Active control of axial-flow fan noise

Gerald C. Lauchle, John R. MacGillivray, and David C. Swanson
Penn State University, Graduate Program in Acoustics and Applied Research Laboratory, P.O. Box 30, State College, Pennsylvania 16804

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Discrete-frequency axial-flow fan noise reduction using active noise control is described. The unique aspect of the current research is the use of the fan itself as the antinoise source in the active noise control scheme. This is achieved by driving the entire fan unit axially with an electrodynamic shaker which mechanically couples the solid surfaces of the fan to the acoustic medium. The fan unit is thus transformed into a crude loudspeaker. A near-field microphone serves as an error sensor, where transfer function measurements between the electrical input to the shaker and the electrical output of the microphone are found to be reasonably free of phase distortions and linear. A feedforward algorithm utilizing the output of a tachometer as a reference signal is used. The experimental apparatus is composed of a baffled fan unit in a free field. A small cylindrical flow obstruction is placed on the inlet side of the fan to enhance noise emissions at the blade-pass frequency and harmonics. The experiment successfully demonstrates the concept of active control of tonal fan noise using a shaken fan as the cancellation source. For the fan operating in a planar baffle, the fundamental blade-passage frequency sound-pressure level at the location of the error sensor is reduced by 20 dB, while the second and third harmonic levels are reduced by 15 and 8 dB, respectively. Placing a cabinet enclosure over the baffled fan did not affect these results significantly, and free-field sound power measurements indicate similar level reductions with the active control in operation. [S0001-4966(97)00112-4]

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INTRODUCTION

Under the long-wavelength (relative to rotor diameter) constraints of compactness, a subsonic fan, either baffled or unbaffled, radiates acoustic energy because of both steady and unsteady aerodynamic forces generated by the rotating blades. For typical fans operating in air, the blades are usually assumed rigid. The steady forces on fan blades generate the thrust of the fan, but because they are rotating, they also create a steady-loading noise component. The mechanism for steady-loading noise (Gutin noise) is the time-dependent distance between the individual steady forces on the blades and the observer. This component radiates sound at the blade-passing frequency (BPF) given by the number of fan blades multiplied by the shaft speed. The predominant direction of radiation is 90° to the fan axis, where the time variation in distance between any given blade and a fixed observation point is maximum. This type of noise begins to dominate other fan noise mechanisms only under the conditions of very high blade tip speeds (very high subsonic and supersonic conditions) and clean inflow/outflow conditions.

Unsteady blade forces result when the blades pass through spatially nonuniform, time invariant flow fields. Such situations occur when the fan is operated close to obstructions that can disrupt an otherwise uniform inflow. As the blades pass through these regions, the magnitude and direction of the local velocity incident to the blade sections varies with circumferential position. This gives rise to a local blade section lift and drag force that varies periodically with time. Dipole sound is produced by these fluctuating forces at harmonics of the BPF. Peak sound pressure occurs along the axis of the fan for the lower harmonics. In addition to the spatially nonuniform, time invariant flows, random flow variations due to turbulence and unsteady upstream conditions may also be present. These stochastic flow variations cause random blade forces which lead to a broadband component of sound radiation. The level of sound at the harmonics of the BPF, however, are usually many dBs above the broadband components. The BPF tones are therefore important to suppress (initially) in a fan noise reduction program.

There has been considerable interest and fundamental research on the use of active noise control (ANC) to reduce the level of discrete-frequency noise radiated from fans, blowers, and turbomachines. One of the first demonstrations of this technology was by Ffowcs Williams[1] on a British Gas Corp. gas turbine. Koopmann *et al.*[2] and Neise and Koopmann[3,4] were able to actively control the tonal emissions from a centrifugal blower operating in a duct, while Mendat *et al.*[5] achieved active attenuation of the random noise components as well. Felli *et al.*[6] demonstrated active blower noise control in a duct while using the reciprocal characteristics of a loudspeaker to permit replacement of the more conventional microphone error sensor by a loudspeaker. Ducted propeller or blower ANC is relatively straightforward to accomplish for the plane-wave propagating pressure components due to their one-dimensional nature. To actively cancel higher-order duct modes requires an array of synchronously phased antinoise sources in the duct.[3,4] Sutliff and Nagel[7] have also made progress at doing this for a ducted propeller through use of a feedforward ANC algorithm that uses a rotor blade position sensor as a reference.

Studies of active noise control for fans situated in the

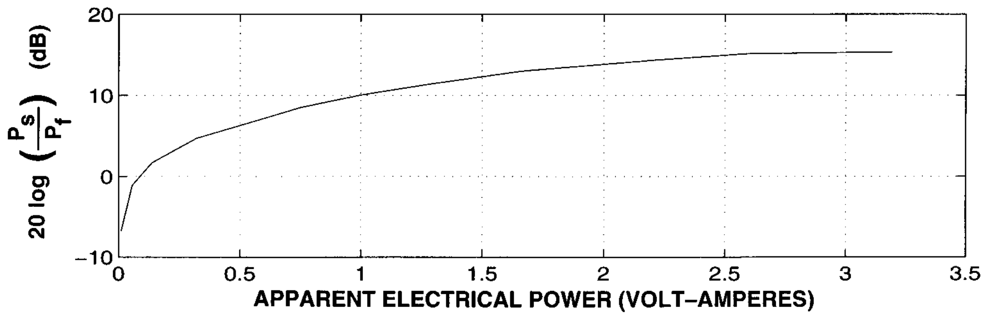

FIG. 1. On-axis far-field sound-pressure level from the shaken fan unit relative to the BPF tonal sound-pressure level radiated by the same fan in free-delivery operation. The independent variable is the apparent electrical power supplied to the mechanical shaker.

open field or in baffles are not numerous. Quinlan[8] was successful in reducing the blade-passage tones of an axial-flow cooling fan mounted in a planar rigid baffle. Located next to the fan in the baffle was the antinoise loudspeaker. The baffle apparently tends to bring the fan directivity into axial symmetry and makes it more uniform. With the antinoise source and baffled (compact) fan noise source having similar directivities, global far-field noise reduction of 10 dB was achieved for the first two harmonics of the blade-passage frequency components.

