ARCHIVES OF ACOUSTICS
33, 4, 509–520 (2008)

ACTIVE NOISE CONTROL – A REVIEW OF CONTROL-RELATED PROBLEMS

Marek PAWEŁCZYK

Silesian University of Technology
Institute of Automatic Control
Akademicka 16, 44-101 Gliwice, Poland
e-mail: Marek.Pawelczyk@polsl.pl

(*received June 15, 2008; accepted October 15, 2008*)

Active noise control is a research area, where both acoustic and control related problems influence success of applications. The aim of the paper is to address the control aspects. After introducing active noise control in general, the fundamental state of the art is presented. The possible control techniques are discussed. Premises for the choice of feedforward, feedback and combined architectures are summarised. Single-channel and multi-channel systems are confronted. Benefits and drawbacks of continuous-time and discrete-time approaches are emphasised. Fixed-parameter and adaptive control systems are referenced. General control system requirements are formulated. Fundamental performance limitations are explained. Various control problem formulations including cost functions and constraints are presented for an exemplary structure.

Keywords: active noise control, optimal control, adaptive control, optimisation.

1. Introduction

In active noise control (ANC) an additional secondary sound source is used to cancel noise from the original primary source. In control system terminology primary noise constitutes an output disturbance that is to be suppressed. In fact, a residual signal as the effect of primary and secondary sounds interference at a given point in space is controlled in the mean-square or peak sense.

In a diffuse acoustic field global active noise control in an entire enclosure is practically unfeasible [1]. The solution is thus local control in a particular area or some areas and creation of the so-called local zones of quiet. Actually, the control is performed at a given point in space and the reduction propagates from this point in the form of a zone. However, it is often impossible to place an observer sensor at this point due to practical inconvenience or technological difficulty. Therefore, another sensor, called error or residual sensor, placed as close as possible to the desired point or area is used.

The error sensor feeds back information about attenuation results, which can also be used to drive the secondary source (feedback control). Sometimes it is beneficial to employ a reference sensor to detect noise upstream, long before it reaches the area of interest (feedforward control). Both techniques can also be combined to support each other. If the control algorithms are required to adapt to changes of the noise character or to variations of the plant physical properties the information from the error sensor supervises an adaptation (Fig. 1).

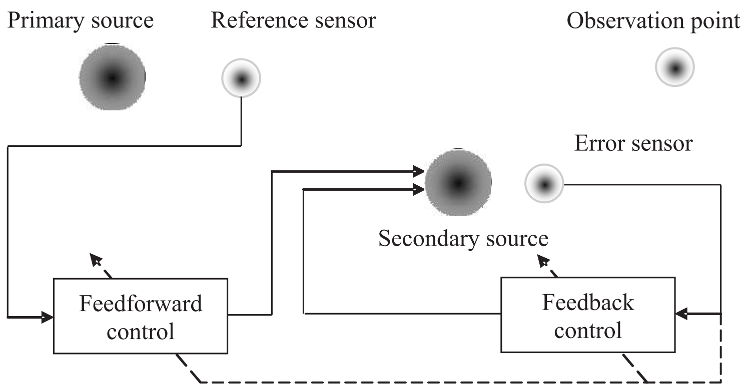

Fig. 1. Active noise control strategies.

In applications, the primary source is usually not a loudspeaker and may often be distributed. It is rather a working mechanism or engine. In turn, the secondary source is a loudspeaker or vibrating plate, and the sensors are usually microphones providing a measure of the acoustic pressure at their location. If the reference microphone in feedforward control were able to detect the secondary sound it would introduce the so-called acoustic feedback, which might deteriorate the performance or even lead to instability of the entire control system [2]. If possible, it is then suggested to substitute a tachometer, pyrometer or accelerometer for this microphone or employ a unidirectional microphone [3].

