MODEL-BASED ACTIVE NOISE CONTROL: A CASE STUDY FOR A HIGH-SPEED CD-ROM SYSTEM

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Abstract: The active attenuation of the airborne noise generated from a high-speed CD-ROM system is investigated using model-based feedback control methods. A mathematical model of the physical system is developed based on a one-dimensional acoustic duct system. The critical system parameter, acoustic terminal impedance, is estimated using the eigenstructure tests through the system identification method. The considered Active Noise Control (ANC) problem is then formulated into a set of standard feedback control design problems. By using standard control methods, a set of simple feedback controllers are developed for the considered system. The simulation and physical test results show a potential to use standard control techniques for a simple, cheap but efficient ANC design and implementation.

Keywords: Active noise control, feedback control, CD-ROM airborne noise

1. INTRODUCTION

The acoustic noise pollution problem is causing more and more attention as the increasing industrialization and urbanization. The traditional approaches often use enclosures or barriers to attenuate the undesired noise. However, these passive approaches become costly and ineffective when they need to deal with the low-frequency noise (Elliott, 2001; Kou and Morgan, 1996). In order to reduce the low frequency noise, the active approaches which are usually referred to as Active Noise Control (ANC) techniques becomes more and more aware by both academic research and industrial development (Elliott, 2001; Kou and Morgan, 1996). Based on the signal superposition principle, the ANC system introduces an anti-noise wave through an appropriate set of secondary sources. Under different system structures, different algorithms can be used, such as adaptive IIR/FIR-based LMS/FXLMS methods (Kou and Morgan, 1996), adaptive or fixed feedback control methods (e.g., pole placement (Hull *et al.*, 1993) and H_∞ control (Morris, 2002)). The efficiency of different ANC systems depends on specific problems and systems.

From the control point of view, several benefits can be observed if model-based feedback control techniques are used for ANC design, such as:

- The problem - *secondary path feedback to reference measurement* (Elliott, 2001; Kou and Morgan, 1996) can be systematically dealt with by introducing a feedback loop into the physical system model (Yang, 2004).

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- The suppression of the transient signal as well as steady-state signal could be controlled through the selection of damping ratio and integral control.
- The robustness issue can be systematically handled using the existing robust control techniques (Morris, 2002).
- The global attenuation and silent range can be systematically explored using the state space models (Hull *et al.*, 1993; Yang and Hicks, 2003).

Furthermore, from the economic and practical point of view, the design of a simple but efficient and reliable ANC system is much more attractive comparing with those complicated adaptive methods, where the stability of the designed system can not be guaranteed. As one of the pioneer work, (Hull *et al.*, 1993) developed a state space model for a 1-D acoustic duct system and then proposed an ANC design using the pole placement method. (Toochinda *et al.*, 2001*b*) discussed the "hybrid ANC" design using the H_∞ and QFT techniques after the ANC design was formulated into a single-input two-output feedback control problem. (Morris, 2002) formulated the ANC problem as an H_∞-optimization problem.

From application point of view, this paper focuses on the model-based ANC design using a simple feedforward/feedback control to deal with the airborne noise generated by the operation of a high-speed CD-ROM system. By some hardware design and construction, the original 3-D acoustic attenuation problem is reduced to a 1-D ANC acoustic duct system. The secondary path feedback and coupling dynamic between the loudspeaker and acoustic duct are explicitly expressed in the developed model. The acoustic terminal impedance is estimated using the system identification method. Some simple controllers are developed for the considered system. The physical tests as well as simulation results show a large potential to use feedback control techniques for a simple but efficient ANC design.

2. PRELIMINARY ANALYSIS

The spectrum of the airborne noise generated from the considered CD-ROM system exhibits the dominant frequency located around 168 Hz (corresponding a spindle velocity of 10080 rpm at "read-mode") as shown in Fig.1.

The original 3-D noise attenuation problem is reduced to be an 1-D ANC problem by constructing an acoustic duct system as shown in Fig.2. At one end the CD-ROM system is enclosed acting as the primary noise source. A cancelling loudspeaker and two microphones are used in the considered system. From the practical point of view, the duct should be sealed properly at all openings in order to achieve low background SNR level.