The goal of the research presented in this paper is to investigate the use of the fan itself as an antinoise source in the active control of the tonal emissions from an acoustically compact, baffled, axial-flow fan. Chiu *et al.*[9] used coherence function measurements between a small, fan-mounted force sensor and a far-field microphone to show that the radiation at the first several harmonics of the BPF was due entirely to the total integrated (over the plane of the rotor) unsteady rotor force. This result suggests that the appropriate antinoise actuator in ANC schemes for compact fans should be a mechanical shaker, or similar device that can generate controlled unsteady forces on the primary source. Furthermore, a shaken fan secondary source, if acoustically efficient, would be collocated with the primary aerodynamic fan noise sources which would be a significant advantage in global noise control where directivity issues are of concern. In the experiments described below, we use a small, commercially available electronic cooling fan as a the primary fan noise source. It is mounted directly to a electrodynamic shaker and the entire assembly is mounted in a rigid planar baffle. A feedforward ANC algorithm is used to control the shaker so that global cancellation of the far-field tonal fan noise emissions is achieved. A tachometer provides the reference signal for the algorithm, which consists of a pulse for each blade passage, and a microphone is used to supply the error signal. A feasibility study is also described that provides information on the efficiency of a shaken fan as an acoustic radiator.

I. FEASIBILITY OF USING SHAKEN FAN AS ANTINOISE SOURCE

A basic issue that needs to be addressed in the determination of whether or not a particular fan unit can be successfully implemented in the proposed ANC scheme is whether the shaken fan unit produces substantial acoustic radiation for a reasonable power input to the shaker. Another issue to be addressed is the fan noise directivity. It is desirable to have the directivity patterns of the primary and secondary sources identical. These issues are addressed experimentally for a given fan unit.

A. Shaken fan feasibility tests

Experiments involving shaker-induced radiation from the fan unit were conducted in the flow-through anechoic chamber[10] located at the Applied Research Laboratory of Penn State University. A Nidec, 82-mm-diam plastic fan was fitted with an aluminum disk on the back of its frame, which allowed for connection to a Wilcoxon, Type F3 electromagnetic shaker via a stinger which was fabricated from stainless-steel rod. The stinger was aligned along the axis of fan rotation. The shaker/fan assembly was mounted to a rigid stand such that the fan axis was vertical. Tests were conducted with the stand isolated above the anechoic wedges; the fan was unbaffled. The effect of the fan flow field on the radiation from the shaken fan unit was investigated by simply operating the fan simultaneously with the shaker.

A B&K Type 4136 microphone (1/4-in.) was suspended 1 m above the fan blades on the inlet side; its signal was analyzed on an HP 35 665A spectrum analyzer. In order to calculate the apparent electrical power[11] consumed by the shaker, a voltmeter was placed in parallel and an ammeter in series with the electrical connections to the shaker. For certain tests the fan was swiveled 180° to reverse the direction of airflow.

The fan was shaken at 264 Hz which is its free-delivery fundamental BPF. Figure 1 shows the measured on-axis acoustic pressure (p_s) at 1 m as a function of the apparent electrical power supplied to the shaker. Here, the dB level is referenced to the on-axis acoustic pressure (p_f, also at 1 m) which is produced by the normal free-delivery operation of the fan at the BPF (while the shaker was off). It is seen that the shaken fan radiation matches the pressure amplitude generated by the aerodynamic fan noise mechanisms when approximately 0.1 W of apparent power is supplied to the shaker. Figure 1 also demonstrates that with a sufficient supply of electrical power, the acoustic pressure response from the shaken Nidec fan can produce much greater acoustic pressures than the fan in operation. These results suggest that the shaken fan can act as a crude loudspeaker and is hence capable of serving as the antinoise source in ANC applications.

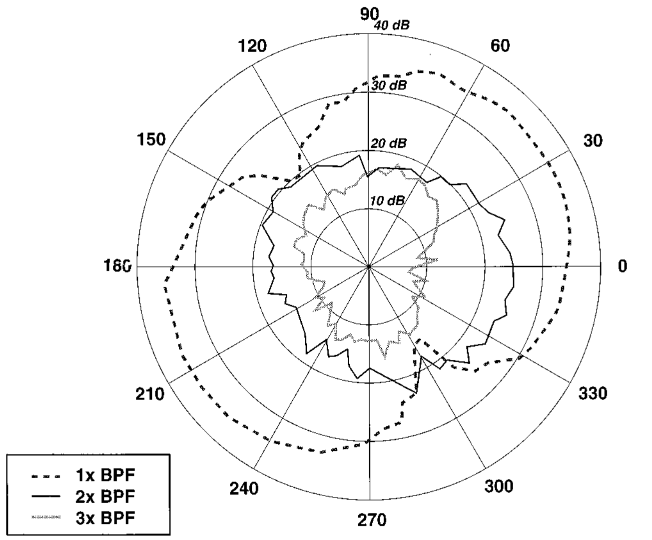

FIG. 2. Far-field directivity patterns for the unbaffled fan radiation at the first three harmonics of the BPF. The 90° axis corresponds to the fan axis on the inlet side.