First ANC applications date back to Coanda [4], Lueg [5], and Olson and May [6]. Coanda's idea was a phase-inverted cancellation but his project was technically incorrect and therefore his work is rarely mentioned. Lueg attenuated a one-dimensional acoustic wave in a duct using feedforward from an upstream microphone. Olson and May applied feedback from a downstream microphone to attenuate ambient noise around the headrest in a seat. Although the above publications are usually referred to as the first reported works on ANC, it should be noticed that their authors took advantage of the theory of Kirchhoff diffraction developed by Rubinowicz [7], a Polish scientist. He described the effect of interference of light waves falling on a screen edge, what resulted in mutual compensation at some areas [8].

In Poland, ANC was first studied by Czarnecki [9], who determined conditions for phase compensation of sound, and it was further promoted by Engel, who initiated setting up dedicated laboratories in the AGH University of Science and Technology, Kraków, and in the Central Institute for Labour Protection, Warsaw. Soon afterwards ANC laboratories were set up in the Silesian University of Technology, Gliwice. Hitherto thousands of scientific and technical papers including books on ANC were published in the world, and hundreds of them in Poland. The first review-type publications authored by Polish scientists are [10–17].

2. Control system approaches

In ANC a feedforward architecture is of considerable interest. Then, the control system is inherently stable if the control filter is stable. There are, however, two primary practical limitations. The reference signal highly correlated with the output disturbance should be available and it should not be influenced by the control signal. Violation of the first assumption decreases the performance while not satisfying the second assumption introduces a feedback loop that can become unstable during the adaptation. The ANC systems are often subject to noises upcoming from different directions and originating from various sources. On the other hand, they are designed to have a general usage or to be used in mobile applications. Therefore, it is often assumed that the reference signal coherent with such a noise is unavailable and the best-developed feedforward control as originally suggested by Lueg cannot be employed. Thus, the idea of Olson and May is then undertaken. Sometimes it is also beneficial to employ a combined structure where feedback and feedforward supplement each other [18]. The feedforward part may provide good performance if the plant is close to nominal conditions where its model is of sufficient accuracy, and the feedback part may compensate for plant modelling errors and nonlinear effects.

The analysis of generated zones of quiet leads to conclusion that for low frequencies they are large enough to reach human ears. For higher frequencies the idea of virtual microphones, which enables to shift the zones, has been formulated (see, e.g. [19–22]). It relies on attenuating the acoustic noise at desired locations without performing measurements at these locations.

In case of fixed controllers, mainly based on the robust theory, independent systems can often control individual channels of a plant [19, 20]. However, multi-channel implementation can provide higher noise reduction and maintain stability of the overall control system [23]. In adaptive systems, due to acoustic coupling between the channels, a multi-channel approach to control is recommended for the sake of convergence and attenuation [24]. Uncompensated paths usually varying in time create additional feedback loops [25]. In a nonlinear system, which in fact any adaptive system is, the feedbacks can generate a chaotic behaviour in a long-time horizon [26]. Such behaviour is particularly evident when the adaptive system is tuned to converge fast, what is very important for practical success of many ANC applications (see also [27]).

First-generation applications were based on analogue designs. Advances in microelectronics, high-speed signal processors and filtering techniques during the 1980’s precipitated a flurry of activity in digital control systems. Digital control give high flexibility because the controllers can be of very high order and respond in a very sophisticated manner, what is particularly useful for multimodal plants. However, even nowadays analogue approach to control gives remarkable benefits if the plant delay should be reduced, because it does not require anti-aliasing and reconstruction filters of large group delays.

3. Control system requirements

A successful active noise control system should guarantee:

1. *High attenuation*. This is directly translated into acoustic comfort perceived by the user and is considered as the primary goal for most active noise control applications.
2. *Wide attenuation band*. This makes the device more universal for a variety of working environments but complicates control algorithms. Apparently, some features of the plant limit the band.
3. *Internal stability.* The control system should be internally stable to limit values of any signals.
4. *Numerical stability*. Calculations must not accumulate excessive numerical errors, even in a very long operation time, because they could make the system unstable.
5. *Convergence in case of adaptive systems*. In order to get a stable control system, parameter estimates cannot diverge. It can be proven that in case of feedback adaptive systems the problems of stability of the structural loop and convergence of the adaptive algorithm are coupled. Moreover, convergence rate should be high to avoid any annoying transient effects. This is particularly difficult if the plant response has large resonance peaks and deep valleys.
6. *High robustness to changes of disturbance and plant properties as well as disturbances like impulse noise*. Most practical applications are subject to such effects. The control system should remain stable and yield acceptable performance.