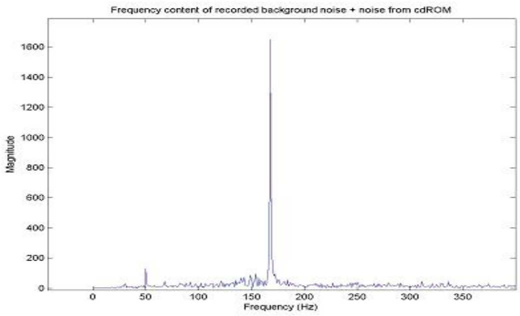

Fig. 1. The FFT spectrum of the measured noise

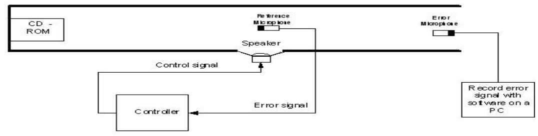

Fig. 2. The considered ANC Strategy

3. MODELLING AND IDENTIFICATION

3.1 Modelling the Loudspeaker

The basic structure of a typical low-frequency loudspeaker can be found in (Bright, 2002). Here we mainly focus on a linear model with the coupled dynamics from the rear enclosure and the front acoustic duct. The loudspeaker's information used in the considered system are listed in table 1.

Table 1 Modelling parameters of a loudspeaker

Parameter	Notation	Value	Unit
Assembly mass	m_s	$5.5 * 10^-3$	Kg
Viscous friction	f_s	0.7874	Ns/m
Suspension stiffness	k_s	$1.45 * 10^3$	N/m
Voice coil resistance	R_s	3.4	Ohm
Voice coil inductance	L_s	0.5	mH
Force factor	Bl	4	N/An
Effective radius	r_s	0.052	m
Assem. displacement	$x(t)$	variable	m
Assem. velocity	$\dot{x}(t)$	variable	m/sec
Assem. acceleration	$a_s(t)$	variable	m/sec^2
coil current	$i_s(t)$	variable	An
Input voltage	$u_{in}(t)$	variable	Volt
EMF voltage	$u_{emf}(t)$	variable	Volt

A state space model is obtained in (Yang, 2004), which can also be represented by two TFs: one denoted as $\bar{G}_{spk1}(s) \hat{=} \frac{n_{s1}(s)}{d_s(s)}$ is the TF from $u_{in}(t)$ to $\dot{x}(t)$, another denoted as $\bar{G}_{spk2}(s) \hat{=} \frac{n_{s2}(s)}{d_s(s)}$ is the TF from $p(x_s, t)$ to $\dot{x}(t)$, where

$$\begin{array}{rl} n_{s1} = & Bls, \\ n_{s2} = & -(S_d L_s s^2 + S_d R_s s), \\ d_s = & L_s m_s s^3 + (m_s R_s + f_s L_s)s^2 + (f_s R_s \\ & +k_s L_s + k_r L_s + (Bl)^2)s + (k_s + k_r)R_s. \end{array} \quad (1)$$

Here $S_d = \pi r_s^2$. Comparing with the existing models (Hull *et al.*, 1993; Morris, 2002; Toochinda *et al.*, 2001*a*), this model considers the coupling dynamics coming from the front duct and the rear enclosure (Yang, 2004).

3.2 Modelling the Acoustic Duct

Table.2 Modelling parameters of the acoustic duct

Parameter	Notation	Value	Unit
Duct length	L	1.70	m
Duct radius	a	0.052	m
Intersection area	S	$8.5 * 10^{-3}$	m^2
Sound speed	c	343	m/s
medium density	ρ	1.21	kg/m^3
Speaker loc.	x_s	variable	m
Microphone loc.	x_m	variable	m
Terminal impedance	K	variable	-
Number of modes	N	variable	-
Particle displac.	$u(x,t)$	variable	m
Particle loc.	x	variable	m
excitation($x=0$)	$p(t)$	variable	N/m^2
Mass flow	$M(t)$	variable	kg/s

In (Yang and Hicks, 2003; Yang, 2004) a state space model of the acoustic dynamics within the duct has been obtained as:

$$\begin{cases} \dot{X}_a(t) = A_a X_a(t) + B_a u_a(t) + B_p p(t) \\ y_a(t) = C_a X_a(t) \end{cases} \quad (2)$$

