Proceedings of the American Control Conference
Philadelphia, Pennsylvania • June 1998

Active Control of Noise using a Pneumatic Servovalve

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Abstract

In this paper we demonstrate the applicability of a high speed pneumatic servovalve for active noise control (ANC). Pressurized air is fluctuated using the servovalve to produce sound at commandable amplitude and frequency. The paper presents results of a noise cancellation experiment with an acoustic duct structure where disturbance produced by a speaker is overwhelmed by the high speed pneumatic servovalve. Single-tone and dual-tone disturbances at unknown frequencies are successfully rejected using the ARMARKOV adaptive control disturbance rejection algorithm.

1. Introduction

Active noise control (ANC) methods for noise suppression have received increasing attention in recent years [6, 8]. ANC involves the use of secondary noise sources such as loudspeakers as control inputs, which are used to generate an "anti-sound" field which destructively interferes with the undesired noise field.

The latest advancements in affordable and high-speed digital signal processing have made ANC techniques practical and realizable. These techniques have been successfully applied in industry, mainly in one-dimensional sound fields such as plane waves in ducts.

Loudspeakers are the most commonly used acoustic actuators. They are inexpensive, reliable and have a broad band of operation. However, loudspeakers frequently lack sufficient control authority at low frequencies (below 200 Hz) and thus a large speaker or a large number of speakers are needed to achieve adequate control authority at low frequencies. For example, in [11] ANC applied to jet engine exhaust noise required 72 actuators in 12 enclosures situated around the periphery of the exit. Furthermore, it was observed that speakers have minimal effect on low frequency noise suppression applied to cabin noise [9]. Thus the use of loudspeakers is severely limited by the availability of space.

This paper proposes an alternative actuator for low frequency noise suppression, namely, a high-speed pneumatic servovalve. Sound at a commandable frequency and amplitude can be produced by fluctuating air flow over its mean flow rate value. The amplitude of the sound can be controlled by changing the flowrate and is limited only by the choking of the valve.

In this paper we consider the use of a high speed servovalve called a *direct drive servovalve* manufactured by HR Textron Inc. The frequency response of the valve has uniform magnitude in the lower frequency range (0-200 Hz). The advantage of using a high speed servovalve is that high amplitude low frequency sound waves can be produced from a very small orifice (about $\frac{1}{4}''$ to $\frac{1}{2}''$ dia.) hence saving space. A much large space would be consumed by a equally capable low frequency woofer (6″ dia).

This research proposes the use of the high-speed valve for active noise and vibration control along with model identification techniques as in [1, 2] and active/adaptive control techniques developed in [12]. The ARMARKOV adaptive control (AAC) algorithm [12] is a powerful adaptive control method requiring minimal knowledge of the plant and disturbance.

Section 2 describes the HR Textron direct drive servovalve. This section points out that the valve as far as its acoustic properties are concerned is inherently nonlinear. Section 3 reviews the AAC algorithm. Section 4 describes the noise control experiment using the servovalve and presents the results.

2. Description of the Servovalve

The HR Textron direct drive servovalve (DDV) is designed for high pressure pneumatic and hydraulic application. A limited angle, rotary electric motor drives the valve spool directly through an eccentric shaft attached to the motor shaft (Figure 1). Thus rotation of the motor results in a linear spool motion which modulates the flow of fluid from the pressure (P) port through the cylinder ($C1$ or $C2$) ports of the valve. The flow is then ported to the system return (R).

The electric motor is a brushless DC motor with the armature immersed inside the fluid thus having minimal chance for leakage.

The valve has an integrated position controller which helps it achieve fast tracking. This controller compares the command input signal with the actual spool position and generates a current output to drive the motor to the com-

*This research was supported in part by the Air Force Office of Scientific Research under grant F49620-98-1-0037 and the University of Michigan Office of the Vice President for Research.

manded position. The signals are enhanced by electronics for optimum valve performance and linearity.

Since an accurate linear model of the valve is not available, the transfer function of the valve is found experimentally. The valve is given a voltage command input so as to fluctuate the inlet air (at 70 psi). The acoustic response was measured using a microphone placed near the outlet of the valve.

