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Active Noise Control in Ducts

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Original scientific paper

After reviewing the development of active noise control principles, we analyzed the elements of the active noise control system in ducts. On the basis of this analysis, we created a model of these elements. Especially, we brought a model of the loudspeaker in the *z*-domain, suitable for description of systems containing analog and digital parts. Such model enabled us to analyze work and convergence of the adaptive signal processing algorithms applied to active noise control. As an example, we analyzed performance of FXLMS algorithm on simplified model of active noise control system in ventilation duct, and have shown a strong influence of the loudspeaker's transfer function on the power spectrum of the error signal.

Key words: active noise control, adaptive filters, electroacoustical transducers, loudspeakers, ventilation ducts

1. INTRODUCTION

Large ventilation systems are often very noisy. Because of their wide application in the human neighbourhood, there is need for decreasing its noise level. Passive noise control can not give the satisfying result for low frequencies – it is too costly and it is not quit efficient.

Active noise control uses principle of destructive interference of the sound waves, e. g. in order to cancel undesired noise a sound wave with inverse sound pressure is generated. To achieve a large amount of cancellation, the anti-noise source must generate an inverted signal of original noise signal with great accuracy.

The principle of the active noise control was firstly introduced in Paul Lueg's patent in the 1930's [1]. A microphone detects unwanted sound (noise) and gives this input signal to the electronic system that drives the loudspeaker. The distance between microphone and loudspeaker, as well as electronic system itself are adjusted in this way that a destructive interference between original and generated sound waves is achieved (Figure 1). The entire process depends on relatively slow propagation of the sound waves compared to the fast processing of the electrical signals. The amplitude and phase response of the electronic system must be adjusted with a great accuracy to get a satisfying performance. There are two big problems in this approach. First, the electronic system must be capable to compensate nonuniform amplitude and phase response of transducers, filters and amplifiers. Second, the influence of the acoustical feedback between loudspeaker and input microphone must be either minimized or compensated in order to avoid unstable operation. Efficacious dealing with these two problems became possible only with application of the adaptive signal processing.

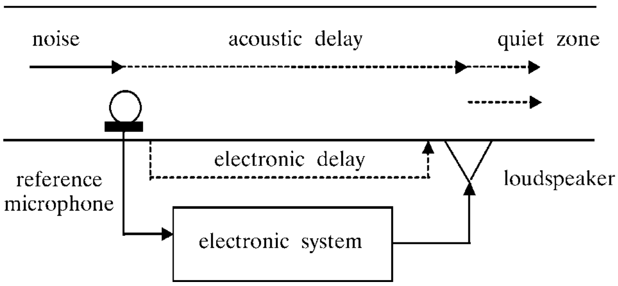

Fig. 1. Lueg's system

After Lueg, in 1953 Olson and May carried their researches on active absorber [2]. In 1956, Conover described an active system for reducing transformer's noise [3]. Conover applied manual adjusting of antinoise's phase and amplitude. Onoda and Kido developed an automated system for transformer's noise reducing in 1968 [4]. In 1981 Burgess suggested application of adaptive digital filters in active noise control [5]. In 1983 Chaplin developed a system for the cancellation of periodic noise with use of waveform generator [6]. Roure described an active noise control system for ventilation ducts based on frequency-domain analysis in 1984 [7].

The influence of acoustical feedback was compensated in ducts with use of Chelsea dipole source [8], (Figure 2), Swinbanks two-elements unidirectional source [9], (Figure 3) and use of Jessel-Mangiante-Canévet tripole source [10]. The Chelsea dipole uses a pair of loudspeakers driven out of phase and spaced one-half wavelength apart. Jessel-Mangiante-Canévet tripole uses a monopole source to cancel upstream propagation from the dipole source. There was also proposed application of FURLMS adaptive signal processing algorithm to solve this problem [11].

