24th INTERNATIONAL CONGRESS ON SOUND AND VIBRATION
23–27 July 2017, London

STUDY ON LOCAL ACTIVE NOISE CONTROL WITHIN A SMALL SEMI ENCLOSED SPACE

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Previous studies on local active noise control usually focused on the cases where the control sound field could be viewed as a fully free field. This paper carried out a study on local active noise control within a small semi-enclosed space where both the direct sound and the reflected sound of the control source exist. A numerical model is established to calculate the control sound field created in such a space using the boundary element method, and a theoretical model is developed to predict the spatial extents of the quiet zones. Results showed that the control sound field created in a small semi-enclosed space is more like a partially diffuse sound field than a fully free or fully diffuse sound field, and the diffuse sound of the control source may help for local ANC to create quiet zones with more profitable spatial extension. A simple experiment is carried out and verifies the results.

Keywords: ANC, semi-enclosed space, boundary element method, quiet zones.

1. Introduction

Active noise control (ANC) has been studied extensively in the past three decades [1]. It can be achieved either globally, with an overall noise reduction in the sound field, or locally, with quiet zones created at desired locations. In many cases, local ANC is more feasible due to the complexity of the sound field at high frequencies [2]. It was demonstrated that a spherical quiet zone of 1/10 wavelength spatial dimension and with 10 dB noise reduction could be achieved at the desired location in a pure tone diffuse sound field [3].

A typical application of local ANC is to attenuate the noise near the ears of a passenger seated in a chair, using an active headrest system. The idea was first proposed in the 1950s [4], and then came into application in the 1990s [5-7]. A laboratory local ANC system with virtual microphone techniques to control broadband noise was constructed by Rafaely et al. [8-9]. The system, implemented with single-channel feedback controllers, could produce 9.5 dB overall noise reductions at both ears when the head was kept still at the central position of the seat. Later, local ANC systems with adaptive multi-channel controllers were set up and studied [10-14]. Most of these studies considered applications located in a large enclosure, where the primary sound field could be assumed to be a perfect diffuse field and the control sound fields could be viewed as free fields with decaying characteristics. Local ANC systems set up in other environments, especially in semi-enclosed spaces, which are now common in many practical applications, however, are less studied.

In this paper, the performance of a local ANC system in a semi-enclosed environment is studied. A numerical model is established to calculate the sound field created in a semi-enclosed space by using the boundary element method (BEM). The BEM is an effective numerical technique for Helmholtz acoustic simulations [15]. A theoretical model is also developed to predict the spatial extents of the quiet zones. It is assumed that in the semi-enclosed space, the primary sound field could be treated as a perfect diffuse field and the control sound field, containing both the direct sound and plenty of reflection sound, is partially diffused. The 10 dB quiet zones created in different sound fields are numerically calculated according to the theoretical model, and a simple experiment is carried out to compare the performance of local ANC in a small semi-enclosed booth with that in a large enclosure.

The paper is organized as follows. In Section 2, a numerical model is established to calculate the sound field created in semi-enclosed booth by using BEM. In Section 3, a theoretical model is developed to predict the extents of the quiet zones created in a partially diffuse control sound field, and the contours of the 10 dB quiet zones are numerically calculated and compared. Section 4 introduces a simple experiment of local ANC in a small semi-closed booth, and compares it with a conventional counterpart. Finally, conclusions are drawn in the last section.

2. Numerical Calculation of Control Field in a Semi-enclosed Space

In local ANC applications, the control sources are usually placed near the head of the passenger to cancel the noise at the ears. Figure 1 presents the geometry of a numerical model for local ANC in a semi-enclosed booth. In the model, the booth is set to bee 800mm wide, 400mm deep and 1000 mm high, with the front side open large. A sphere with the diameter of 150mm is placed at the central position of the booth to represent the head of a passenger. A monopole sound source is placed at one of the wainscots of the booth, denoting the control source.

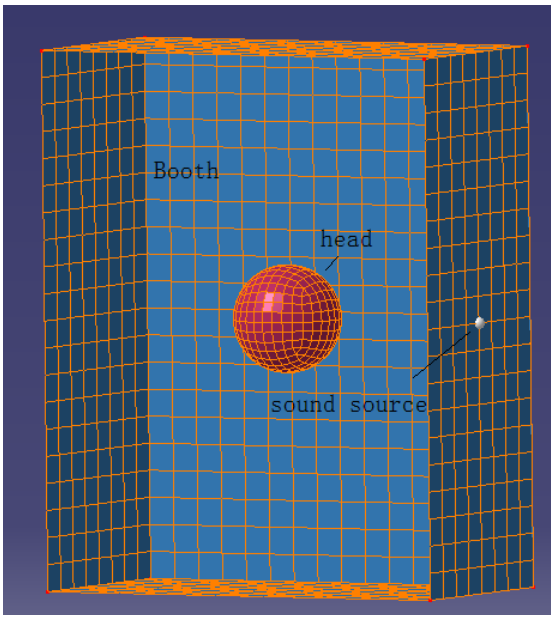

Figure 1. Geometry of a local ANC system in a semi-enclosed booth.