The measured frequency response of the microphone (in volts) to unit strength white noise voltage input is plotted in Figure 3. Figure 3 also shows the identified model as compared to the measured frequency response. An approximate transfer function of the valve using frequency domain identification [1] was found to be

$$G(s) = \frac{1.24 \cdot 10^6 (s^2 - 5.34 \cdot 10^3 s - 8.74 \cdot 10^5)}{s^4 + 2400 s^3 + 1.59 \cdot 10^7 s^2 + 10^{10} s + 2.74 \cdot 10^{13}} \tag{1}$$

The measured response of the valve to a sinusoidal input as shown in Figure 2 indicates the nonlinearity in the valve. The frequency response of the valve to a sinusoidal input (not shown) shows harmonics in addition to the fundamental confirming the presence of an inherent nonlinearity.

In the next section we briefly review the ARMARKOV adaptive control algorithm which is used for active noise control using the servovalve.

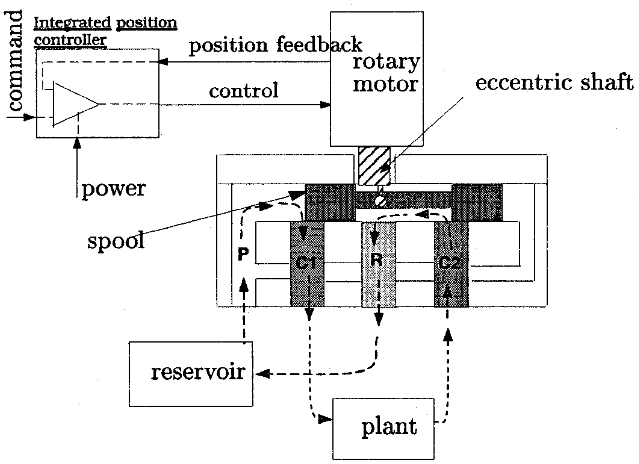

Figure 1: Schematic of HR Textron Servovalve.

3. Adaptive Control

Due to the inherent nonlinearity and complex flow properties inside the valve, an accurate physical model is not available. This calls for a more robust controller design which relies as little as possible on the accuracy of the model. The adaptive control technique developed in [12, 13] requires limited knowledge of the plant and has been demonstrated to be effective in active noise suppression in acoustic ducts.

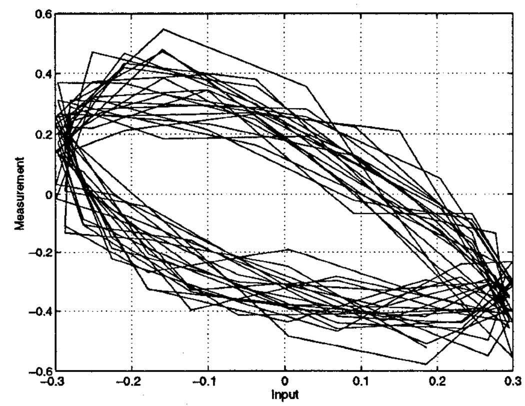

Figure 2: Phase plot of measured response of the servovalve to a sinusoidal input.

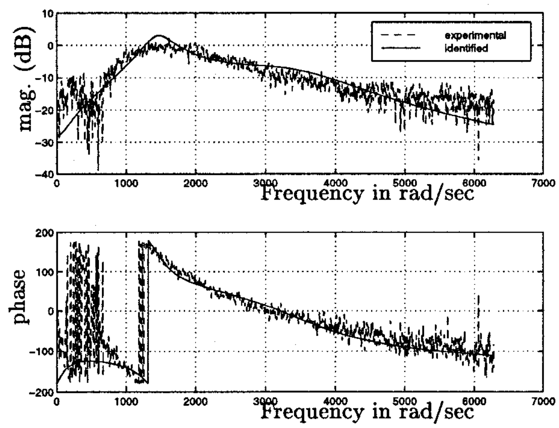

Figure 3: Measured frequency response of the servovalve.

