 2 x 2 (adj.), 2 x 2 (opp.), and so forth.

### A. Fan directivity

The radiation at the BPF (370 Hz) and subsequent three harmonics from the enclosed fan with obstruction is shown in Fig. 8, as measured by the boom microphones. Both the radius and shading of the surface represent the calibrated sound pressure level (SPL) at a particular angle. The mesh hemisphere which overlays the shaded surface plot is intended to help visualize the degree to which the radiation is omnidirectional. The offset of the location of the fan in the positive $z$ direction due to the height of the enclosure causes the radiation to be skewed somewhat toward the $z$ axis in the plots. As can be seen from the plots, the fan's radiation becomes less smooth and more directional as frequency increases.

### B. 1 x 1 Control

A single loudspeaker (LS-3) and error microphone represents a similar active control configuration as that of the previous single channel investigations.[6-8] The fan's radiation with ANC engaged is displayed in Fig. 9, with the mesh surface in this case indicating SPL without control. The MSP reduction for each of the four harmonics of the BPF was 6.5, 8.3, 5.3, and 2.0 dB, respectively. The attenuation for 370 Hz is greatest on-axis, which seems to be consistent with most single channel tests performed. Comparison with the more global attenuation achieved for 740 Hz suggests an inability of the loudspeaker to significantly affect the overall radiation resistance seen by the fan at the BPF. Attenuation for the third harmonic, significant toward the negative $y$ axis where the error microphone was located, was less away from the error microphone. Finally, consistent with most single channel tests, global reduction of the fourth harmonic, 1480 Hz, was rather insignificant.

### C. 2 x 2 (adj.) Control

For the trial discussed here, LS-2 and LS-3 are the control system actuators used. The active control results for the

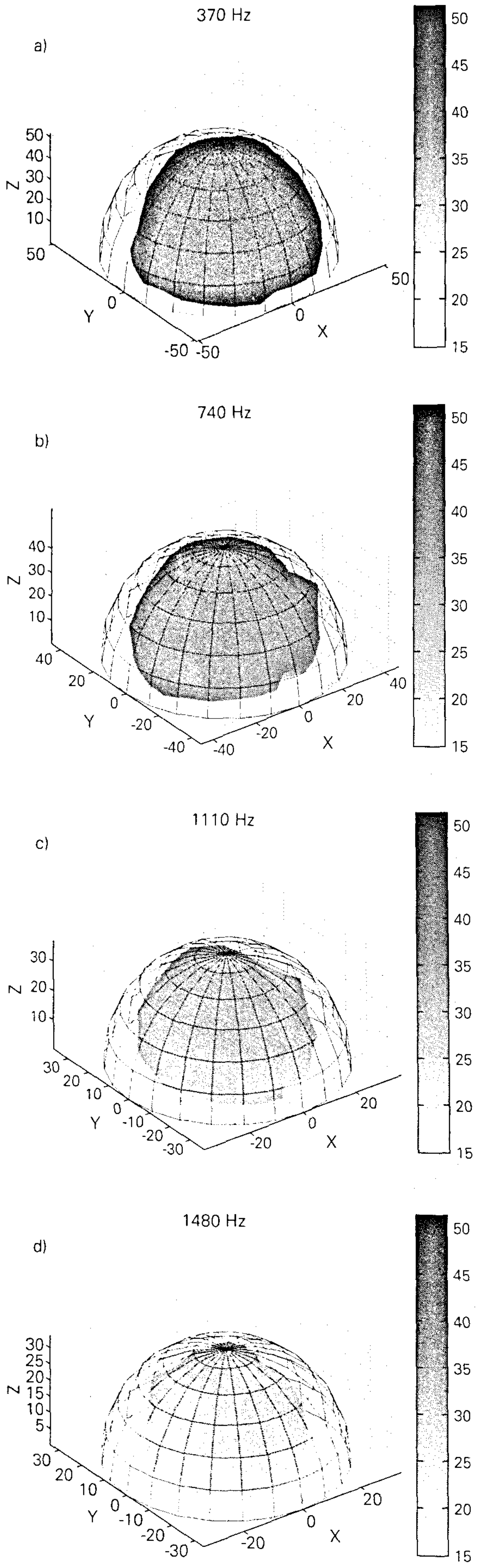


*Fig. 8– Fan directivity measurement without control made at a) 370 Hz, b) 740 Hz, c) 1110 Hz, and d) 1480 Hz.*