$X_a(t) = [\cdots\ a_{-1}(t)\ a_0(t)\ a_1(t)\ \cdots]^T$ is called the vector of modal wave amplitude. The control input $u_a(t) = \frac{d(M(t))}{dt}$ corresponds the mass flow rate generated by the cancelling loudspeaker. The output $y_a(t) = p(x_m, t)$ is the air pressure (measured by microphone) at location x_m. System matrices are defined as

$$A_a = diag(c\lambda_n) = \begin{bmatrix} \ddots & 0 & 0 & 0 & \cdots \\ \cdots & c\lambda_{-1} & 0 & 0 & \cdots \\ \cdots & 0 & c\lambda_0 & 0 & \cdots \\ \cdots & 0 & 0 & c\lambda_1 & \cdots \\ \cdots & 0 & 0 & 0 & \cdots \end{bmatrix}$$

$$B_a = col\_vector(-\frac{1}{4c\lambda_n^2 L\rho S}\frac{d(\varphi_n(x_s))}{dx})_{n=0,\pm1,\cdots} \quad (3)$$

$$B_p = col\_vector(\frac{1}{2c\lambda_n L\rho})_{n=0,\pm1,\cdots}$$

$$C_a = row\_vector(-\rho c^2 \frac{d(\varphi_n(x_m))}{dx})_{n=0,\pm1,\cdots}$$

where $\varphi_n(x)$ and λ_n are defined as

$$\lambda_n = \frac{1}{L}log_e(\frac{1-K}{1+K}) - \frac{n\pi i}{L}, \quad n = 0, \pm1, \pm2, \cdots \quad (4)$$

$$\varphi_n(x) = e^{\lambda_n x} + e^{-\lambda_n x}, \quad n = 0, \pm1, \pm2, \cdots \quad (5)$$

This model (2) can also be represented by two transfer functions as: one denoted as

$$G_{duct1}(s) \hat{=} \frac{n_{a1}(s)}{d_{a1}(s)} = C_a(sI - A_a)^{-1}B_p \quad (6)$$

is from $p(t)$ to $y_a(t)$; the other denoted as

$$G_{duct2}(s) \hat{=} \frac{n_{a2}(s)}{d_{a2}(s)} = C_a(sI - A_a)^{-1}B_a \quad (7)$$

is from $u_a(t)$ to $y_a(t)$. It can be noticed that $d_{a1}(s) = d_{a2}(s) \hat{=} d_a(s)$.

Compared with the original model developed in (Hull *et al.*, 1993), here two corrections are made and verified through simulations and tests:

- The signs of B_a and B_p are changed; and
- The formula (4) for computing the real part of λ_n.

If the terminal impedance K can be approximated by some constant value, this acoustic model (4) reduces to be a complex-valued-based two-input one-output infinite-dimensional LTI system (Hull *et al.*, 1993; Yang and Hicks, 2003; Yang, 2004).

3.3 Modelling the Entire System

The entire system model can be obtained by considering the interaction between the loudspeaker and the acoustic duct. According to the property of the acoustic mass flow, there is $M(t) = \rho S_d \dot{x}(t)$, such that $u_a(t) = \frac{d(M(t))}{dt} = \rho S_d a_s(t)$. Define:

$$\begin{array}{l} G_{spk1}(s) = \bar{G}_{spk1}(s)s, \\ G_{spk2}(s) = \bar{G}_{spk2}(s)s, \end{array} \quad (8)$$

where $\bar{G}_{spk1}(s)$ and $\bar{G}_{spk2}(s)$ are defined in (1).

The air pressures at two locations x_s and x_m need to be known. The one denoted as $p(x_s, t)$ is required by the model of the loudspeaker for modelling the dynamic coupling. The other denoted as $p(x_m, t)$ is used by the ANC controller later. Furthermore, in order to check the global noise reduction behavior, a movable *performance point* x_p (Morris, 2002) is also defined. Therefore, the following TFs are defined:

$$\begin{array}{rl} G^s_{duct1}(s) = & G_{duct1}(s) \text{ when } x_m = x_s, \\ G^s_{duct2}(s) = & G_{duct2}(s) \text{ when } x_m = x_s, \\ G^{sm}_{duct}(s) = & \dfrac{G^m_{duct1}(s)}{G^s_{duct1}(s)} = \dfrac{n^m_{a1}(s)}{n^s_{a1}(s)}, \\ G^{mp}_{duct}(s) = & \dfrac{G^p_{duct1}(s)}{G^m_{duct1}(s)} = \dfrac{n^p_{a1}(s)}{n^s_{a1}(m)}, \end{array} \quad (9)$$