Additional data of the type shown in Fig. 1 have been obtained for off-axis positions of the microphone, the fan, and shaker in operation at the same time, and again with the flow direction reversed.[12,13] In all of these situations the shaken fan was found to create sufficient acoustic energy to warrant its use as an antinoise source. However, it was observed that when the flow from the fan was reversed and directed toward the microphone (which was 1-m away and uninfluenced by the flow), the shaken fan acoustic pressure amplitude was about 2 dB less than the case when the flow was away from the microphone. This is explained as a change in radiation impedance due to flow. An analysis by Muehleisen[14] has predicted this level of change for the typical mean flow velocity of this fan.

B. Fan directivity characteristics

Unbaffled axial flow fan units are reported to produce skewed directivity patterns.[8] A nonsymmetric fan directivity pattern would suggest that complicated ways of shaking the fan would be necessary in order to most effectively cancel the fan noise. Radiation patterns for the type of fan units considered here are measured under unbaffled and baffled conditions in order to obtain knowledge about the source type and to determine the effect of the baffle on the radiation pattern. All directivity patterns were measured at 1 m from the fan in the free field. A movable microphone boom provided measurements in increments of 5°. The unbaffled directivity patterns for the first three harmonics of the BPF of the Nidec fan are shown in Fig. 2. In order to obtain results which were repeatable within ±1.5 dB, each data point required 300 spectral averages over a frequency bandwidth of 1.6 kHz. The sampling rate was 4096 Hz. It is clear from the results shown on this figure that the unbaffled fan has complicated directivity characteristics. The pattern for the first harmonic could be interpreted as dipolelike, but it is skewed off-axis by some 60°. The patterns for the second and third

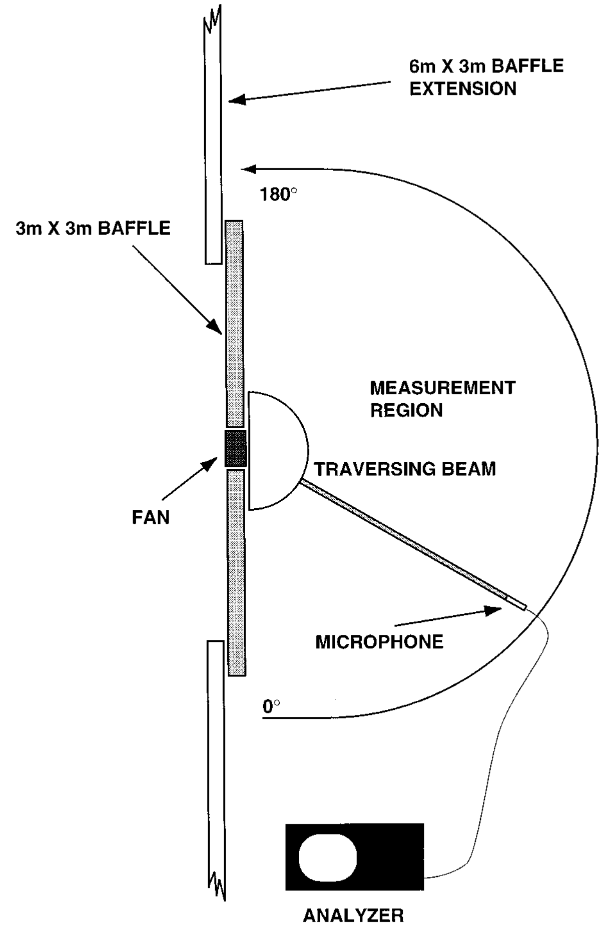

FIG. 3. Sketch of the experimental setup for measuring baffled fan directivity patterns.

harmonics cannot be interpreted in terms of simple dipole radiation patterns. These results are very consistent with those reported by Quinlan[8] for a different but similar fan. The reason for the skewness is not known precisely, but it may be a result of asymmetries in the fan construction that result in peak aerodynamic forces being directed off-axis.

When the acoustically compact fan unit is shaken as described above, one would expect dipole directivity of the sound pressure with an on-axis peak. Placement of the fan in a baffle, such as that depicted schematically in Fig. 3, would transform the dipole source directivities into monopole directivities. Figure 4 shows the baffled fan directivity patterns measured on the inlet side of the fan. The patterns are clearly more uniform than in the unbaffled case, which is favorable from the ANC viewpoint. The directivity patterns on the outlet side of the fan were measured also[13] and found to be nearly identical to those shown in Fig. 4. These experimental results confirm the findings of Quinlan.[8] They indicate that baffled fan ANC should be simpler to achieve than unbaffled fan ANC when the secondary source is of the dipole type (loudspeaker or shaken fan unit) and located in the planar baffle containing the fan.