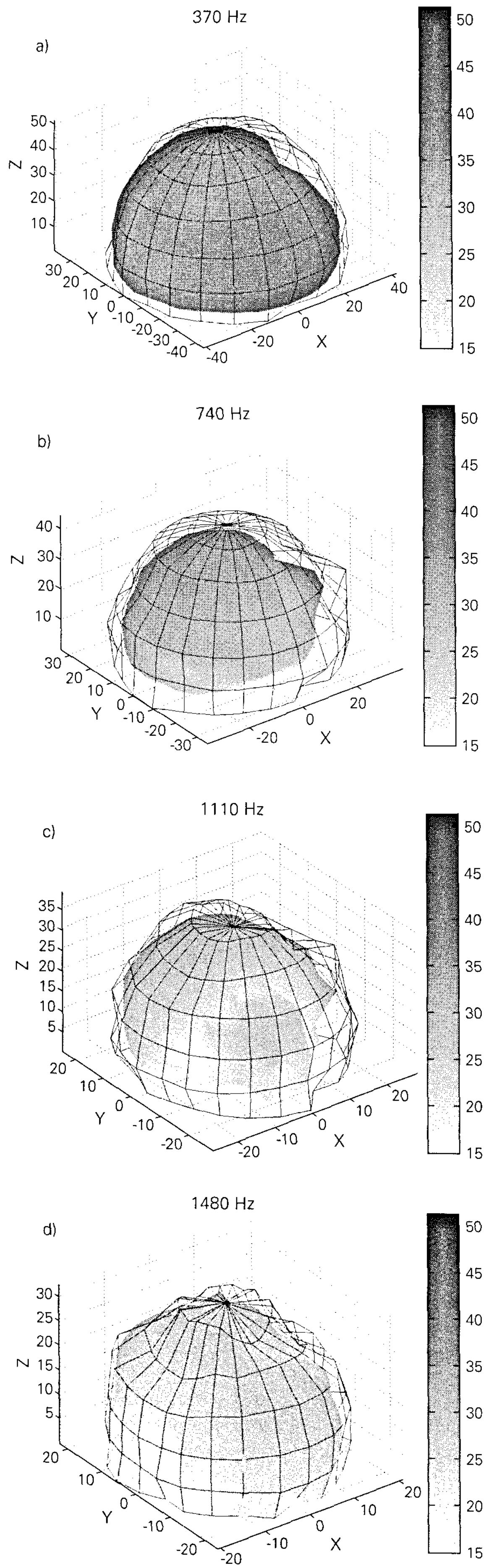


*Fig. 9– Example ANC results for single channel control.*

four tonal frequencies, shown in Fig. 10, correspond to MSP reductions of 12.1, 14.8, 0.2, and 4.2 dB. This particular trial was chosen for discussion because of the characteristics of the control at the third harmonic. The attenuation is both more global and greater at the other three harmonics than for the single loudspeaker with the addition of the second control channel. However, the behavior at 1110 Hz is very different; while the control system is successful in decreasing the tonal radiation along one axis, the radiation significantly increases along the other, resulting in virtually no net attenuation. There was a small number of trials for single and two channel configurations, both with adjacent and opposite loudspeakers, for which this sort of mismatched coupling behavior was observed, usually for the third or fourth harmonic.

## D. 2 x 2 (opp.) Control

The loudspeakers used in the particular test shown in this section are LS-2 and LS-4. The fan's radiation with the control system active, for which the achieved tonal MSP reductions were 7.1, 13.5, 5.6, and 5.4 dB, is displayed in Fig. 11. These results show a significant increase in the radiation of the BPF towards the positive $y$ axis, where the SPL of the fan without control was less. The reduction for the second harmonic is the most global of the four; the attenuation for the third harmonic is much greater towards the positive $y$ axis. Other 2 x 2 trials using opposite loudspeakers yielded different results; for example one particular configuration resulted in an 11.0 dB MSP attenuation for the fundamental, but a 0.5 dB increase in the fourth harmonic. A different arrangement resulted in only a 5.8 dB decrease in the MSP at the BPF, but 16.4 and 11.9 dB reductions for the second and third harmonics, respectively.

