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DSP IMPLEMENTATION OF ACTIVE NOISE CONTROLLER FOR CENTRIFUGAL FANS

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ABSTRACT
This paper describes an application of active noise control to reduce blade passing noise for centrifugal fans using an adaptive feedforward control structure with a control loudspeaker mounting at the cutoff region of fan is designed. The controller based on the filtered-X LMS with multi-sine reference input and frequency counting device has accelerated the convergence of the algorithm by using a digital signal processor. Experiments are carried out to evaluate the proposed design for reducing the fundamental blade passing noise and harmonic frequency noise at various flow rates. The results of experiment indicate that synthetic, periodic noise generated by loudspeakers can be reduced up to 25dB and blade passing noise from a practical centrifugal fan can be reduced up 6 to 7 dB.

1 - INTRODUCTION
The noise generated by fan is usually caused by the fluctuating loads on the fan blades, which results in a source of dipole order [1] and many of the sources are around the end of blades or near them. The periodic vortices can excite resonators very strongly. Most of the studies on centrifugal noise reduction were concentrated on the control of blade passing noise. The reason for the interest in this particular component of the overall noise spectrum is obvious. From a subjective point of view, the tone is generally the most annoying component and thus needs to be reduced. Many effective researches have been done to reduce the noise emitted by the fan and a lot of specialized techniques have been developed for reduce both noise sources. They have been associated with aerodynamic, and many passive silence designs have been developed for its reduction [2], [3]. In 1987, Morinushi [4] used experimental method discussed the influence of five major geometric parameters on noise and aerodynamic performance of forward curved centrifugal fans. In 1980, Neise and Koopmann [5], [6] used a quarter-wavelength acoustic resonator on the cutoff region for reducing the blade passing noise. In this paper, they indicated that the production of the tone is originated from within a concentrated region around the cutoff of the fan casing. In 1988 [7], they designed two noise control systems with control loudspeakers and vibration plates mount on the cutoff region of casing to suppress the noise using phase shifter tracking unit. In the meantime, active noise control is being advanced rapidly in the last decade due to the progress of digital signal processor technology. In these ANC configurations up to date, feedforward control has become most widely used for reduce duct noise such as fan noise when the non-acoustical reference signal is available. In the present study, a proposed application using filtered-x LMS algorithms in a centrifugal fan which showed as Figure 1 is investigated from the viewpoint of active noise control. The proposed ANC application was validated by experiments. The results showed that the system achieved blade passing noise cancellation of the centrifugal fan with a control loudspeaker mounting into the cutoff region of casing.

2 - ANC SYSTEM AND CONTROLLER IMPLEMENTATION
The filtered-x least mean squares (FXLMS) algorithm is an essential component of many ANC control systems. This algorithm is based on the well-known least mean squares algorithm (LMS). In this study, the FXLMS algorithm with synthetic reference is utilized in this control structure. The block diagram of

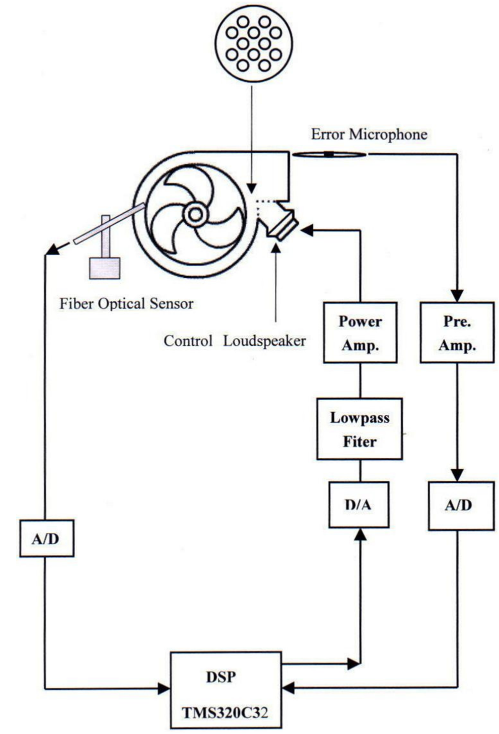

Figure 1: The experimental setup of the duct ANC system.

the FXLMS algorithm is given in Figure 2, where $y(n)$ is the control signal, W(z) is the transfer matrix of the FIR filter, $d(n)$ is the primary noise, $e(n)$ is the error signal, and the matrix $\hat{\mathbf{S}}(z)$ represents the estimated secondary path $S(z)$. The error signal can be expressed as

$$e(n) = d(n) - s(n) * \left[w^T(n)\, x(n)\right] \tag{1}$$

where $s(n)$ is the impulse response of secondary path $S(z)$.