These constitute criteria that must be taken into account while designing active noise control systems and usually a reasonable trade-off between them must be chosen. Generally, it is desired that the control system responds as presented in Figs. 2 and 3, i.e. it reacts very fast to all the changes and keeps stable as well as guarantees satisfactory attenuation with low overshoots allowed.

4. Fundamental control system limitations

Noise reduction as a function of frequency can be expressed as [16]:

$$J(\omega) = -10\log_{10}\left(\frac{S_{yy}(e^{\omega T_S})}{S_{dd}(e^{\omega T_S})}\right) = -20\log_{10}\left(\left|V(e^{-j\omega T_S})\right|\right) \quad [\mathrm{dB}], \qquad (1)$$

where $|V(j\omega)| < 1$ is the sensitivity function, which maps the disturbance to the control system output, and $S_{dd}(e^{\omega T_S})$ and $S_{yy}(e^{\omega T_S})$ are power spectral densities (PSD) of the disturbance and the output, respectively. It means that $|V(j\omega)| < 1$ should be less than one for frequencies where reduction is required, and the smaller the modulus the larger the reduction. However, there are many control system performance limitations. Some of them are highlighted below.

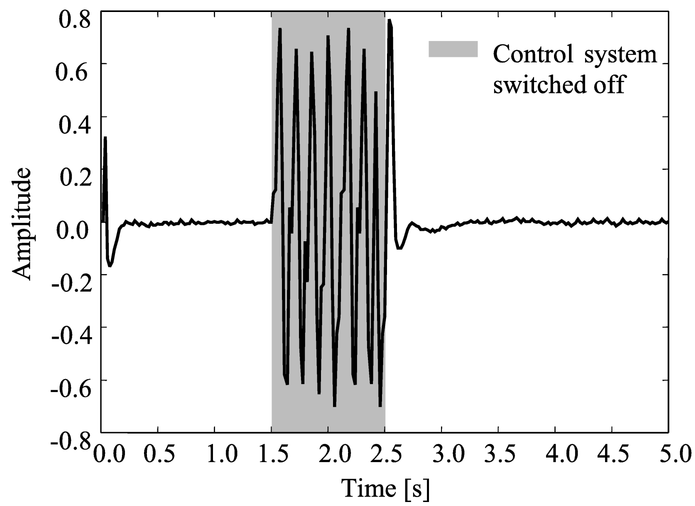

Fig. 2. Control system response to changes of plant parameters.

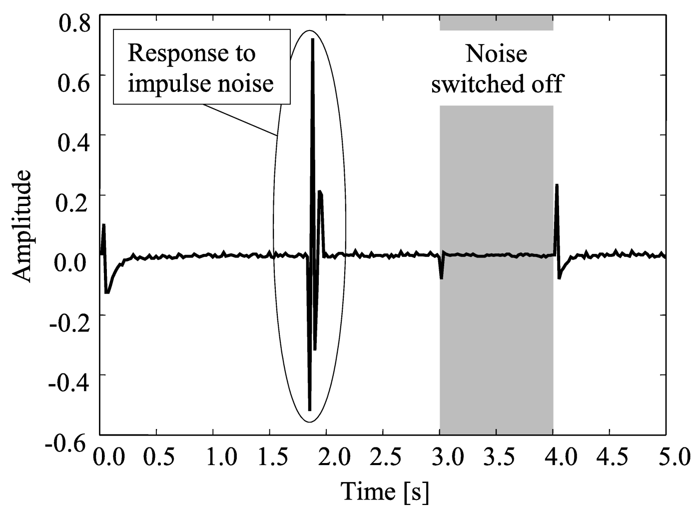

Fig. 3. Control system response to changes of the noise to be reduced.