The BEM is used to calculate the sound field created in the booth. By considering a boundary (Γ) of a domain (Ω), the problem is solved in terms of the pressure $p(\mathbf{x})$ and the particle velocity $q(\mathbf{x})$. The boundary integral equation for Helmholtz problem can be written as

$$C(\mathbf{x}`)p^{j}(\mathbf{x}`)+\int_{\Gamma}q^{*}(\mathbf{x}`,\mathbf{x})p(\mathbf{x})d\Gamma(\mathbf{x})=\int_{\Gamma}p^{*}(\mathbf{x}`,\mathbf{x})q(\mathbf{x})d\Gamma(\mathbf{x})+\int_{\Omega}p^{*}(\mathbf{x}`,\mathbf{x}^{s})\frac{1}{c^{2}}b(\mathbf{x}^{s})d\Omega(\mathbf{x}^{s}) \quad (1)$$

where $p^{*}(\mathbf{x}`,\mathbf{x})$ and $q^{*}(\mathbf{x}`,\mathbf{x})$ are the pressure and particle velocity fundamental solutions, respectively, and C($\mathbf{x}`$ ) depends on the location of the point $\mathbf{x}`$, and the last term refers to the presence of the sources within the domain Ω with the strength $b(\mathbf{x}^{s})/c^{2}$ and c is the sound velocity.

Numerical calculation results are shown in Figure 2. It is shown that the sound field created in the booth lagged with perfect sound absorption materials, which makes it similar with that located directly in a large open room, is more like a free sound field with decaying characteristics. The sound field created in the booth lagged with normal sound absorption materials (in this paper, the absorption coefficients of the materials are set to be 0.01), however, is more like a partially diffuse field where both the direct sound and the diffuse sound both exist rather than a fully diffuse or a fully free sound field.