The complete system block-diagram is shown in Fig.3. Within the design and numerical simulation procedures, this model can be truncated by taking

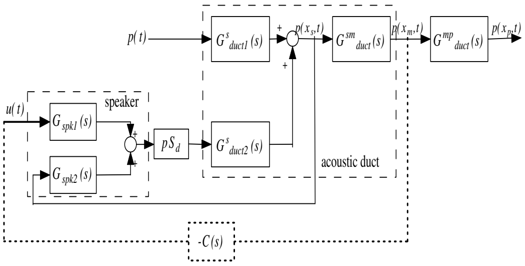

Fig. 3. Block diagram of the entire system

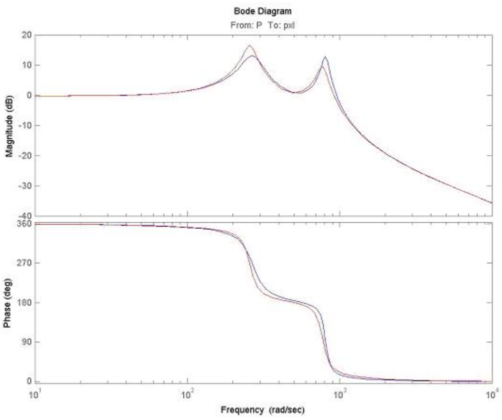

Fig. 4. Comparison between the developed model (2) (red line) and the model estimated through SI-Toolbox (blue line)

several acoustic modes through selecting N with respect to the fact that ANC system is mainly used to deal with low frequency noise.

3.4 Identification of the Terminal Impedance

The acoustic terminal impedance K is a complex and frequency-dependent parameter (Beranek, 1986). However, K should be approximated by certain constant value in order to employ the developed models. Here we use the eigenstructure test (Hull and Radcliffe, 1991) based on the system identification method to estimate this parameter. The average of obtained Ks derived from (4) based on the estimated eigenvalues is used into the system model (2) so as to check the consistency between this model with the model obtained through SI Toolbox. One result for our considered system has obtained as $K = 0,2265+1.0025i$. The comparison of two models within frequency period from 100 Hz to 250 Hz is shown in Fig.4. It can be seen that these two models have nearly identical resonance frequencies and resonance peaks.

3.5 Validation of the Acoustic Duct Model

Further comparison between the developed model (2) and the real system has been done in a lab. physical setup. From Fig. 5 it can be observed that the resonance frequencies of the developed model are almost consistent with the practical test. The real test has a slightly smaller/larger amplification around the first/second resonance frequency comparing with the developed model. This phenomenon may be due to the modelling limitation that the terminal impedance K can only be a constant while it should be frequency dependent (Beranek, 1986).

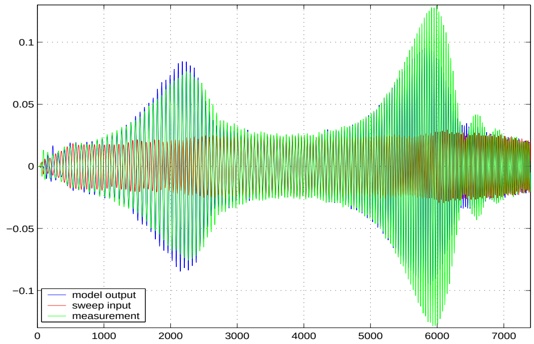

Fig. 5. Responses of the model and the duct for a sweep input

4. ANC DESIGN USING FEEDBACK CONTROL TECHNIQUES

The system diagram shown in Fig.3 can be simplified as shown in Fig.6, where

$$\begin{aligned} G_p(s) &= \frac{G^s_{duct1}}{1-\rho S_d G^s_{duct2} G_{spk2}} \\ G_u(s) &= \frac{\rho S_d G^s_{duct2}}{1-\rho S_d G^s_{duct2} G_{spk2}} \end{aligned} \quad (10)$$

The closed-loop system from the primary noise input $p(t)$ to the measured pressure $p(x_m, t)$ using a feedback controller, denoted as $C(s)$, can be expressed as

$$G_{cl}(s) \hat{=} \frac{P(x_m, s)}{P(s)} = \frac{G_p(s) G^{sm}_{duct}}{1 + C(s) G^{sm}_{duct} G_u(s)} \quad (11)$$

Denote (11) as $G_{cl}(s) \hat{=} \frac{n_{cl}(s)}{d_{cl}(s)}$.