1. Performance of any feedback control system is limited by the waterbed effect. It can be proven that if the open-loop system is a stable rational strictly proper function (in continuous time) with relative degree at least two, then provided the closed-loop system is stable, the following integral is fulfilled [16, 28]:

$$\int_0^{\infty} J(\omega)\,\mathrm{d}\omega = 0. \tag{2}$$

This means that noise reduction for some frequencies implies its amplification for other frequencies. Moreover, according to the maximum modulus principle, requiring high reduction over a range of frequencies necessarily leads to a large sensitivity peak outside that range. Thus, if the disturbance has spectral components at those frequencies, they will be meaningfully amplified. An important difference between the integral in the continuous-time and discrete-time cases is that the latter involves restrictions over a finite interval.

2. Active noise control plants are non-minimum phase, including delay. This makes complete wideband noise reduction impossible, because for that purpose the controller transfer function should be the inverse of a plant model over those frequencies. If disturbance reduction is required throughout a frequency interval, for which the non-minimum phase zero contributes significant phase lag then the disturbance is greatly amplified at some higher frequencies.

3. The plants usually exhibit strong nonlinear effects at low frequencies and are only approximated there by linear models. Such effects significantly reduce the performance over that range. Moreover, nonlinear elements with saturation characteristics are present both at the input and output of the plants, because the D/A converter (A/D onverters) cannot deliver (accept) voltages outside some bound. Other, not mentioned here, nonlinear phenomena can also be present in active control [29].

4. For discrete-time systems inter-sample effects associated with continuous-time signal sampling and reconstruction are present. To avoid them high-order low-pass analogue filters of large group delays are usually used, what degrades the performance or even make feedforward control unjustified. The filters may be avoided by applying nonuniform signal sampling and over-sampling techniques [30]. However, such approach is a source of other problems.

5. Frequency response of many acousto-electric plants exhibits large peaks and deep valleys. Parametric modelling is then very complicated and usually involves a large number of parameters, incomparable to classical plants met in control system design.

6. Frequently, noise reduction is required at an area different than that, where a sensor monitoring control results is present. Moreover, both the plant response and properties of the disturbance may significantly vary in time. In such cases models of sufficient accuracy are necessary and they should also be efficiently updated during operation of a number of control algorithms.

7. Limited speed of the signal processor employed may require simplification of control algorithms and thus reduce the performance.

5. Control problem formulations

In this section different approaches to optimal control are presented. The considerations are carried out in a reference to Internal Model Control (IMC) structure (see Fig. 4). It is in fact a feedback structure with output disturbance being estimated based on a plant model. However, such approach allows for using well developed feedforward techniques for design and analysis. Moreover, feedforward structure with acoustic feedback may be considered as the IMC structure as well. The analysis can also be converted to any control system structure. In the figure P represents the real plant, $\hat{P}$ is its model, and W is the control filter. The most frequently used cost function for active noise control is expressed using the H_2 norm ($\|.\|_2$) as follows [31]:

$$L = E\{y^2(i)\} \equiv \left\| V(e^{-j\omega T_S})\sqrt{S_{dd}(e^{\omega T_S})} \right\|_2^2 \equiv \frac{1}{2\pi}\int_0^{2\pi} S_{yy}(e^{\omega T_S})\mathrm{d}(\omega T_S), \quad (3)$$

where E stands for the expectation operator, ω is the angular frequency, and T_S is the sampling interval. Minimisation of such cost function corresponds to minimisation of the sound pressure level at the real microphone.