II. EXPERIMENTAL ACTIVE FAN NOISE CONTROL

An experiment is constructed to demonstrate reduction in tonal noise from an axial-flow fan in which the fan is

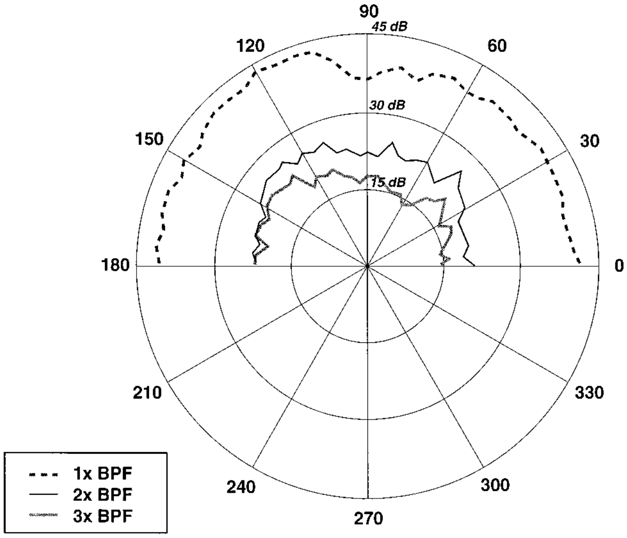

FIG. 4. Far-field directivity patterns for the baffled fan radiation from the inlet side at the first three harmonics of the BPF.

shaken to produce antisound. A Filtered-X control algorithm[15,16] is implemented and responds only to the tonal components in the fan spectrum for this experiment. The Filtered-X algorithm is a feedforward algorithm that uses an independent reference signal related to the tones of the primary source. Because the tones of interest are the harmonics of the BPF, a simple optical-type tachometer is used for the reference.

A. Experimental setup

Elements of the Filtered-X active noise control experimental setup are shown schematically in Fig. 5. In this system, the primary noise is the noise generated by the fan in operation and the secondary noise is the acoustic signal produced by the shaken fan. The error signal is the electrical output of a microphone placed on the inlet side of the fan unit, and an independent reference signal is provided by the optical tachometer. The optical sensor is placed above the fan and is sensitive to the passing of reflective strips located on the leading edge of each blade. The signal from the sensor is coherent at the BPF in addition to the higher harmonics because the voltage pulse is rectangular in shape; the Fourier transform of these periodic pulses produces a harmonic train. The optical sensor is insensitive to the motion of the axially shaken fan because the reflective strips, which the optical sensor monitors, pass orthogonally to the axis of the fan. An independent reference signal for the controller is therefore produced by the optical sensor.

The height of the error microphone above the fan unit was typically a fan diameter or more and was often moved to verify noise cancellation observations. The seven-bladed fan was mounted in a plywood baffle such that its inlet side was flush to the surface of the baffle. The gap between the fan housing and the baffle was 5 mm such that the housing was not in physical contact with the baffle. The fan was supported completely by the Wilcoxon shaker which was rigidly mounted to a platform located underneath the baffle. A small cylindrical rod was placed across the center of the fan at an axial distance of approximately $0.1R$ from the fan hub, where R is the fan blade tip radius. This distance is close enough to cause an increase in the tonal fan noise components due to the wake of the obstruction.[17] Enhancement of the BPF tones by operating the fan in a time-invariant, nonuniform inflow field seems appropriate because most practical installations result in such an inflow. The experiment was

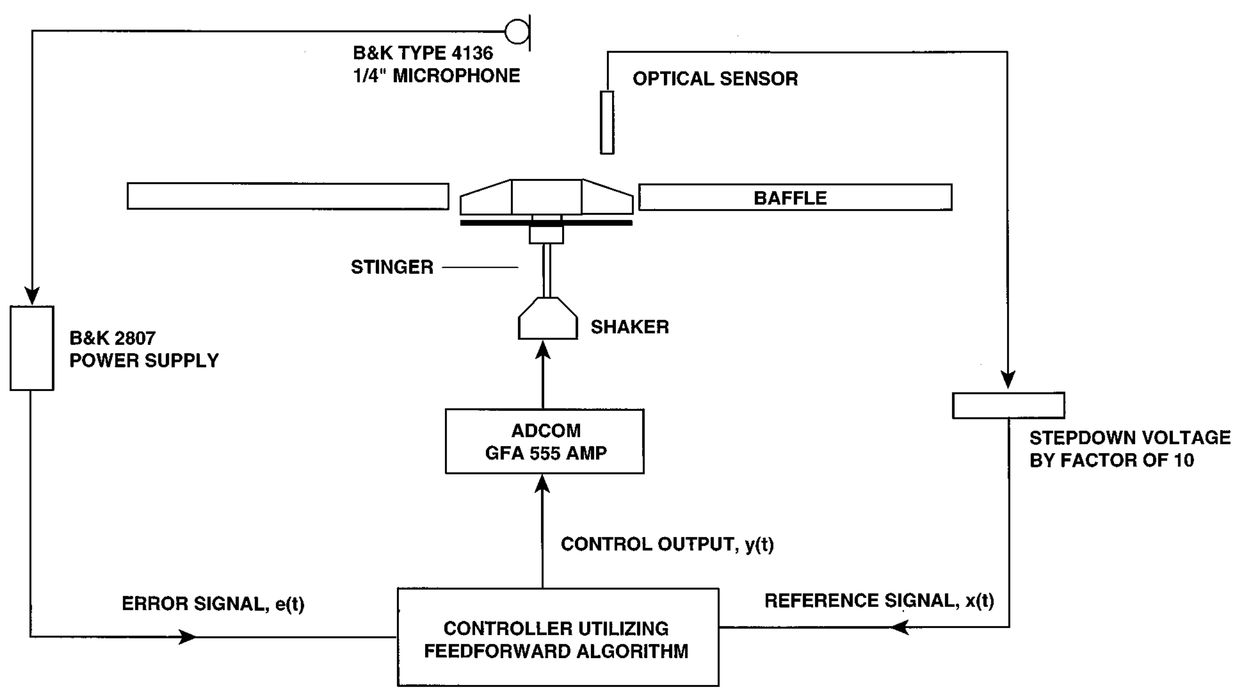

FIG. 5. Sketch of the experimental setup used to demonstrate active fan noise control.