The *basic ANC design problem* is defined as (Yang and Hicks, 2003): to find two polynomials $n_c(s)$ and $d_c(s)$ such that

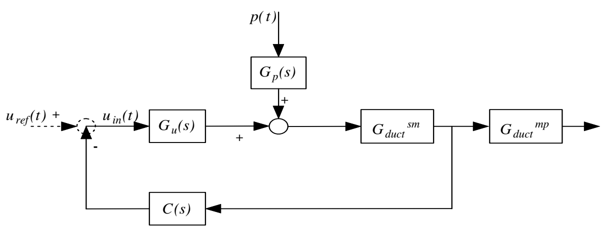

Fig. 6. ANC Control Using Disturbance Attenuation Idea

- (i) Polynomial $d_{cl}(s)$ is stable;
- (ii)There is $|n_{cl}(s)| \ll |d_{cl}(s)|$ for $\omega \in [0, B_{bw}]$; and
- (iii) The order of $n_{cl}(s)$ is not higher than the order of $d_{cl}(s)$.

where frequency period $[0, B_{bw}]$ is usually referred to as *ANC effective bandwidth* (Elliott, 2001).

By using the movable performance point x_p, the *global ANC design problem* is defined as (Yang and Hicks, 2003): to find two polynomials $n_c(s)$ and $d_c(s)$ such that

- condition (i) and (iii) in the basic design problem should be satisfied; and
- (iv) there is $|n_{cl}(s)G^{mp}_{duct}(x_p)| \ll |d_{cl}(s)|$ for $\omega \in [0, B_{bw}]$ as well as any $x_p \in (0, L)$, where the complex $G^{mp}_{duct}(x_p)$ can be calculated through (9).

Several simple PID-like/lead-lag controllers have been developed using the disturbance attenuation method and implemented for the considered system (Yang and Hicks, 2003; Yang, 2004).

5. CD-ROM CASE STUDY: SIMULATION AND PHYSICAL TESTS

5.1 Simulation Tests

When a total of 8 acoustic modes are considered in the developed model (2), the frequency properties of the system with and without developed ANC are illustrated in Fig.7. It can be observed that the noise attenuation can be achieved at nearly all frequency points. If we shift the performance point x_p (virtual microphone) along the duct, the global attenuation can also be clearly observed (Yang, 2004).

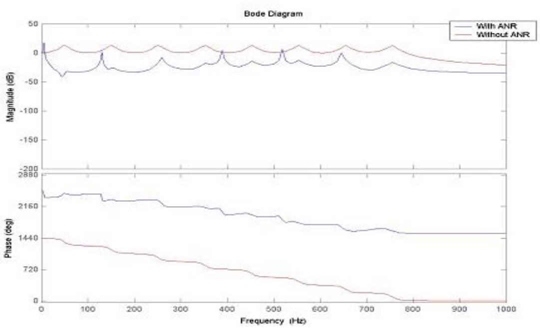

Fig. 7. Frequency properties with and without the developed ANC

5.2 Physical Tests

It can be observed that the CD-ROM airborne noise can be obviously attenuated when the ANC system is switched on as shown in Fig.8. The attenuation level is around 4.5dB.

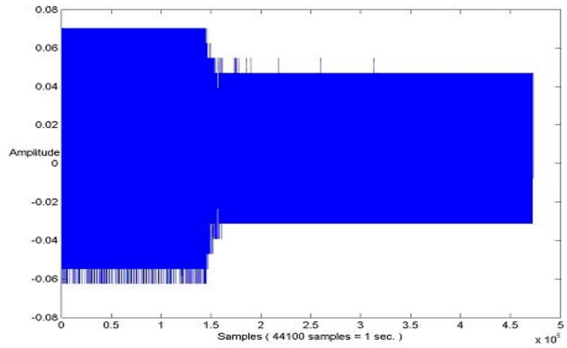

Fig. 8. Recorded signal though the performance microphone when the ANC is switched on or off

In order to test the developed ANC for the airborne noise generated from the total CD-ROM operation which includes read, start-up and shutdown modes, a sweep signal (from 60 Hz to 250 Hz) is used to test the developed system. Figure (9) showed that some resonance frequency peaks have been attenuated, but the peaks over 300Hz frequency range have been amplified by the ANC system. This phenomenon is called *waterbed phenomenon* (Elliott, 2001).