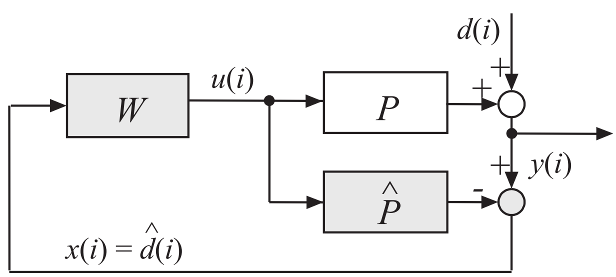

Fig. 4. Internal Model Control system structure.

The controller can be designed in the time domain, frequency domain, and transform domain [25]. Detailed analysis of the optimal control system designed based on the plant model leads to a conclusion that variations in the plant or disturbance degrade the performance, in general. However, there are also some circumstances, where modelling errors may enhance the performance what corresponds to shifting the zone of quiet to a location different than the assumed one [22].

In case of plant parameter variations and modelling errors the plant response can be expressed in the form of multiplicative plant uncertainty [25]:

$$P(e^{-j\omega T_S}) = \widehat{P}(e^{-j\omega T_S})\left[1 + \delta P(e^{-j\omega T_S})\right], \qquad \underset{\omega T_S}{\forall}\ \overline{\delta P}(e^{\omega T_S}) \geq \left|\delta P(e^{-j\omega T_S})\right|, \quad (4)$$

where $\overline{\delta P}(e^{\omega T_S})$ is the upper bound of the uncertainty. Such description is rather conservative since it assumes that for any angular frequency, ω_0, responses of all plants fall within a disc with centre at $P_o(e^{-j\omega_0 T_S})$ and radius $\overline{\delta P}(e^{\omega T_S})\left|P_o(e^{-j\omega T_S})\right|$. Then, the necessary and sufficient condition for robust stability is ($\|.\|_\infty$ stands for the H_∞ norm)

$$\left\|\widehat{P}(e^{-j\omega T_S})\overline{\delta P}(e^{\omega T_S})W(e^{-j\omega T_S})\right\|_\infty < 1. \tag{5}$$

Based on the above considerations the following optimisation problems can be formulated for IMC [22, 25, 31–33]:

1. H_2 control problem with robust stability constraint:

$$\min_W \left\|\left(1+\widehat{P}(e^{-j\omega T_S})W(e^{-j\omega T_S})\right)\sqrt{S_{dd}(e^{\omega T_S})}\right\|_2^2. \tag{6}$$

The robust stability constraint is given by (5).

2. H_∞ control problem with robust stability constraint:

$$\min_W \left\|\left(1+\widehat{P}(e^{-j\omega T_S})W(e^{-j\omega T_S})\right)\sqrt{S_{dd}(e^{\omega T_S})}\right\|_\infty. \tag{7}$$

The robust stability constraint is given by (5).

3. H_∞ control problem with robust performance:

$$\min_W \Big\|\left|\left(1+\widehat{P}(e^{-j\omega T_S})W(e^{-j\omega T_S})\right)\right|\sqrt{S_{dd}(e^{\omega T_S})} + \left|\widehat{P}(e^{-j\omega T_S})W(e^{-j\omega T_S})\right|\overline{\delta P}(e^{\omega T_S})\Big\|_\infty. \tag{8}$$

The robust stability constraint is given by (5).

4. Modified H_2 control problem:

$$\min_W \left\{\left\|\left(1+\widehat{P}(e^{-j\omega T_S})W(e^{-j\omega T_S})\right)\sqrt{S_{dd}(e^{\omega T_S})}\right\|_2^2 + \beta\left\|\widehat{P}(e^{-j\omega T_S})W(e^{-j\omega T_S})\overline{\delta P}(e^{\omega T_S})\right\|_2^2\right\}. \tag{9}$$

5. H_2 control problem with filter parameters weighting:

$$\min_W \left\{\left\|\left(1+\widehat{P}(e^{-j\omega T_S})W(e^{-j\omega T_S})\right)\sqrt{S_{dd}(e^{\omega T_S})}\right\|_2^2 + \beta\left\|W(e^{-j\omega T_S})\right\|_2^2\right\}. \tag{10}$$