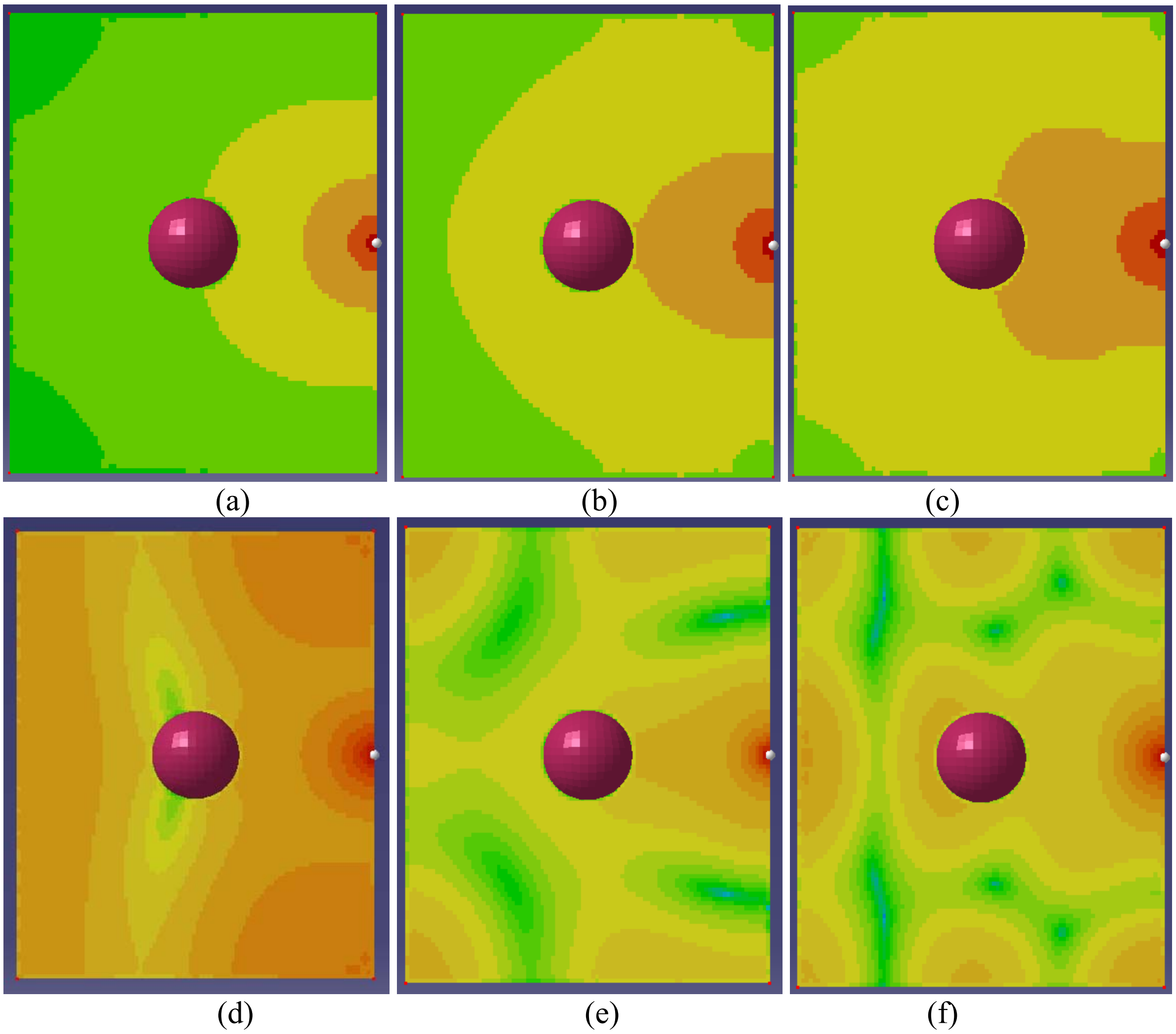

Figure 2. Sound fields created in a semi-enclosed booth lagged with different absorption materials. (a)/(b)/(c) for sound field in a booth lagged with perfect absorption materials with the excitation frequency of 300 Hz /450 Hz /600Hz; (d)/(e)/(f) for sound field in a booth lagged with normal absorption materials with the excitation frequency of 300 Hz /450 Hz /600Hz.

3. A Theoretical Model to Predict the Quiet Zones

From Section 2 we can see that the interior sound field created in a semi-enclosed space is more like a partially diffuse field, where both the direct sound and the diffuse sound exist.

Assuming that ANC is carried out in a semi-enclosed space with a partially diffuse control sound field and the primary sound field is perfectly diffuse, as shown in Figure 3. Consequently, the total sound pressure $p(\mathbf{r},t)$ at $\mathbf{r}$, a calculation point near the cancellation point $\mathbf{r}_0$, can be expressed as

$$p(\mathbf{r},t) = p_p(\mathbf{r},t) + p_s(\mathbf{r},t) \tag{2}$$

where $p_p(\mathbf{r},t)$ is the pressure due to the primary source, and $p_s(\mathbf{r},t)$ is the pressure due to the control source. $p_s(\mathbf{r},t)$ can be divided into two parts:

$$p_s(\mathbf{r},t) = p_{s1}(\mathbf{r},t) + p_{s2}(\mathbf{r},t) \tag{3}$$

where $p_{s1}(\mathbf{r},t)$ is the direct sound from the control source, and $p_{s2}(\mathbf{r},t)$ is the diffuse sound. The two parts are assumed to be uncorrelated with each other.

If the noise $p(\mathbf{r}_0,t)$ at the cancellation point $\mathbf{r}_0=(r_0,\theta,\phi)$ can be completely cancelled, so that

$$\begin{aligned} p(\mathbf{r}_0,t) &= p_p(\mathbf{r}_0,t)+p_{s1}(\mathbf{r}_0,t)+p_{s2}(\mathbf{r}_0,t) \\ &= 0 \end{aligned}, \tag{4}$$

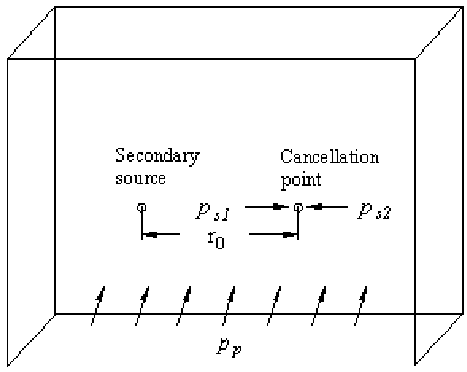

Figure 3. ANC in a semi-enclosed space with a partially diffuse control sound field.

then, the space-average mean square of the primary pressure p $p_p(\mathbf{r}_0,t)$ is given by

$$\begin{aligned} \langle |p_p(\mathbf{r}_0,t)|^2\rangle &= \langle |p_{s1}(\mathbf{r}_0,t)|^2\rangle + \langle |p_{s2}(\mathbf{r}_0,t)|^2\rangle \\ &= \langle |p_p(\mathbf{r}_1,t)|^2\rangle \end{aligned} \tag{5}$$

where $\langle\cdot\rangle$ denotes space averaging over different positions [16]. The space-averaged mean square pressure of $p(\mathbf{r},t)$ at $\mathbf{r}$ can be written as