## E. 3 x 3 Control

It is with three channel control that consistent, significant reduction of the third and fourth harmonics of the BPF is first achieved. For example, the reductions in MSP for the trial shown in Fig. 12 were 10.5, 16.5, 10.6, and 9.6 dB for the first four harmonics of the BPF. The control actuators used were LS-1, LS-2, and LS-3. Although the directivity of the radiation at the BPF with active control is quite similar to the previous configurations, with the greatest attenuation on-axis, the reductions at the other three harmonics are quite global in nature and begin to approach the broadband noise levels, which may explain the directivities' lack of smoothness.

## F. 4 x 4 Control

The arrangement with all four loudspeakers used and four error microphones located on the surface of the enclosure and in the near field of the sources represents the most complex control configuration used in this research. For the test displayed in Fig. 13, reductions of 9.4, 17.7, 16.1, and 10.4 dB were achieved for the four targeted harmonics of the BPF. Again, the BPF's radiation with control is similar to that of the other configurations. Also, the other three harmonics' reductions are quite global like the three channel example shown previously, with the third harmonic being attenuated almost completely to broadband levels.

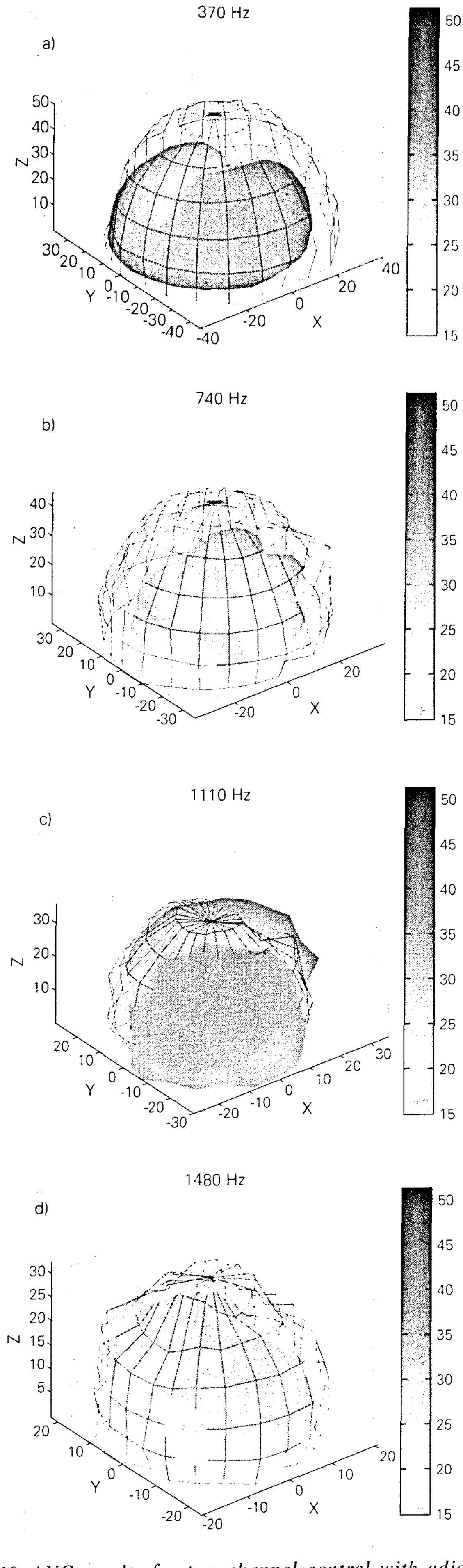


*Fig. 10–ANC results for two channel control with adjacent loudspeakers.*

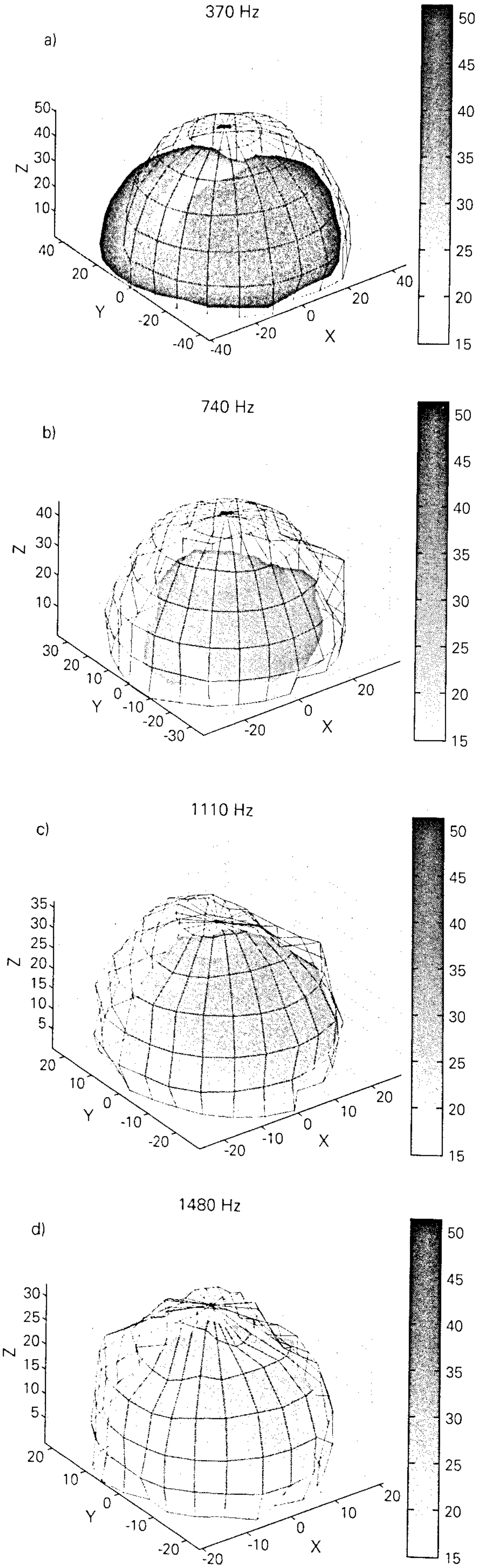


*Fig. 11–ANC results for two channel control with opposite loudspeakers.*

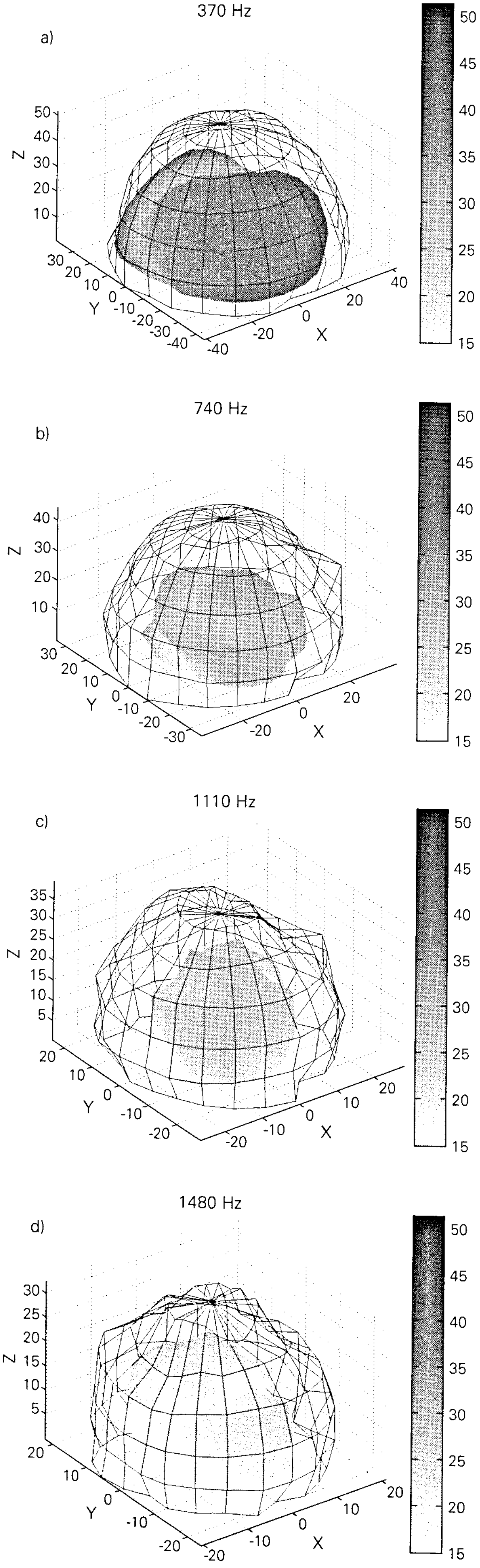


*Fig. 12–ANC results for three channel control.*

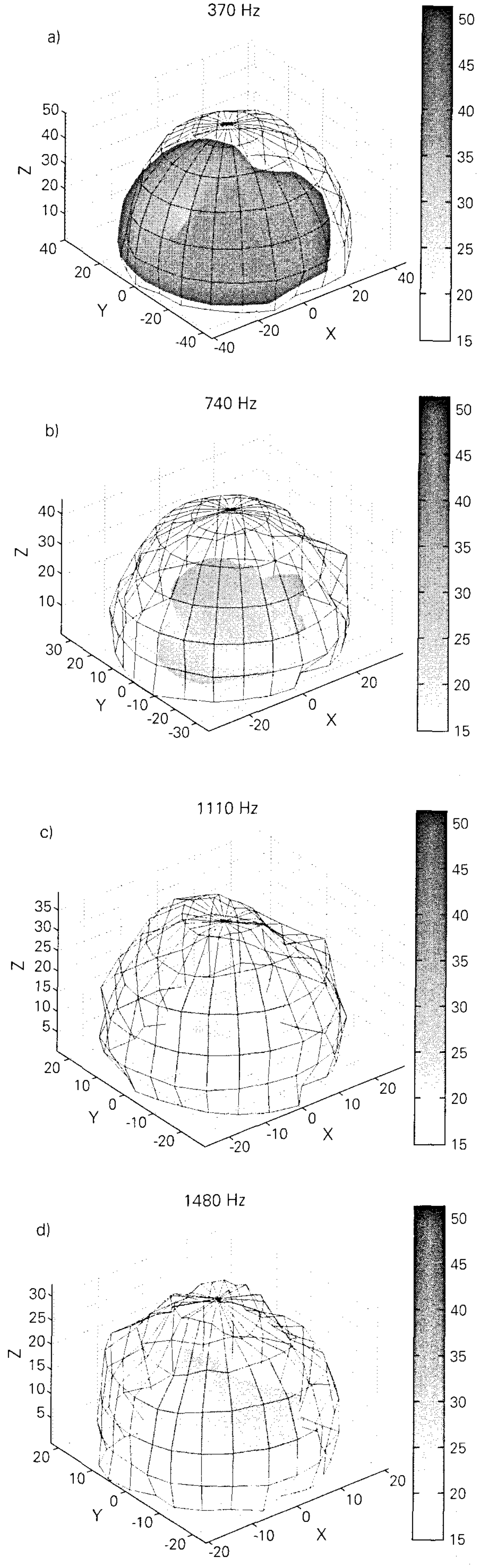


*Fig. 13– ANC results for four channel control.*

## G. Comparison between actuator configurations

It would be difficult to compare the overall performance of the various control arrangements without considering the results from several trials for each configuration and without a metric for the overall reductions achieved. For that reason, the eight trials for each actuator configuration which yielded the greatest A-weighted overall tonal reduction for the first four harmonics of the BPF have been selected. While this may or may not accurately represent the actual perceived performance of the system, it is used to provide some quantitative objectivity to what could otherwise be a highly subjective descriptor. From these eight trials, the average reductions in MSP and associated standard deviations for each of the four frequencies have been calculated for the various control configurations, as well as for the overall A-weighted tonal reductions. These results are tabulated in Table 1. The first number in each cell represents the mean reduction in MSP in dB and the second number is the standard deviation, also in dB.