6. H_2 control problem with control signal weighting:

$$\min_W \left\{\left\|\left(1+\widehat{P}(e^{-j\omega T_S})W(e^{-j\omega T_S})\right)\sqrt{S_{dd}(e^{\omega T_S})}\right\|_2^2 + \beta\left\|W(e^{-j\omega T_S})\sqrt{S_{dd}(e^{\omega T_S})}\right\|_2^2\right\}. \tag{11}$$

The above definitions can be supplemented by a constraint protecting against reinforcing sound of more than defined by the function $J^{-1}(e^{\omega T_S})$:

$$\left\| \left(1 - \widehat{P}(e^{-j\omega T_S}) W(e^{-j\omega T_S})\right) J(e^{\omega T_S}) \right\|_\infty < 1. \tag{12}$$

Another constraint, particularly useful in case of 1–3 definitions is to limit the power supplied to the loudspeaker less than M^{-1}:

$$\left\| W(e^{-j\omega T_S}) \sqrt{M S_{dd}(e^{\omega T_S})} \right\|_2^2 < 1. \tag{13}$$

Cost functions defined by 4–6 implicitly take into account constraints related to control system stability and power of the control signal. In a more general case the term $S_{dd}^{-1/2}(e^{\omega T_S})$ may denote the upper bound of the sensitivity function. Then, requiring the H_2 or H_∞ norm in the cost functions to be less than one allows for reducing both narrowband and wideband acoustic noise. However, such general statement may limit noise reduction compared to the case where noise PSD is explicitly taken into account. A much less conservative approach, allowing to obtain a better performance and reduce the controller order is to formulate the problem as follows [1]:

7. Nonlinear problem:

$$\min_W \int\limits_{\omega_{\min}}^{\omega_{\max}} 20 \log_{10} \left| 1 + \widehat{P}(e^{-j\omega T_S}) W(e^{-j\omega T_S}) \right| \mathrm{d}(\omega T_S). \tag{14}$$

A constrained related to the gain margin of χ dB is given as:

$$20 \log_{10} \left| \widehat{P}(e^{-j\omega_\pi T_S}) W(e^{-j\omega_\pi T_S}) \right| \leq -\chi,\ \varphi(\omega_\pi) = -\pi + 2k\pi. \tag{15}$$

The constraint (12) protecting against excessive noise reinforcement has a special meaning due to minimisation of an integral of the reduction curve over required frequency band for the nominal plant. The drawback of this approach is a strong nonlinearity and possibility to find a local solution.

Most of the optimisation problems can be solved by discretising frequency responses and using relevant methods, e.g. Sequential Quadratic Programming (SQP). Intelligent control algorithms based on neural networks, evolutionary approach, and fuzzy sets can also be used for active control [34, 35].

6. Conclusions

It the paper control-related problems met when designing and implementing active noise control systems have been reviewed. The benefits of using feedforward control have been mentioned. However, it has also been stressed that feedback is a more general approach. Their combination into one system allows taking advantage of both of them and is advised if the system set up hardware efficiency permits. It has been noticed that although active control systems are usually performed in discrete-time, the continuous-time approach gives remarkable benefits if the plant delay should be reduced. Virtual-microphone control systems capable to shift the zones of quiet to desired locations have been addressed. They are useful if it is impossible to place a physical sensor at the areas where noise reduction is required and the zones of quiet are too small to reach those areas. Multi-channel control has been appreciated in order to extend the zones of quiet and avoid uncompensated loops, which could make the system unstable or adaptive algorithms divergent. Drawbacks of such systems, like complexity and high hardware requirements have been pointed out. Control system requirements and limitations have been clearly analysed. Optimisation problems for fixed-parameter control system design have been formulated by defining different cost functions and constraints using both H_2 and H_∞ theory. They generally aim at providing satisfactory noise reduction and guaranteeing stability of the overall system. Plant parameter changes, disturbance non-stationarity, out-of-band behaviour and power supplied are also considered. Properties and advantages of adaptive control systems have been discussed and referenced to fix-parameter systems. Adaptive systems allow responding to the plant parameter changes and disturbance non-stationarities, although they may suffer from algorithm divergence and unpleasant sound effects. They need particular care if applied to multi-channel or feedback systems, where problems related to stability of the structural feedback loop and convergence of the adaptive algorithms overlap each other.