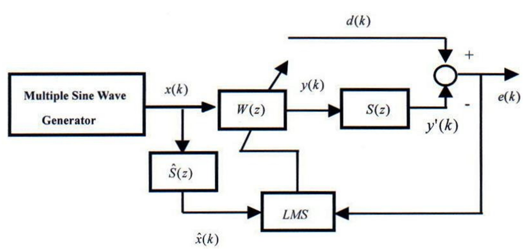

Figure 2: Adaptive feedforward control algorithm with synthetic reference input.

$w(n) = [w_0\ (n)\, w_1(n) \ldots\ldots w_{L-1}(n)\ ]^T$ is the coefficient vector of $W(z)$ at time n, $w(n) = [x\ (n)\, x(n-1) \ldots\ldots x(n-L+1)\ ]^T$ is the signal vector at time n, and L is the order of filter $W(z)$. The objective of the adaptive filter is to minimize the instantaneous squared error. Assuming a mean square cost function $\xi(n) = E\left[e^2(n)\right]$, the adaptive filter minimizes the instantaneous squared error

$$\hat{\xi}(n) = e^2(n) \tag{2}$$

using the steepest descent algorithm, which updates the coefficient vector in the negative gradient direction with step size μ.

$$w(n+1) = w(n) - \frac{\mu}{2} \nabla \hat{\xi}(n) \tag{3}$$

where $\nabla\hat{\xi}(n)$ is an instantaneous estimate of the mean-square-error (MSE) gradient at time n. In practical ANC applications, $s(z)$ is unknown and must be estimated by an additional filter $\hat{S}(z)$. The FXLMS algorithm can be written as

$$w(n+1) = w(n) + \mu x'(n)\, e(n) \tag{4}$$

Since the fan noise contains tones at the fundamental frequency and its multiples, we use a tachometer to detect the revolution of fan. In the practical measurement, the blade passing frequency is moving lightly. The blade passing frequency is moved about in the frequency range of 6 Hz. A frequency counting technique used in DSP is to actually count the time between zero crossings to a direct measure of frequency. When a zero crossing occurs, the counter is started to increment each program cycle until the next zero crossing is encountered shown as Figure 3. The count value N is stored and the counter is reset. The blade passing frequency can be obtained from

$$f = \frac{1}{2NT} \tag{5}$$

where T is the sampling period. Then we use synthetic scheme to generate a non-acoustical reference for the FXLMS algorithm. A sine wave generator based on a marginally stable two-pole resonator is employed to generate synthetic signals $x(n)$ with prescribed frequency. The transfer function of the generator is

$$H(z) = \frac{\sin(\omega_n T)\, z^{-1}}{1 - 2\cos(\omega_n T)\, z^{-1} + z^{-2}} \tag{6}$$

where ω_n is the prescribed frequency of the sinusoid. The two major benefits of this approach are that the problem of acoustic feedback is virtually eliminated and can detect actual revolution for avoid the effect of blade passing frequency variable lightly.

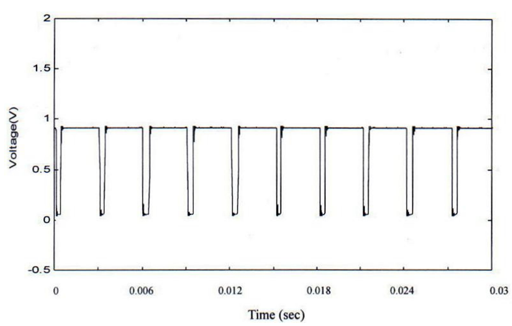

Figure 3: The output signal of tachometer.

3 - EXPERIMENTAL VERIFICATION

The first experimental setup is shown in Figure 4. A centrifugal fan of 150 mm impeller diameter with six backward curved blades. Three loudspeakers are mounted around the casing of fan for simulating the blade passing and harmonic frequency noise sources, around the end of blades or near them. These signals of noise source are generated from a function generator. A controller loudspeaker is mounted on the cutoff region of fan casing. The controllers are implemented on a TMS320C32 DSP. The results of experiment indicated that the ANC controller achieved the reduction of blade passing noise up to 25dB and harmonic frequency noise 13dB as shown in Figure 5. To explore the practicality of the proposed ANC system, a practical fan noise source is generated from the same centrifugal fan (rotating frequency is 3300 rpm) as the primary noise source. The experimental setup is shown as Figure 1. A tachometer is arranged to detect the speed of fan. The error microphone outside the fan is designed with a nose cone and a slit-tube to minimize the effect of turbulent pressure fluctuations on the microphone. The experimental results show that about 7 dB attenuation at blade passing frequency (330Hz) and 6 dB attenuation at harmonic frequency (660Hz) are achieved in flow rate is 0.01 m^3 /sec as shown in Figure 6. Obviously, the attenuation performance in sound pressure spectrum level of this implementation is much worse than without flow. It should be due to plant uncertainty and some physical effects special in flow. In order to understand the effect of controller in different flow rate, the proposed ANC system is applied in different conditions (flow rate from 0.01 to 0.02 m^3 /sec). In this experimental investigation, the effects of attenuation performance due to change in centrifugal fan flow rate are examined. The attenuation of different flow rate is summarized in Table 1. From the results, although ANC system achieved the attenuation in different conditions, the flow rate acted an important factor for ANC attenuation performance.