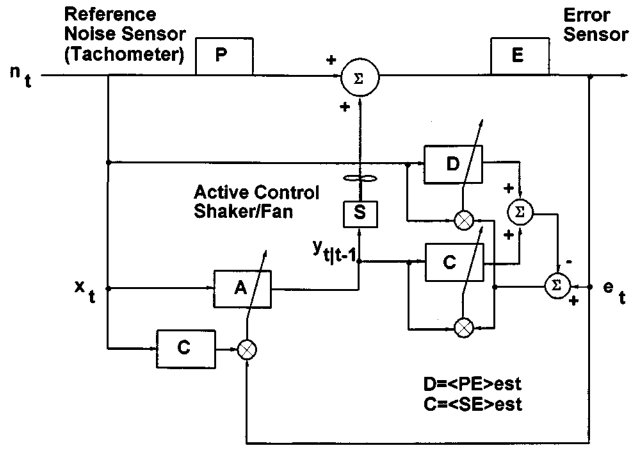

FIG. 6. Filtered-X adaptive noise cancellation block diagram used for axial fan noise reduction.

conducted in the anechoic chamber described in Sec. I A, with acoustic spectral analysis being performed using the same HP analyzer. Sound power measurements followed standard procedure.[18]

B. Control algorithm

Figure 6 presents a block diagram representing the signal processing and control for the active fan noise controller using a Filtered-X algorithm.[15] The reference signal x_t is generated by the optical sensor described in Sec. II A. The Fourier transform of the reference signal pulse train reveals exactly the same harmonic frequencies as the tonal acoustic noise generated by the fan. The block P in Fig. 6 represents the transfer function relating the magnitude and phase of the reference signal pulse train to that for the acoustic noise at the fan location. The block S represents the transfer function for the shaker/fan assembly relating the electrical input $y_{t/t-1}$ to the acoustic response at the fan location. The block E represents the transfer function of the error sensor where the input is the acoustic noise in the near field of the fan, and the output is the electrical response of the microphone to this field. Clearly, if the adaptive control filter A adapts to a transfer function approximating $-$P/S, the vibrations of the shaker/fan assembly and the unsteady forces of the blades are superimposing in a way which suppresses the radiated acoustic tonal noise. This is because the reference signal only contains harmonics coherent with the tonal acoustic noise.

We chose a passive system identification strategy[16] for the adaptive controller because we do not wish to risk increasing the broadband noise of the actively controlled fan. For the reference signal to be properly correlated to the error signal in the adaptive least-mean-square (LMS) algorithm for the controller A, we must filter x_t by the transfer function represented by the product SE, where the output is the electrical signal from the error microphone and the input is the electrical control signal to the shaker/fan assembly. The C adaptive block in Fig. 6 models the SE transfer function and the D adaptive block models the PE transfer function at the frequencies present in the reference signal x_t and the control output signal $y_{t/t-1}$. The notation $y_{t/t-1}$ depicts the fact that the digital control signal at the shaker at time t was generated by the adaptive filter last updated at time $t-1$. This delay is important physically as it indicates an unavoidable linear phase component in the SE transfer function. If the filter model C has the proper phase response at the reference signal frequencies, the Filtered-X should converge without difficulty. It is necessary to update the three LMS filters D, C, and A in real time due to the changing transfer function responses in a real fan noise application.

To insure C has the correct phase, a second system identification filter D is used to model the forward PE. The sum of the outputs of the forward model D and error model C give a prediction of the error signal e_t. The difference between the true error signal from the microphone and the prediction of the error signal is used to update the two LMS filters used in the passive system identification. Its operation is self-correcting as long as a reference signal x_t and a control signal $y_{t/t-1}$ are present (i.e., both signals are nonzero). If the control output is low, then the error signal is dominated by the forward plant and D is a good match to PE, allowing C to model SE with the residual. Conversely, if the control output is exceedingly high, the error signal is dominated by the SE loop allowing C to closely model the error plant and D to model PE with the residual. If the error signal becomes quite small (the goal of the ANC system), all of the adaptive filters slow down and converge on the desired result. Passive on-line system identification is important to axial fan ANC because flow rates and the corresponding plant time delays are always changing and the addition of broadband noise is unacceptable. It is noted that SE is the transfer function defined as the response of the error sensor when a white noise input signal to the shaker amplifier is applied in the absence of a primary excitation signal. Figure 7 shows the magnitude and phase of this function determined experimentally. The linear phase response is expected for a simple delay path associated with acoustic propagation from the fan to the error sensor.

A numerical simulation of the operations depicted in Figs. 5 and 6 has been carried out.[13] The results indicate that the algorithm of Fig. 6 operates only on causal, periodic signals. Discrete-frequency noise was thus canceled completely in the simulations. The simulation also included broadband random components of primary noise, but these were not canceled by the Filtered-X algorithm because they simply were not part of the tachometer reference signal. If a microphone reference sensor was used, rather than the tachometer, the broadband noise detected at the fan would very likely have low coherence with the far-field acoustic broadband noise due to localized turbulent flow noise at the reference position which would not be present at the far-field position. A method for suppressing local turbulent pressure fluctuations on a microphone in a flow field has recently been demonstrated,[19] but was not used in the subject investigation.