Activity of Polish scientists have been stressed in the paper. They have contributed to active noise control from its early stage [7–17], to current findings. Their recent books are: [22, 30, 36–38].

Acknowledgments

Financial support from the state budget for science in Poland for 2008 is gratefully acknowledged. The author is the prize winner of the SIEMENS Scientific Award for 2007.

References

[1] NELSON P.A., ELLIOTT S.J., *Active Control of Sound*, Academic Press, Cambridge 1994.

[2] KUO S.M., MORGAN D.R., *Active Noise Control Systems. Algorithms and DSP Implementations*, J. Wiley & Sons, New Jersey 1996.

[3] TOKHI M.O., LEITCH R.R., *Active Noise Control*, Clarendon Press, Oxford 1992.

[4] COANDA H., *Procédé de protection contre les bruits* [in French], French Patent, FR 722.274, 1930.

[5] LUEG P., *Process of silencing sound oscillations*, U.S. Patent, No 2,043,416, 1936.

[6] OLSON H.F., MAY E.G., *Electronic sound absorber*. Journal of the Acoustical Society of America, **25**, 1130–1136 (1953).

[7] RUBINOWICZ A., *Die Beugungswelle in der Kirchoffschen Theorie der Beugungserscheinungen* [in German], Ann. Phys., **53**, 257–278 (1917).

[8] JESSEL M., *Some evidences for a general theory of active sound absorption*, Proc. Inter Noise, 1979.

[9] CZARNECKI S., *Investigations of sound transmission properties of a medium resulting from sound wave compensation caused by other sources*, Journal of Sound and Vibration, **11**, 225–233 (1970).

[10] ENGEL Z., *Aktywna redukcja drgań i hałasu* [in Polish], Proc. XI Symp. Drgania w Układach Fizycznych, Poznań, Poland, 1984, 124–125.

[11] ENGEL Z., *Aktywne metody redukcji drgań i hałasu (rys historyczny, zadania, problemy)* [in Polish], Proc. I Szkoła Metody Aktywne Redukcji Drgań i Hałasu, Rabka-Kraków, 1993, 5–18.

[12] ENGEL Z., KOWAL J., *Sterowanie Procesami Wibroakustycznymi* [in Polish], Univesity of Mining and Metalurgy Press, Kraków 1995.

[13] ZAWIESKA W.M., *Analiza i Synteza Układu Aktywnej Redukcji Hałasu* [in Polish], Ph.D. Dissertation, University of Mining and Metalurgy, Kraków 1991.

[14] MAKAREWICZ G.J., *Problemy stabilności w układach aktywnej redukcji dźwięku* [in Polish], Ph.D. Dissertation, University of Mining and Metalurgy, Kraków 1993.

[15] OGONOWSKI Z., *Adaptive noise control using direct method*, Mat. XXII Zimowa Szkoła Zwalczania Zagrożeń Wibroakustycznych, Wisła, Poland, 1994, 65–66.

[16] PAWEŁCZYK M., *Active Noise Control for Compact Acoustic Plants*, JSCS, Gliwice 1999.

[17] BISMOR D., *Adaptive Algorithms for Active Noise Control in an Acoustic Duct*, JSCS, Gliwice 1999.

[18] TAMMI K., *Identification and Active Feedback-Feedforward Control of Rotor*, International Journal of Acoustics and Vibration, **12**, 1, 7–14 (2007).