Flow rate (m^3/sec)	**BPF (330Hz)**	**BPF** $\times 2$ (660Hz)
0 (Synthetic noise)	**25 dB**	**13 dB**
0.010	**7 dB**	**6 dB**
0.012	**5.5 dB**	**4 dB**
0.014	**5 dB**	**3 dB**
0.016	**4 dB**	**3.5 dB**
0.018	**4 dB**	**3.5 dB**
0.020	**3 dB**	**3 dB**

Table 1: Blade passing noise attenuation in different flow rate.

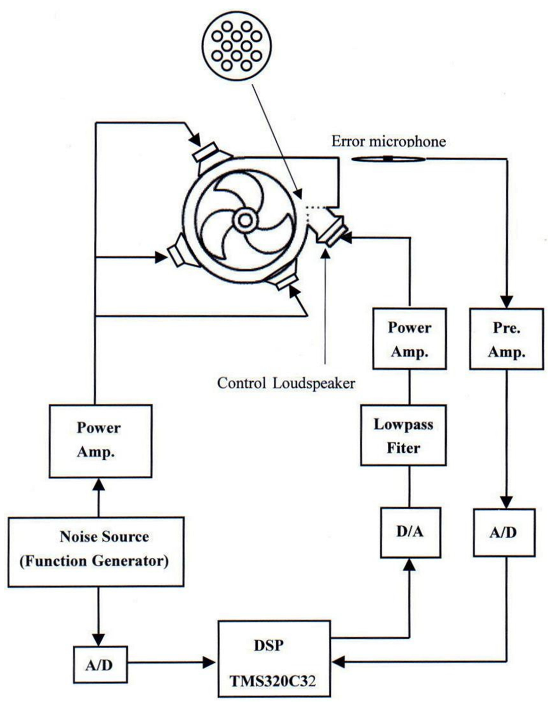

Figure 4: The experimental arrangement of the ANC system with control source mounted at cutoff region.

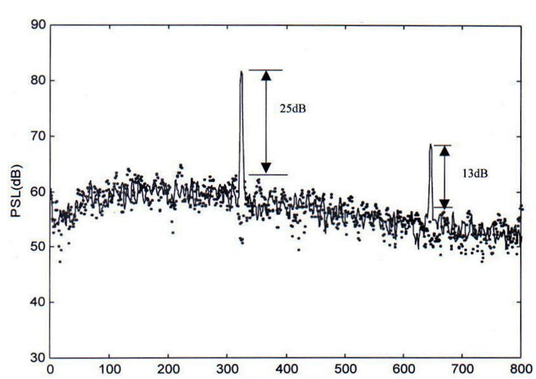

Figure 5: Experimental results of noise cancellation for stationary source; ___: control off, —: control on.

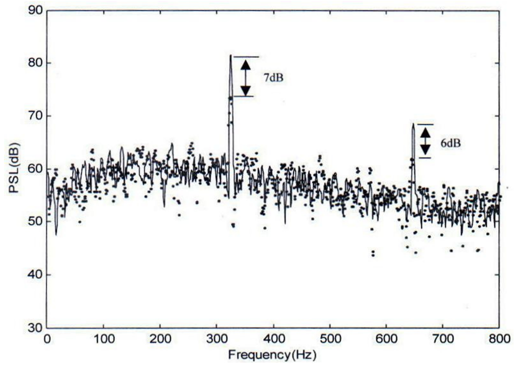

Figure 6: Experimental results of noise cancellation for fan noise; ---: control off, —: control on.

4 - CONCLUSIONS

In present study, an application of blade passing frequency noise and harmonic frequency noise from a centrifugal fan, using adaptive feedforward structure with FXLMS algorithms is investigated on the basis of acoustic application. The experimental results indicated that the system achieved attenuation up to 25dB for stationary noise and 6-7 dB for practical fan noise in blade passing frequency. The effects of attenuation performance due to change fan flow rate are examined. Plant uncertainty (flow rate in this case) is one of the major contributing factors that could affect the performance as well as stability of control systems. Development a robust adaptive controller to accommodate perturbations and plant uncertainties in the practical system is necessary.

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