For the active fan noise control experiments, the Filtered-X algorithm was programmed in C on a WE-DSP32C floating-point digital signal processing board which was installed in an IBM PC equipped with an Intel 486DX processor operating at 33 MHz with a Windows 3.1 real-time

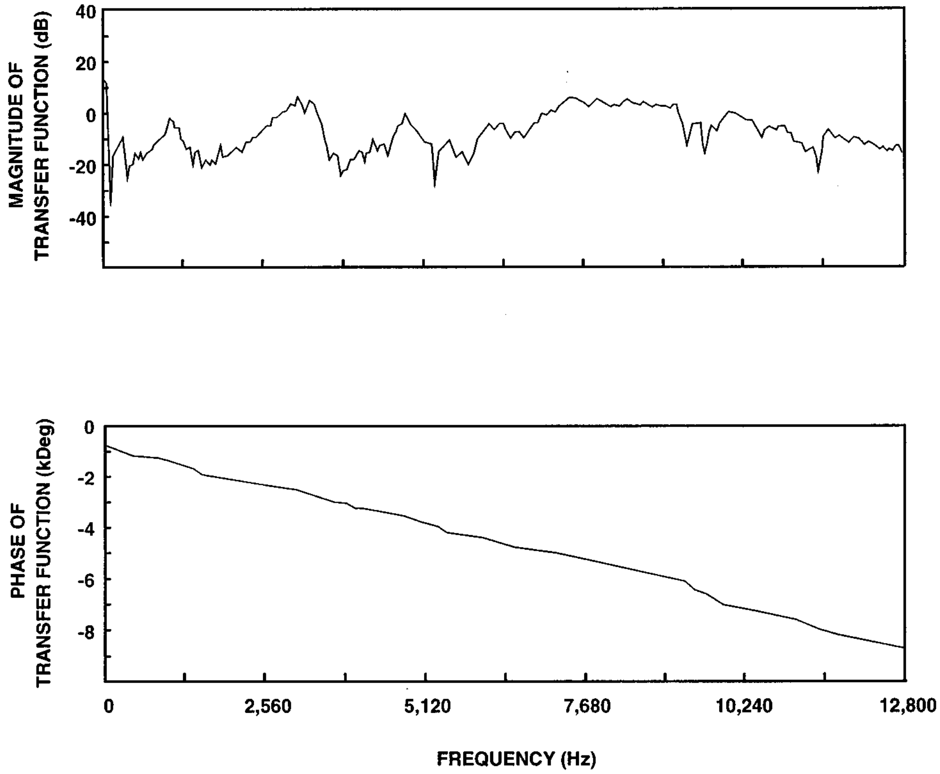

FIG. 7. Frequency response characteristics of the measured error path defined by the ratio of the Fourier transform of the input to the shaker-to-the Fourier transform of the microphone output.

user interface, also written in C. The sample rate of the controller was selected to be 2 kHz and 24 dB/octave low-pass filters were set at 900 Hz and placed at the pre-A/D and post-A/D stages of the digital signal processing board. Twenty taps (coefficients) with a step size of 0.0005 were selected for the adaptive control filter. Ten taps with a step size of 0.04 were selected for the error plant SE and ten taps with a step size of .05 were selected for the forward plant PE. On-line passive identification of the error plant (being the path representing the input to the shaker amplifier-to-the output of the error sensor) was executed in real time with the adaptive control. Using the largest step size for the D filter (in Fig. 6) modeling PE allows it to converge fastest, followed by C which models SE, and then finally the A adaptive filter, which converges to a transfer function approximating -P/S at the frequencies of the tachometer reference signal.

C. Results

Typical sound-pressure level spectra for the error sensor microphone with and without the controller on are presented in Fig. 8. The error microphone was situated approximately 19 cm above one edge of the fan frame on the inlet side. The amount of cancellation achieved at the third and higher harmonics of the BPF was found to be sensitive to the actual location of the error sensor. This is expected because the baffled fan directivity patterns become less uniform at higher harmonics of the BPF. Comparing the two spectra in Fig. 8 reveals a 20-dB reduction of the fundamental BPF tone, while the second and third harmonic levels are reduced by 15 and 8 dB, respectively. A frequency at approximately 980 Hz is also reduced. This component is related to the shaft rotation speed of 40.57 Hz and is perhaps due to a mechanical resonance of the fan unit. The fourth harmonic of the BPF at 1.136 kHz shows an 8-dB increase with the controller on. This is suspected to be due to aliasing because the low-pass filters used in the digital signal processing board were set at 900 Hz, the sampling Nyquist frequency was 1 kHz, and the 24 dB/octave filter roll-off may not be sufficient to prevent a residual component from entering the presented spectra.

The sound power radiated by the fan with and without the controller on was measured using a standard 12-point measurement procedure[18] on the inlet side of the fan over the baffle. A hemispherical surface, 0.5 m in diameter was used. Figure 9 shows the reduction in sound power level (which is indicative of global noise reduction) with the controller on as a function of frequency. The sound power at the fundamental and second harmonic BPF tones is reduced by 13 dB and 8 dB, respectively.