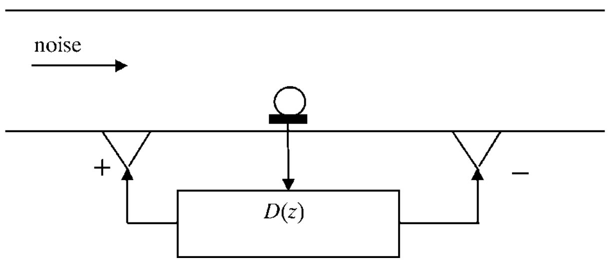

Fig. 2. Chelsea dipole

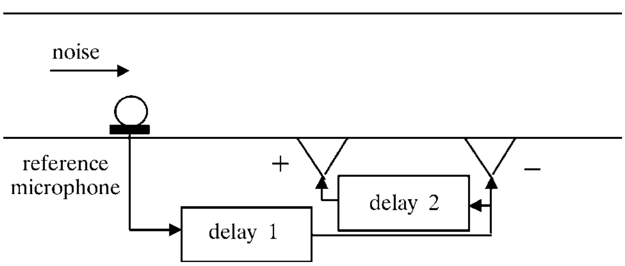

Fig. 3. Swinbanks system

From a geometric point of view, active noise control systems may be divided on duct noise control systems, interior noise control systems, free-space control systems and personal hearing protection systems. Interior noise was actively controlled in passenger cars [12], and in the propeller-driven aircrafts [13]. The occupational noise can be reduced by use of active acoustic barriers [14]. Active hearing protectors were proposed in [15]. Active control of sound has some close associations with the active control of vibration. In many cases active control of sound and vibration can be treated as one problem, especially when dealing with structure-borne sound.

In this paper a model of active noise control system for infinite ducts will be developed, useful for modeling of the active noise control systems in the ventilation systems. On the basis of this model, the efficiency of the adaptive signal processing algorithms will be analyzed.

2. MODELING OF SOUND FIELD IN INFINITE DUCTS

If the sound wave propagates through the infinite duct, and if its wavelength is much longer than duct's diameter then we can assume that we have a plane acoustical wave. In the cases when the wavelength is less of the doubled diameter of the duct, we have high-order modes in the soundwaves propagation. Although most work on active noise control systems in ducts has focused on plane-wave propagation, higher order modes can become problem for large duct dimensions even for low frequencies. This problem may be solved through the use of a partitioned duct, as described in [16]. There are efficient passive noise control measures for high frequencies with high-order propagation modes, and we will discuss ahead only low frequency soundwaves' propagation, e. g. plane acoustical waves.

Plane wave propagation of the sound in the $+x$ axis direction of the Cartesian coordinate system is described by the differential equation [1]

$$\frac{\partial^2 p}{\partial x^2} - \frac{1}{c_0^2} \times \frac{\partial^2 p}{\partial t^2} = 0 \tag{1}$$

where p denotes sound pressure, t is the time and c_0 is the velocity of the sound propagation.

The simplest antinoise source for the infinite duct is one loudspeaker fixed on the wall of the duct. For the frequencies low enough, neglecting the area close to the loudspeaker, we can assume the antinoise sound wave to be a plane wave, and we can model the loudspeaker as a plane acoustical monopole in the infinite duct [1].

A plane acoustical monopole can be imagined as a pair of two massless pistons on the infinitesimal distance, moving on the opposite directions. Their moving is directed by the time-variant volume velocity of the monopole $q(t)$. For the soundwave's lengths long enough compared to the physical dimensions of the loudspeaker, the monopole's and the loudspeaker's volume velocities will be the same.

If we have the noise source in the x_P point on the x-axis, and the secondary source positioned in

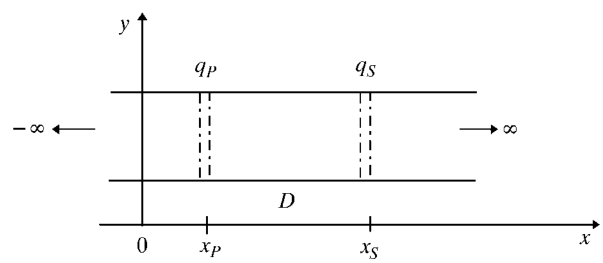

Fig. 4. Two plane acoustical monopoles in an infinite duct

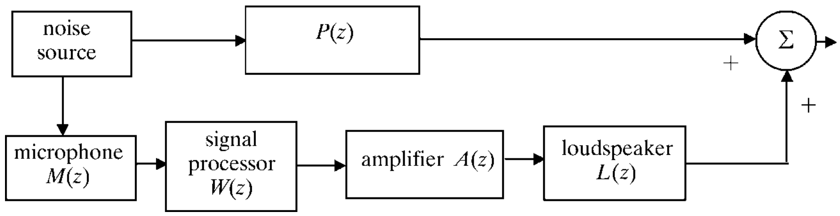

Fig. 5. The elements of active noise control system

the x_S point (Figure 4), and if their volume velocities are q_P an q_S, respectively, the sound pressure will be canceled for $x > x_S$ if following equation is satisfied:

$$q_S = -q_P\left(t - \frac{D}{c_0}\right), \tag{2}$$

where D denotes the distance between two sources [1].