The mean reduction in MSP at the BPF (370 Hz) is actually greatest for the two channel system with adjacent loudspeakers; the reason for this is not clearly understood. According to the theoretical predictions in Fig. 1, the four channel system should have yielded greater reductions at the BPF than the other configurations, and especially a much larger attenuation than this two channel configuration. However, it appears that the poor response of the loudspeakers at the BPF limits the mean attenuation possible for any of the multi-channel configurations to around 10-12 dB. This is not only supported by the numerical results; the graphical results from the trials shown are representative of the shape of the directivity at the BPF for the majority of tests performed. The greatest attenuation is achieved roughly on axis, with lesser attenuation at large angles. Apparently, whether two, three, or four loudspeakers are used, the radiation resistance seen by the fan at the BPF is affected similarly, resulting in like directivities and MSP reductions.

Analysis of the mean attenuations at 740 Hz is more straightforward. There is a natural progression in attenuation as channel count or symmetry increases until 4 x 4 control is reached. The similar attenuations for three and four channel control suggest that perhaps the practical limit in reduction possible has been reached, at least for the control system considered. The maximum reduction in MSP at 740 Hz achieved for any of the configurations, 18.5 dB, was for a four channel arrangement; however mean performance shows little difference between the three and four channel systems.

A similar trend applies to the third and fourth harmonics; however, it should be noted that the performance of the two channel system with opposite loudspeakers is little better, and in the case of the fourth harmonic, worse than the system with adjacent loudspeakers. The theoretical power reduction plotted previously for the various configurations (see Fig. 1) suggests that using opposite loudspeakers should be a significant improvement over adjacent speakers. In fact, the reductions should approach those of the three channel system, which they obviously do not, especially for the case of the third and fourth harmonics. The reason for the two channel system with opposite loudspeakers' failure to perform better can only be speculated at, but because both loudspeaker combinations

TABLE 1—Mean-square pressure reduction (MPR) and standard deviation ($\sigma$), in dB, averaged over the eight best trials for each control actuator configuration. Results for the first four harmonics of the BPF and the overall A-weighted tonal levels are shown.

| | 370 Hz | | 740 Hz | | 110 Hz | | 1480 Hz | | A-weighted Total | |
|---|---|---|---|---|---|---|---|---|---|---|
| | MPR (dB) | $\sigma$ (dB) | MPR (dB) | $\sigma$ (dB) | MPR (dB) | $\sigma$ (dB) | MPR (dB) | $\sigma$ (dB) | MPR (dB) | $\sigma$ (dB) |
| 1 x 1 | 7.2 | 0.9 | 8.3 | 2.3 | 4.3 | 1.7 | 2.4 | 1.7 | 6.9 | 1.2 |
| 2 x 2 adj. | 10.3 | 1.5 | 12.0 | 2.6 | 5.9 | 2.7 | 5.1 | 2.3 | 9.7 | 1.2 |
| 2 x 2 opp. | 8.1 | 2.0 | 13.6 | 2.0 | 6.3 | 3.5 | 3.6 | 2.8 | 8.6 | 0.8 |
| 3 x 3 | 9.5 | 1.8 | 15.7 | 2.5 | 11.8 | 3.2 | 8.9 | 4.5 | 10.9 | 1.9 |
| 4 x 4 | 10.1 | 1.0 | 15.3 | 2.8 | 12.8 | 2.4 | 8.7 | 3.5 | 11.4 | 0.4 |

(LS-1/LS-3 and LS-2/LS-4) and multiple error microphone locations were tried with similar results, it likely has to do with the number of error microphones used. The analysis performed in Ref. 9 for two opposite point sources shows that there are four theoretically ideal regions in which to locate error microphones; two microphones may therefore be insufficient to lead to the generation of control signals which enable the loudspeakers to couple effectively with the fan and reduce the radiation resistance which acts on it. Future investigations may involve the use of a mismatched transducer count (e.g. two symmetric loudspeakers and three or four microphones) for this actuator configuration in an attempt to increase global pressure reductions.