[19] RAFAELY B., ELLIOTT S.J., GARCIA–BONITO J., *Broadband performance of an active headrests*, Journal of the Acoustical Society of America, **106**, 2, 787–793 (1999).

[20] TSENG W.K., RAFAELY B., ELLIOTT S.J., *Performance limits and real-time implementation of a virtual microphone active headrest*, Proc. ACTIVE'02, Southampton, UK, 2002, 1231–1242.

[21] KESTELL C., CAZZOLATO B., HANSEN C., *Active noise control in a free field with a virtual microphone and a virtual energy density sensor*, J. Acoustical Society of America, **48**, 4, 475–483 (1999).

[22] PAWEŁCZYK M., *Feedback Control of Acoustic Noise at Desired Locations*, Habilitation Dissertation, Gliwice, Silesian University of Technology Press, No. 1684, (141), (2005).

[23] PAWEŁCZYK M., *Adaptive noise control algorithms for active headrest system*, Control Engineering Practice, **12**, 9, 1101–1112 (2004).

[24] PAWEŁCZYK M., *Multiple input-multiple output adaptive feedback control strategies for the active headrest system: design and real-time implementation*, International Journal of Adaptive Control and Signal Processing, **17**, 10, 785–800 (2003).

[25] ELLIOTT S.J., *Signal Processing for Active Control*, Academic Press, London 2001.

[26] FIGWER J. , BLAŻEJ M., *Chaos in active noise control*, Proc. 10th Int. Congress on Sound and Vibration, Stockholm, Sweden, 2003, 203–209.

[27] SIMON A., FLOWERS G.T., *Adaptive Disturbance Rejection and Stabilisation for Rotor Systems with Internal Damping*, International Journal of Acoustics and Vibration, **13**, 2, 73–81 (2008).

[28] SERON M.M., BRASLAVSKY J.H., GOODWIN G.C., *Fundamental Limitations in Filtering and Control*, Springer-Verlag, London 1997.

[29] CHEN L., HANSEN C.H., HE F., SAMMUT K., *Active Nonlinear Vibration Absorber Design for Flexible Structures*, International Journal of Acoustics and Vibration, **12**, 2, 51–59 (2007).

[30] CZYŻ K., *Active Noise Control Systems with Nonuniform Signal Sampling*, Ph.D. Dissertation, Gliwice, Institute of Automatic Control, Silesian University of Technology, 2006.

[31] RAFAELY B., *Feedback Control of Sound*, Ph.D. Thesis, University of Southampton, 1997.

[32] PAWEŁCZYK M., *Optimal active noise control for time-varying plants* [in Polish], Mat. XXXIV Zimowej Szkoły Zwalczania Zagrożeń Wibroakustycznych, Ustroń 2006.

[33] AKESSON H., SMIRNOVA T., CLAESSON I., HAKANSSON L., *On the Development of a Simple and Robust Active Control System for Boring Bar Vibration in Industry*, International Journal of Acoustics and Vibration, **12**, 4, 139–152 (2007).

[34] SIMON A., FLOWERS G.T., *Non-singleton Fuzzy Sets for Disturbance Attenuation*, International Journal of Acoustics and Vibration, **12**, 4, 171–178 (2007).

[35] ALAM M.S., TOKHI M.O., *Design of Command Shaper using Gain-delay Units and Particle Swarm Optimisation Algorithm for Vibration Control of Flexible Systems*, International Journal of Acoustics and Vibration, **12**, 3, 99–108 (2007).

[36] MICHALCZYK M.I., *Adaptive control algorithms for three-dimensional zones of quiet*, JSCS, Gliwice 2004

[37] ENGEL Z., MAKAREWICZ G., ZAWIESKA W.M., MORZYŃSKI L., *Metody Aktywne Redukcji Hałasu* [in Polish], CIOP Press, Warszawa 2001.

[38] ZAWIESKA W.M., *Wybrane zagadnienia aktywnej redukcji hałasu na przykładzie transformatorów* [in Polish], CIOP Press, Warszawa 2007.