3. THE ELEMENTS OF THE ACTIVE NOISE CONTROL SYSTEM

If we want to satisfy (2), we have to assure time delay between q_S and q_P equal to the time of sound waves' propagation between two sources. In the real conditions, it means that we first have to detect original noise signal using an appropriate sensor, and transduce acoustic signal to its electrical equivalent. The electric signal has to be processed for ensuring time delay condition, and after amplifying, brought to the actuator (loudspeaker).

This chain can be described by the Figure 5. All elements of an active noise control (ANC) system have their own transfer functions. $P(z)$ is transfer function of the primary channel, ideally representing by a pure time delay. The microphone, amplifier and the loudspeaker are introducing amplitude and phase changes in the system. The purpose of the signal processor is to compensate these changes, and to ensure overall time delay for satisfying condition (2) by its transfer function $W(z)$.

4. SENSORS AND AMPLIFIERS

As the noise sensor in an active control of sound system, one can use acoustic or non-acoustic sensor. Non-acoustic sensor, for instance the accelerometer, can be used in cases when noise is produced by mechanical vibrations of the noise source. For periodical noise signals, other non-acoustic sensors, for instance tachometers can be used too. When using acoustical sensors, there is a problem of unwanted acoustic feedback.

Overall characteristics of a microphone are influenced by its acoustical and electrical properties too. Good microphones could have very uniform frequency response.

Piezoelectric accelerometers have the best properties, and this type of accelerometer is almost only in use. Lower frequency limit in vibration measurements is defined by the lower frequency limit of the preamplifier and the lower frequency limit of the accelerometer itself. The lower frequency limit of the preamplifier usually can be below 1 Hz. Tempe-rature sensitivity of the accelerometer define its lower frequency limit, and using shear type one can bring this limit below 1 Hz too.

In the middle frequencies range, a power amplifier can be modeled as a voltage source with adjustable clipping distortion. Dynamic range of the power amplifier is extremely important for the proper work of the system.

5. LOUDSPEAKERS

As a part of the active noise control system, the loudspeaker influences in great measure on overall work and efficiency of such a system. The loudspeaker is a complex system consisted of electric, mechanical and acoustical parts. In this analysis an assumption that the cone moves as uniform piston is taken. Such assumption can be made for the low--frequency range.

The terminal voltage E on the voice coil of the loudspeaker is given by:

$$E = Z_E I + Blv. \tag{3}$$

Z_E is electrical impedance of the voice coil

$$Z_E = R + sL \tag{4}$$

where R is the resistance of the voice coil, L is inductance of the voice coil and s denotes complex frequency. Also l denotes wire length of voice coil, B is flux density in region of voice coil, and v is cone velocity amplitude. The relationship between terminal voltage and cone movement is given by [21]:

$$\frac{v}{E} = \frac{Bl}{Z_E Z_{MT} + (Bl)^2} \tag{5}$$

where Z_{MT} denotes total mechanical impedance. Z_{MT} is defined by

$$Z_{MT} = sm + D + \frac{K}{s} \tag{6}$$

where m denotes total effective mass of moving system (i.e. cone, voice coil and effective air mass), D denotes damping, including mechanical damping in moving system and radiation resistance, and K denotes effective stiffness of suspension and back enclosure, if any.