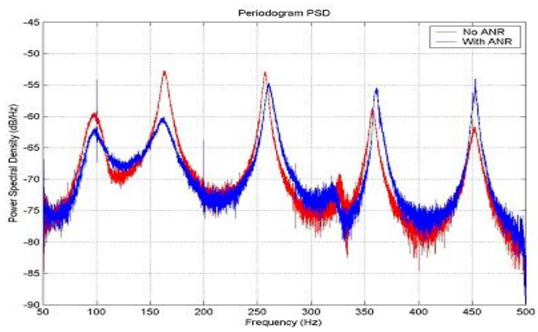

Fig. 9. The PSD of the measured (performance) signal for a sweep input

5.3 ANC Design Using Feedforward Control

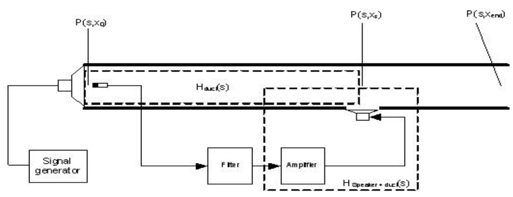

Fig. 10. Feedforward ANC structure

When the microphone used for the feedback control purpose moves very close to the original noise resource (CD-ROM or a loudspeaker used to simulate the noise resource) as shown in Fig.10, the specific ANC design problem can be dealt with using the feedforward control strategy. The difficulty in the design of feedforward controller/filter is to deal with the nonminimum-phase system model (Yang and Hicks, 2003).

A simple PID-like analog controller was developed using the all-pass filter decomposition for the CD-ROM ANC problem. One laboratory test is shown in Fig.8, where the input signal is centralized around 168Hz (to simulate the CD-ROM noise). An obvious attenuation can be observed at the end of the duct when the developed ANC switches on. The maximum attenuation can reach 13dB.

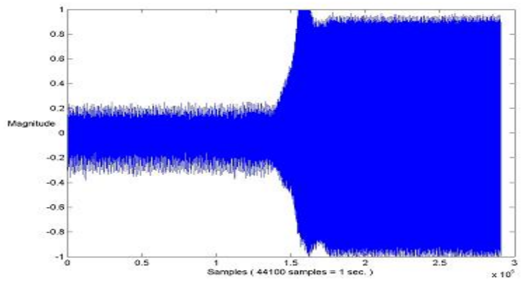

Fig. 11. Recorded (performance) signal when the feedforward ANC is switched on or off

The developed feedforward ANC system is further tested using a sweep signal from 50Hz to 250Hz. From Fig.12 similar results as shown in Fig.8 are observed.

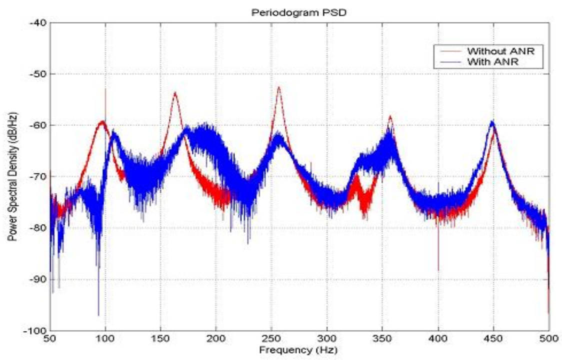

Fig. 12. The PSD of the Measured Residual for a Sweep signal

6. CONCLUSIONS

Both feedback and feedforward ANC systems developed for the CD-ROM noise problem are second-order fixed-coefficient controllers, which are easy and cheap to be designed, implemented and maintained. Concerning to the airborne noise generated during the CD-ROM's read mode, the feedforward controller gained more attenuation at the end of the duct comparing with the case using feedback controller. However, both controllers achieved more or less same attenuation performances when the CD-ROM carried a complete operation. The simulation and physical testing show a large potential to use standard control techniques for a simple, cheap but efficient ANC design and implementation.

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