The sound power level reduction is some 6 or 7 dB less than the reduction in sound-pressure level measured at the error microphone location. This observation can be explained from the directivity patterns presented in Fig. 10. These patterns were measured over the baffled fan on the inlet side at the BPF, with and without the controller on. The effect of the small cylinder placed in front of the fan is evident in the ''control off'' pattern when compared to the unobstructed case of Fig. 4. The obstruction causes an approximate 10-dB increase in sound-pressure level over most observation positions. Figure 10 reveals a null in the direc-

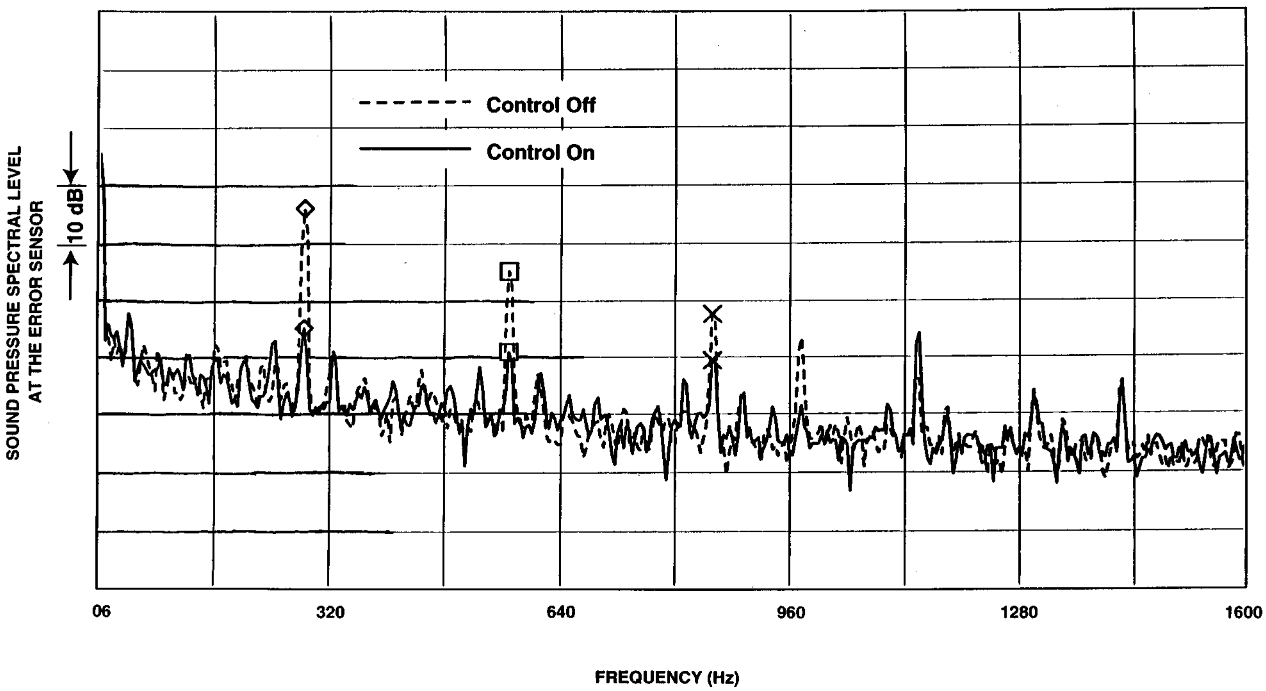

FIG. 8. Spectra of the fan sound-pressure level sensed at the error sensor position when the controller is on and off. A small cylindrical flow obstruction was placed near the fan inlet during these experiments, and the sensor is approximately 19 cm away from the edge of the fan frame, normal to the baffle.

tivity pattern along the fan axis when the control is in operation. This null clearly shows that the axial radiation is almost completely canceled by the applied axial force. Sound radiation reduction at 0° and 180° is of the order 10 dB. Obviously, these directions are less influenced by the secondary source because of its axial dipole characteristics. The error sensor was placed near the axis of the fan, so it was in the null region of the ''control on'' directivity pattern. The sound-pressure reductions are very large in this region relative to all other angular positions. This is the apparent cause of the sound power reductions being less than the sound-pressure reductions.

Additional noise reduction measurements were performed with the error microphone located on the opposite

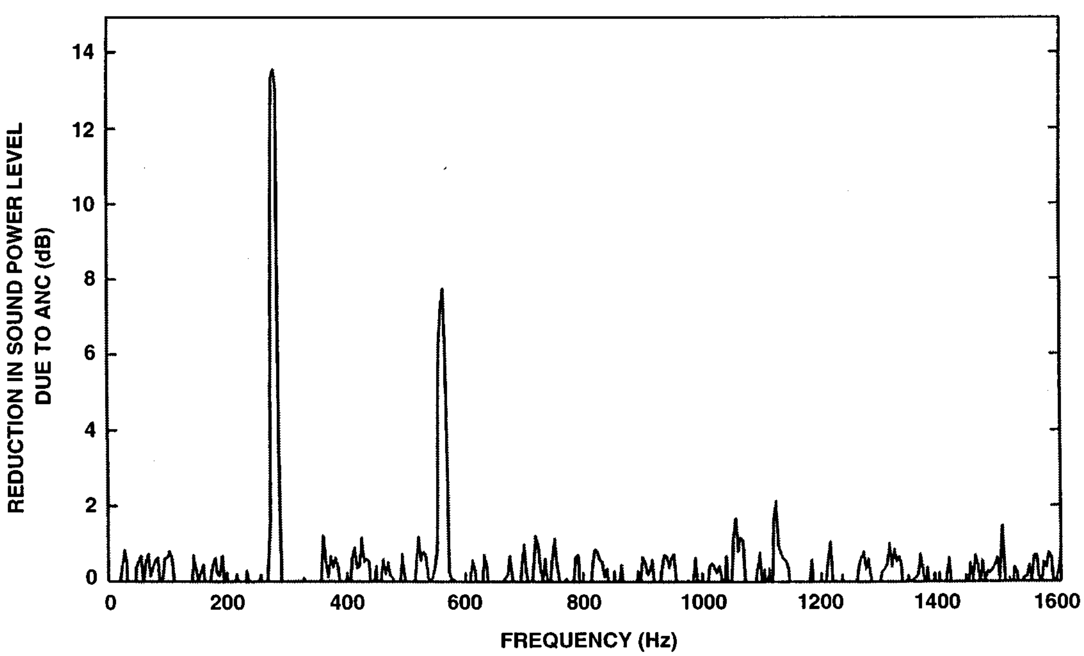

FIG. 9. Reduction in baffled axial-flow fan sound power level in dB as a function of frequency achieved using the active noise control procedures described in this paper.