Generally, effective air mass and radiation resistance are frequency-varying, and are depending on mounting conditions. For loudspeaker fixed on one wall of the infinite duct, it is assumed, following results given in [18], that its radiation impedance (mechanical) can be approximated at low frequencies (below cut-on frequency of the first higher order mode) with

$$R_{MA} = \rho_0 c_0 \frac{S_L^2}{2S_D} \tag{7}$$

where ρ_0 is density of air, c_0 is velocity of sound in air, S_L is cone area, and S_D is duct area. This approximation is verified with an experiment on the short duct. A broadband loudspeaker was laterally mounted on a 70 cm long rigid duct with circular cross-section. Radius of the duct was 38 mm. Physical constants of the loudspeaker were measured, and they are listed in the Table 1. Loudspeaker was mounted 14 cm away from the one side of the duct.

Table 1. **Physical constants of the broadband loudspeaker**

Magnetic coupling factor	$Bl = 2.306$ Tm
Resistance of voice coil	7.33 Ω
Inductance of voice coil	0.2 mH
Mechanical stiffness of suspension	1962 N/m
Mass of voice coil, cone, etc.	$4.031 \cdot 10^{-4}$ kg
Mechanical damping	0.102 Ns/m
Piston diameter	77 mm

The transfer function between sound pressure on the more distant end of the duct (56 cm away from the loudspeaker) and terminal voltage was measured. The both ends of the duct were opened, and a miniature *Brüel&Kjaer* microphone was used in these measurements. The transfer function was also calculated. The radiation impedance on the ends was taken as for the pulsation sphere, and driving point impedance was calculated using transmission lines theory [19]. The losses in the duct were not taken into account. The calculated overall driving point impedance was transformed using area ratio, as in (7) on the loudspeaker loading, and the modified total mechanical impedance Z_{MT} was calculated for the each frequency. Effective loading of the air mass on the back of the loudspeaker was estimated by mass loading of a semi infinite space on a equivalent piston. From the calculated piston velocity (5) and known load impedance, the driving point sound pressure was determined, and finally, sound pressure on the more distant end. Resistance of the voice coil was increased for 50 Ω output impedance of the source. Figure 6 presents calculated and measured response. The discrepancies between two curves can be explained with the losses in the duct and directivity characteristic of the microphone.

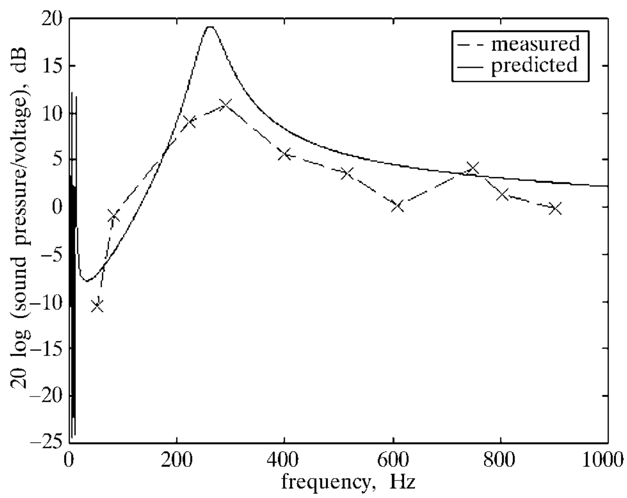

Fig. 6. Sound pressure on the end of the opened short duct

Using relations (3–7), a transfer function of the loudspeaker in s-domain can be derived:

$$H_L(s) = \frac{v(s)}{E(s)}, \tag{8}$$

so we have finally

$$H_L(s) = \frac{s}{K_3 s^3 + K_2 s^2 + K_1 s + K_0}, \tag{9}$$

with coefficients:

$$\begin{aligned} K_3 &= \frac{Lm}{Bl} \\ K_2 &= \frac{LD + Rm}{Bl} \\ K_1 &= \frac{LK + DR}{Bl} + Bl \\ K_0 &= \frac{KR}{Bl}. \end{aligned} \tag{10}$$