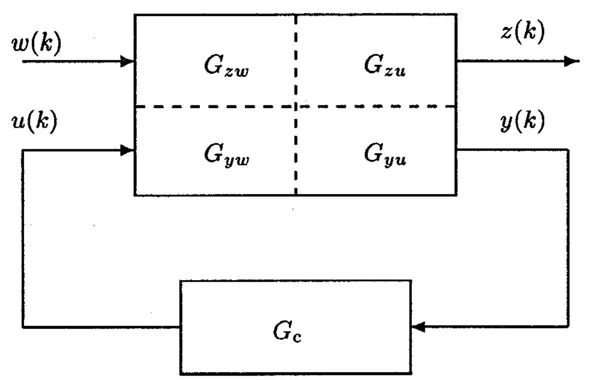

Figure 4: Standard problem with fixed-gain controller

For disturbance rejection the ARMARKOV adaptive control (AAC) algorithm [12] uses a standard problem representation for disturbance rejection. Consider the linear discrete-time two vector-input, two vector-output (TITO) system shown in Figure 4. The *disturbance* $w(k)$, the *control* $u(k)$, the *measurement* $y(k)$ and the *performance* $z(k)$ are in $\mathcal{R}^{m_w}, \mathcal{R}^{m_u}, \mathcal{R}^{l_y}$ and $\mathcal{R}^{l_z}$, respectively. The system can be written in state space form as

$$\begin{aligned} x(k+1) &= Ax(k) + Bu(k) + D_1 w(k), && (2)\\ z(k) &= E_1 x(k) + E_2 u(k) + E_0 w(k), && (3)\\ y(k) &= Cx(k) + Du(k) + D_2 w(k), && (4) \end{aligned}$$

or equivalently in terms of transfer matrices

$$\begin{aligned} z &= G_{zw} w + G_{zu} u, && (5)\\ y &= G_{yw} w + G_{yu} u. && (6) \end{aligned}$$

The controller G_c generates the control signal $u(k)$ based on the measurement $y(k)$, that is,

$$u = G_c y. \qquad (7)$$

The objective of the standard problem is to determine a controller G_c that produces a control signal $u(k)$ based on the measurement $y(k)$ such that a performance measure involving $z(k)$ is minimized.

The ARMARKOV adaptive control algorithm is based on ARMARKOV/Toeplitz representation of linear systems. These representations are parametrized by μ Markov parameters, order n and a integer p which determines the predictive data window and the length of the regressor vector. We can write the ARMARKOV/Toeplitz model of (2)-(4) [12, 13] as

$$\begin{aligned} Z(k) &= W_{zw}\Phi_{zw}(k) + B_{zu}U(k), && (8)\\ Y(k) &= W_{yw}\Phi_{yw}(k) + B_{yu}U(k). && (9) \end{aligned}$$

Here W_{zw} and W_{yw} are the block-Toeplitz *ARMARKOV weight matrices.* $B_{zu} \in \mathcal{R}^{pl_z \times p_c m_u}$ and $B_{yu} \in \mathcal{R}^{pl_y \times p_c l_u}$ are the block-Toeplitz *ARMARKOV control matrices* as defined in [12, 13]. The *extended performance vector* $Z(k)$, the *extended measurement vector* $Y(k)$ and the *extended control vector* $U(k)$ are defined as [12, 13]

$$\begin{aligned} Z(k) &\triangleq [\, z(k) \ \cdots \ z(k-p+1)\,]^{\mathrm{T}},\\ Y(k) &\triangleq [\, y(k) \ \cdots \ y(k-p+1)\,]^{\mathrm{T}},\\ U(k) &\triangleq [\, u(k) \ \cdots \ u(k-p_c+1)\,]^{\mathrm{T}}, && (10) \end{aligned}$$

where $p_c \triangleq \mu + n + p - 1$. The *ARMARKOV regressor vectors* $\Phi_{zw}(k)$ and $\Phi_{yw}(k)$ are given by

$$\begin{aligned} \Phi_{zw}(k) &\triangleq [\, z(k-\mu) \ \cdots \ z(k-\mu-p-n+2)\\ &\qquad w(k) \ \cdots \ w(k-\mu-p-n+2)\,]^{\mathrm{T}},\\ \Phi_{yw}(k) &\triangleq [\, y(k-\mu) \ \cdots \ y(k-\mu-p-n+2)\\ &\qquad w(k) \ \cdots \ w(k-\mu-p-n+2)\,]^{\mathrm{T}}. && (11) \end{aligned}$$