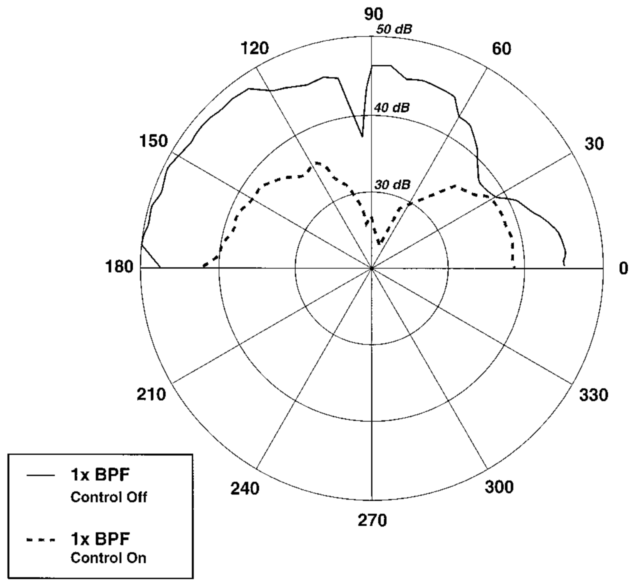

FIG. 10. Sound-pressure level directivity patterns measured for the baffled fan with the small flow obstruction in place, with and without the ANC in operation.

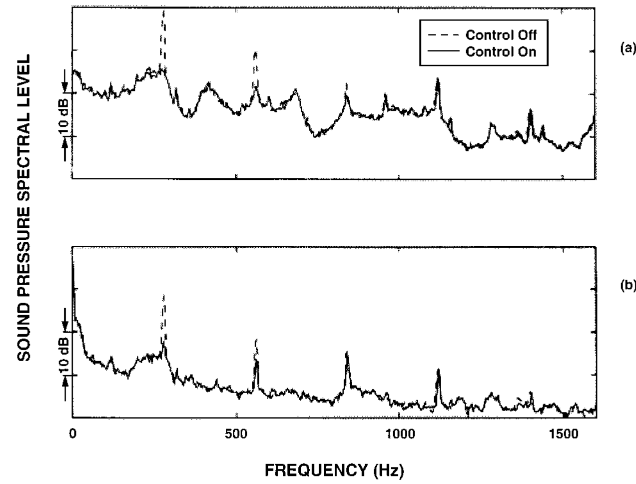

FIG. 11. Spectra of the sound pressure measured with and without the ANC in operation for the baffled fan operating inside a desktop computer cabinet: (a) the error sensor spectra measured inside the cabinet, and (b) the spectra measured at a remote location outside the cabinet.

side of the baffle (outlet side) along with a remote microphone located anywhere from 0.4 to 1.0 m away from the fan.[13] Again, the sound-pressure level at first two harmonics of the BPF was found to be reduced by 6 or more dB at the remote (and error) microphone positions. Sound power was not measured on this side of the baffle because of the proximity of the apparatus to the hard reflecting floor of the hemianechoic chamber.

As one last experiment to explore the potential applicability of the subject methodology, an empty desktop computer cabinet was placed on the planar baffle over the fan. In this arrangement, the fan pulled air into the cabinet. The air exited through the opposite side of the baffle. The error microphone was placed inside the cabinet approximately 10 cm away from the fan, and slightly off-axis. A remote microphone was positioned outside the cabinet 1 m away from the fan and 0.5 m above the baffle. Figure 11 shows the results for this experiment. The sound pressure level at the BPF is seen to be reduced by 21 dB at the external (remote) position, and by 26 dB at the internal (error microphone) position. Tone level reductions for the second harmonic are 10 and 17 dB, respectively. Sound power was not measured for the cabinet configuration, but is the subject of future applied research. The results of Fig. 11 suggest that the active noise control method employed in this study has the potential for success in a typical cooling fan application.

III. CONCLUSIONS

Although previously published research has shown that the tonal emissions from fans can be reduced through various active noise control strategies, the current research is the first to show that the fan itself, if shaken adaptively, can act as the antinoise source. This is a very significant finding because the need for a separate, secondary source is eliminated. Under the conditions of aeroacoustic compactness, the shaken fan (the secondary source) is collocated with the primary fan noise source. This tightly coupled configuration produces the excellent global noise reduction reported, and also leads to the possibility of analog feedback control strategies. This may possibly permit random noise components of the fan noise radiation spectrum, in addition to the tonal components, to be reduced. With either feedback or feedforward control, and under the premise that unsteady forces are the mechanism of subsonic fan sound production, it would be a straightforward extension of the subject methodology to utilize an internal unsteady force sensor[20] as the error sensor. Future efforts also include shaking the rotor only as opposed to shaking the entire fan assembly as was done here. Other possible modifications include multiple shakers either on individual fan blades or located at fan frame mounting lugs to aid in high-frequency cancellation when the directivity characteristics are nonuniform, and when the aeroacoustic compactness assumption is no longer valid.

ACKNOWLEDGMENTS

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