Applying the bilinear transformation [20]

$$s = \frac{2}{T} \cdot \frac{1-z^{-1}}{1+z^{-1}}, \tag{11}$$

a z-domain transfer function can be achieved:

$$H_L(z) = \frac{A_3 z^{-3} + A_2 z^{-2} + A_1 z^{-1} + A_0}{B_3 z^{-3} + B_2 z^{-2} + B_1 z^{-1} + 1} \tag{12}$$

with coefficients:

$$\begin{aligned}
M &= 8K_3 + 4TK_2 + 2T^2 K_1 + T^3 K_0 \\
A_3 &= \frac{-2T^2}{M} \\
A_2 &= \frac{-2T^2}{M} \\
A_1 &= \frac{2T^2}{M} \\
A_0 &= \frac{2T^2}{M} \\
B_3 &= \frac{-8K_3 + 4TK_2 - 2T^2 K_1 + T^3 K_0}{M} \\
B_2 &= \frac{24K_3 - 4TK_2 - 2T^2 K_1 + 3T^3 K_0}{M} \\
B_1 &= \frac{-24K_3 - 4TK_2 + 2T^2 K_1 + 3T^3 K_0}{M},
\end{aligned} \tag{13}$$

where T is sampling period.

Following the results from section 2, sound pressure in duct is defined for plane waves by:

$$p = \rho_0 c_0 \frac{v \cdot S_L}{2 \cdot S_D}. \tag{14}$$

To check the feasibility of the entire analysis, we compared our results with more sophisticated numerical analysis given by Shepherd et al. [21], as well as with their measurements. In their study, acoustic load on the face of the piston was determined numerically, following analysis given by Doak

Table 2. **Physical constants of the Plessy-Foster loudspeaker**

Magnetic coupling factor	$Bl = 12.8$ Tm
Resistance of voice coil	6.7 Ω
Inductance of voice coil	1 mH
Mechanical stiffness of suspension	1400 N/m
Mass of voice coil, cone, etc.	0.028 kg
Mechanical damping coefficient	4.0
Piston diameter	200 mm