The standard deviation calculations for the reductions of the fan's tones by the various actuator configurations suggest that near field placement of the error microphones leads to substantial variation in the control achieved, especially for the higher harmonics. In other words, attenuation for these harmonics with a particular control system is fairly sensitive to the location of error microphones. However, the overall A-weighted tonal reductions and standard deviations suggest that these inconsistencies for the attenuation of individual harmonics appear to average themselves out, resulting in a fairly stable overall mean performance, especially for the four channel system.

## 5. CONCLUSIONS

The research presented constitutes a discussion of the practical application of multi-channel ANC to the reduction of tonal noise from a small axial cooling fan. The use of multiple loudspeakers placed in close proximity to the fan greatly increases the global control capability of the higher frequency tonal noise. In addition, the near field placement of the error microphones and the use of an inexpensive infrared emitter-detector pair as a reference sensor yields a system transducer configuration that is both compact and practical. This research also suggests that small, inexpensive loudspeakers may be employed in such an application, but consideration will have to be given to, and possibly a compromise made between, the properties of the loudspeakers, desired performance, and cost effectiveness of the control system.

The similarity in performance between the three and four channel systems described, despite the expected disparity based on theoretical predictions, suggests that for at least this application, there is little benefit gained from the additional control channel. This assertion is qualified by the statement that the four channel system's overall performance appears to be slightly more consistent; perhaps this is due to the increased symmetry afforded by the four loudspeakers. For that reason, future applied research may involve the study of three symmetrically placed loudspeakers, for which theoretical predictions suggest system performance should be consistent with that of the four channel setup.[9,11] An additional future investigation involves the use of loudspeakers with better responses at the fan BPF, so as to better understand the configurations' experimental possibilities for global attenuations at low frequencies.

## 6. REFERENCES


1. W. Chiu, G. C. Lauchle, and D. E. Thompson, "Subsonic axial flow fan noise and unsteady rotor forces," J. Acoust. Soc. Am. **85**, 641-647 (1989).
2. K. D. Kryter and K. S. Pearsons, "Judged noisiness of a band of random noise containing an audible pure tone," J. Acoust. Soc. Am. **38**, 106-112 (1965).
3. K. B. Washburn and G. C. Lauchle, "Inlet flow conditions and tonal sound radiation from a subsonic fan," Noise Control Eng. J. **31**, 101-110 (1988).
4. J. M. Fitzgerald and G. C. Lauchle, "Reduction of discrete frequency noise in small, subsonic axial-flow fans," J. Acoust. Soc. Am **76**, 158-166 (1984).
5. Colin H. Hansen and Scott D. Snyder, "Control of free field sound radiation," Chap. 8 in *Active Control of Noise and Vibration*, (E & FN SPON, London, 1997).
6. D. A. Quinlan, "Application of active control to axial flow fans," Noise Control Eng. J. **101**, 95-101 (1992).
7. G. C. Lauchle, J. R. MacGillivray, and D. C. Swanson, "Active control of axial-flow fan noise," J. Acoust. Soc. Am. **101**, 341-349 (1997).
8. M. Q. Wu, "Active cancellation of small cooling fan noise from office equipment," *Proc. INTER-NOISE 95*, edited by Robert J. Bernhard and J. Stuart Bolton (Noise Control Foundation, Poughkeepsie, New York, 1995), pp. 525-528.
9. K. L. Gee and S. D. Sommerfeldt, "Application of theoretical modeling to multi-channel active control of cooling fan noise," J. Acoust. Soc. Am. **115**, 228-236 (2004).
10. K. L. Gee and S. D. Sommerfeldt, "Multi-channel active control of axial cooling fan noise," *Proc. INTER-NOISE 2002*, edited by A. Selamet, R. Singh, and G. C. Maling (Noise Control Foundation, Poughkeepsie, New York, 2002), Paper N616.
11. K. L. Gee, "Multi-channel active control of axial cooling fan noise," MS Thesis, Brigham Young University, Provo Utah, 2002.
12. P. A. Nelson, A. R. D. Curtis, S. J. Elliott, and A. J. Bullmore, "The minimum power output of free field point sources and the active control of sound," J. Sound Vib. **116**, 397-414 (1987).
13. S. D. Sommerfeldt, "Multi-channel adaptive control of structural vibration," Noise Control Eng. J. **37**, 77-89 (1991).