Next we review the ARMARKOV adaptive control disturbance rejection algorithm for the TITO system represented by (8) and (9). The ARMARKOV/Toeplitz representation of the controller of order n_c and μ_c Markov parameters is given by

$$U(k) = \sum_{i=1}^{p_c} L_i \theta(k-i+1) R_i \Phi_{uy}(k). \qquad (12)$$

Here we define the *controller parameter block vector* $\theta(k)$ as

$$\begin{aligned} \theta(k) \triangleq [\, &-\alpha_{c,1}(k) I_{m_u} \ \cdots \ -\alpha_{c,n_c}(k) I_{m_u} \ \ H_{c,0}(k)\\ &\cdots \ H_{c,\mu_c-2}(k) \ \ \mathcal{B}_{c,1}(k) \ \cdots \ \mathcal{B}_{c,n_c}(k)\,]. && (13) \end{aligned}$$

where $H_{c,j} \in \mathcal{R}^{m_u \times l_y}$ are the Markov parameters of the controller and $\alpha_{c,i}(k)$ are parameters associated with the μ_c-ARMARKOV model of the controller. Furthermore

$$\begin{aligned} \Phi_{uy}(k) \triangleq [\, &u(k-\mu_c) \ \cdots \ u(k-\mu_c-n_c-p_c+2)\\ &y(k-1) \ \cdots \ y(k-\mu_c-n_c-p_c+2)\,]^{\mathrm{T}}, && (14) \end{aligned}$$

and

$$\begin{aligned} L_i &\triangleq [\, 0_{(i-1)m_u \times m_u} \ \ I_{m_u} \ \ 0_{(p_c-i)m_u \times m_u}\,]^{\mathrm{T}}, && (15)\\ R_i &\triangleq \begin{bmatrix} 0_{q_1 \times (i-1)m_u} & I_{q_1 \times q_1} & 0_{q_1 \times (p_c-i)m_u} & 0_{q_1 \times (i-1)l_y} & 0_{q_1 \times q_2} & 0_{q_1 \times (p_c-i)l_y}\\ 0_{q_2 \times (i-1)m_u} & 0_{q_2 \times q_1} & 0_{q_2 \times (p_c-i)m_u} & 0_{q_1 \times (i-1)l_y} & I_{q_2 \times q_2} & 0_{q_2 \times (p_c-i)l_y} \end{bmatrix} && (16) \end{aligned}$$

with $q_1 \triangleq n_c m_u$ and $q_2 \triangleq (n_c+\mu_c-1)l_y$. The control input $u(k)$ is given by the first element of the extended control vector $U(k)$. Thus from (8) and (12) we obtain

$$Z(k) = W_{zw}\Phi_{zw}(k) + B_{zu} \sum_{i=1}^{p_c} L_i \theta(k-i+1) R_i \Phi_{uy}(k). \qquad (17)$$

Next, we define a cost function that evaluates the performance of the current value of $\theta(k)$ based upon the behavior of the system during the previous p_c steps. Therefore, we define the *estimated performance* $\hat{Z}(k)$ by

$$\begin{aligned} \hat{Z}(k) &\triangleq W_{zw}\Phi_{zw}(k) + B_{zu} \sum_{i=1}^{p_c} L_i \theta(k) R_i \Phi_{uy}(k),\\ &= Z(k) - B_{zu}\left(U(k) - \sum_{i=1}^{p_c} L_i \theta(k) R_i \Phi_{uy}(k) \right), && (18) \end{aligned}$$

which has the same form as (17) but with $\theta(k-i+1)$ replaced by the current parameter block vector $\theta(k)$. Using (18) we define the *estimated performance cost function*

$$J(k) = \frac{1}{2}\hat{Z}^{\mathrm{T}}(k)\hat{Z}(k). \qquad (19)$$