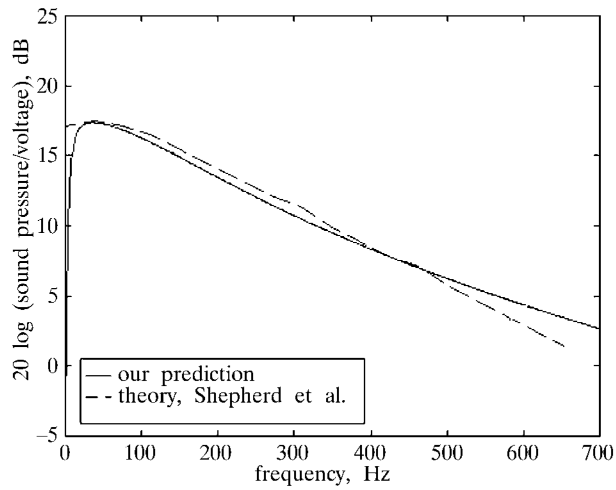

Fig. 7. Frequency response of the loudspeaker's model – magnitude

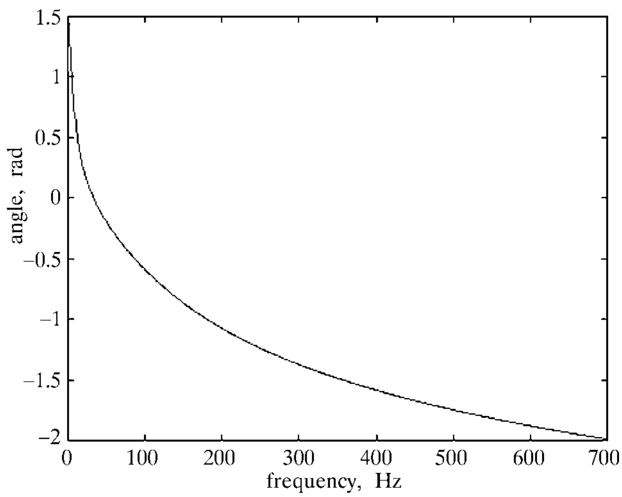

Fig. 8. Frequency response of the loudspeaker's model – phase

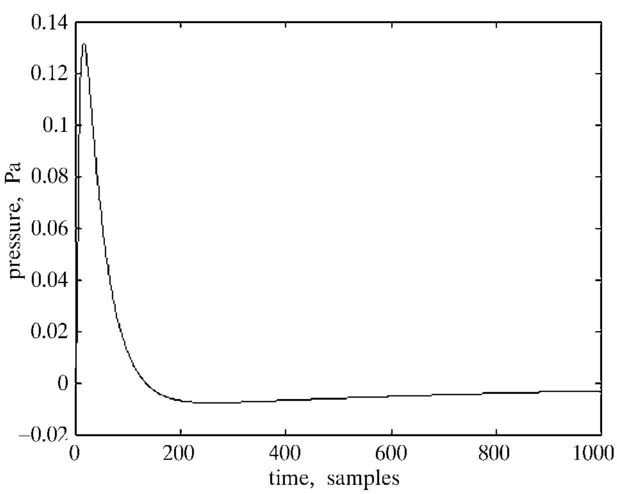

Fig. 9. Unit pulse response of the loudspeaker's model

[22], and taking into account both propagating and evanescent modes. They used 200 mm diameter Plessy Foster 200FO5 mounted in an unvented enclosure of 20 litres. The attenuator was set up on a rigid walled duct of 244 mm square section. Physical constants of that loudspeaker are listed in Table 2. The effective air mass was estimated by mass loading of a semi infinite space on a piston, and suspension stiffness was not increased to account for the enclosure [21].

With sampling frequency $1/T = 40$ kHz, covering audio frequencies range, we obtained loudspeaker's response in examined duct presented with figures 7–9. There are also presented predictions, given by Shepherd et al. Frequency response is presented only for the frequencies below the cut-on frequency of the first higher order mode in this duct. Comparing their results with our predictions, one can see very good matching.

6. SIGNAL PROCESSING

To ensure proper work of the system, signal processing has to be applied. It can be done by fixed or adaptive filters. The need for a very accurate phase and amplitude response leads toward application of the adaptive signal processing. It is caused by the fact that properties of transducers, as well as the properties of electronic elements are time and temperature varying. The air temperature also influences on the sound propagation speed.

If we idealize properties of microphones and amplifiers, as well as acoustic path properties between noise source and secondary source, and if we apply loudspeaker model described in section 5, the application of the classical FXLMS adaptive algorithm [23, 24] can be represented with Figure 10, where $e(n)$ denotes error signal, and $d(n)$ is unwanted noise signal that is acoustically summing with antinoise signal $y'(n)$.

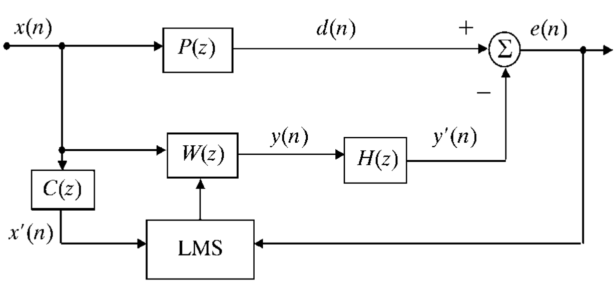

Fig. 10. FXLMS algorithm

Filter output is defined by

$$y(n) = \mathbf{X}^T(n)\mathbf{W}(n), \tag{12}$$

error signal is

$$e(n) = d(n) - y'(n), \tag{13}$$

and coefficients updating is

$$\mathbf{W}(n+1) = \mathbf{W}(n) + 2\mu\mathbf{X}'(n)e(n) \tag{14}$$

where $\mathbf{W}$ is weight vector of the adaptive FIR filter, μ is step size that satisfies stability condition [23], $\mathbf{X}$ is vector of the input samples and $\mathbf{X}'$ is vector of input samples filtered by secondary path estimate $C(z)$. Generally, $C(z)$ is achieved by some method of system identification.

For simulation purposes, it is assumed that $P(z)$ includes only time delay:

$$P(z) = z^{-60}. \tag{15}$$

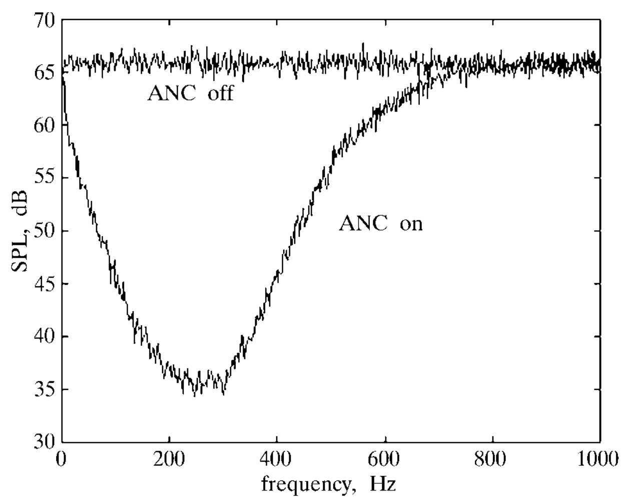

Fig. 11. Power spectra of the residual noise (mean of 50 independent trials) before adaptive processing (ANC off) and during adaptive procesing (i.e. 1300 iterations, ANC on)

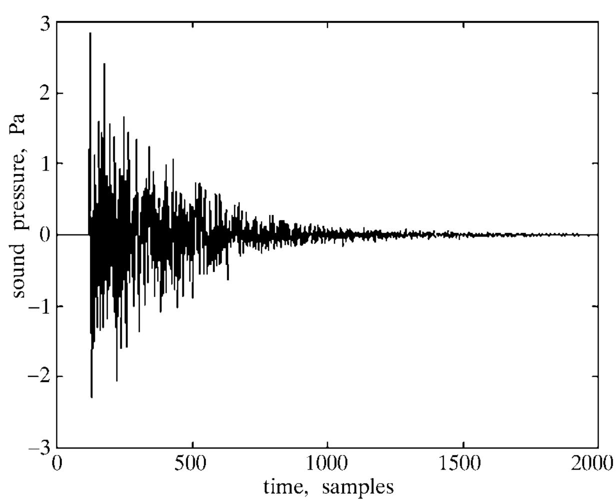

Fig. 12. Residual noise in time-domain with $H(z) = C(z) = z^{-1}$

For the secondary path transfer function is assumed:

$$H(z) = H_L(z), \quad (16)$$

with loudspeaker and duct values presented in section 5.

In ideal case it is also

$$C(z) = H(z), \quad (17)$$

so we are capable by applying these idealizations to investigate loudspeaker's transfer function influence on active noise control system performance. For Gaussian white noise as the noise signal, sampling frequency $1/T = 2$ kHz and filter length of 100 coefficients, we get residual noise spectrum presented with figure 11. As a comparison, modeling $C(z) = H(z) = z^{-1}$ gives as result almost absolute canceling of noise signal (figure 12).

7. CONCLUSION

An active noise control system in small area ducts is analyzed. Acoustically, we problem established as the active noise control in the infinite duct. Such system is consisted of acoustic paths, sensors, signal processor, amplifier and actuator (loudspeaker). The need for very accurate amplitude and phase relations between noise source signal and generated anti-noise signal, as well as the time-varying characteristics of the system elements, leads toward application of the adaptive digital filters. In order to verify performance and convergence speed of the adaptive signal processing algorithms, we presented elements of these systems, and a z-domain model of the loudspeaker mounted on the wall of a small-area duct is developed. Comparing frequency response of our models with much more sophisticated numerical calculations as well as with measurements, we achieved a very good matching. Our model allows time-domain simulations, and using this model, we analyzed influence of the loudspeaker's transfer function on FXLMS algorithm performance. A great influence of the loudspeaker on the overall system performance and on the power spectrum of the error signal is shown. Such a model allows us to analyze efficiency of particular signal processing algorithms for active noise control purposes.

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Aktivni sustavi za zaštitu od buke u cijevima. Uz dani prikaz razvoja aktivne zaštite od buke, analizirani su elementi sustava aktivne zaštite od buke u cijevima, a na temelju te analize napravljen je model elemenata. Posebno je provedena analiza zvučnika te je napravljen model zvučnika u *z*-domeni, prikladan za analizu sustava koji imaju analogne i digitalne dijelove. Primjenom takvog modela možemo u vremenskoj domeni analizirati rad i konvergenciju pojedinih adaptivnih algoritama obrade signala. Kao primjer provedena je analiza rada sustava aktivne zaštite na pojednostavljenom modelu ventilacijskog kanala primjenom FXLMS algoritma, te je pokazan jak utjecaj prijenosne karakteristike zvučnika na spektar snage zvučnog signala preostale buke.

Ključne riječi: adaptivni filtri, aktivna zaštita od buke, elektroakustički pretvornici, ventilacijski kanali, zvučnici

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