$$\begin{aligned} \langle |p(\mathbf{r},t)|^2\rangle &= \langle |p_p(\mathbf{r},t)|^2\rangle + \langle |p_{s1}(\mathbf{r},t)|^2\rangle + \langle |p_{s2}(\mathbf{r},t)|^2\rangle \\ &+ 2\langle p_p^*(\mathbf{r},t)p_{s1}(\mathbf{r},t)\rangle + 2\langle p_p^*(\mathbf{r},t)p_{s2}(\mathbf{r},t)\rangle \\ &+ 2\langle p_{s1}^*(\mathbf{r},t)p_{s2}(\mathbf{r},t)\rangle \end{aligned} \tag{6}$$

where * denotes the complex conjugate. Take ANC in a pure tone sound field for example, the direct sound $p_{s1}(\mathbf{r},t)$ can be expressed as

$$p_{s1}(\mathbf{r},t) = f(\mathbf{r}_0,\mathbf{r})p_{s1}(\mathbf{r}_0,t)e^{-j\omega\Delta t} \tag{7}$$

where ω is the angular frequency of the disturbance source; $f(\mathbf{r}_0, \mathbf{r})$ is the spatial correlation coefficient between the sound pressure at $\mathbf{r}_0$ and $\mathbf{r}$ in a free field; Δt denotes the time lag given by $\Delta t=\Delta r/c$, $\Delta r=\left\| |\mathbf{r}|-|\mathbf{r}_0| \right\|$; and c is the sound speed. The diffuse sound $p_{s2}(\mathbf{r},t)$ in Eq. (6) can be expressed as [3]

$$p_{s2}(\mathbf{r},t) = \rho(\Delta\mathbf{r})p_{s2}(\mathbf{r}_0,t)+p_{su}(\mathbf{r},t) \tag{8}$$

where $p_{su}(\mathbf{r},t)$ is the component uncorrelated with $p_{s2}(\mathbf{r}_0,t)$, and $\rho(\Delta\mathbf{r})$ is the correlation coefficient between the sound pressure at $\mathbf{r}_0$ and $\mathbf{r}$ in a diffuse field, and

$$\begin{aligned} \rho(\Delta\mathbf{r}) &= \mathrm{sinc}(k\Delta\mathbf{r}) \\ &= \sin(k\Delta\mathbf{r})/(k\Delta\mathbf{r}) \end{aligned} \tag{9}$$

where $\Delta\mathbf{r}=|\mathbf{r}-\mathbf{r}_0|$, k is the wavenumber [17]. Similarly, $p_p(\mathbf{r}_1,t)$ can be written as

$$p_p(\mathbf{r},t) = \rho(\Delta\mathbf{r})p_p(\mathbf{r}_0,t)+p_{pu}(\mathbf{r},t) \tag{10}$$

where $p_{pu}(\mathbf{r}_1,t)$ is perfectly uncorrelated with $p_p(\mathbf{r}_0,t)$. Substituting Eqs. (7) and (8) into (6) yields

$$\begin{aligned} \langle |p(\mathbf{r},t)|^2\rangle &= (f^2(\mathbf{r}_0,\mathbf{r})-2f(\mathbf{r}_0,\mathbf{r})\rho(\Delta\mathbf{r},\Delta t))\langle |p_{s1}(\mathbf{r}_0,t)|^2\rangle \\ &+ \langle |p_p(\mathbf{r},t)|^2\rangle + (1-2\rho(\Delta\mathbf{r}))\langle |p_{s2}(\mathbf{r}_0,t)|^2\rangle \end{aligned} \tag{11}$$

where $\rho(\Delta\mathbf{r}, \Delta t)$ is given by [8]

$$\begin{aligned} \rho(\Delta\mathbf{r},\Delta t) &= \rho(\Delta\mathbf{r})e^{-j\omega\Delta t} \\ &= \mathrm{sinc}(k\Delta\mathbf{r})e^{-j\omega\Delta t} \end{aligned} \tag{12}$$

In a control sound field produced by a monopole source, $f(\mathbf{r}_0, \mathbf{r}_1)=r_c/r_d$, where r_c is the distance between the cancellation point $\mathbf{r}_0$ and the location of the control source, and r_d is the distance between the calculation point **r** and the location of the control source. Substituting Eqs. (5) and (10) into (11), the noise reduction (*NR*) is given by

$$NR(\omega)=\langle|p(\mathbf{r},t)|^2\rangle/\langle|p_p(\mathbf{r},t)|^2\rangle = \frac{\left(1-2\rho(\Delta\mathbf{r},\Delta t)r_c/r_d+r_c^2/r_d^2\right)A+\left(2-2\rho^2(\Delta\mathbf{r})\right)}{1+A} \quad (13)$$

where $A=\langle|p_{s1}(\mathbf{r}_0,t)|^2\rangle/\langle|p_{s2}(\mathbf{r}_0,t)|^2\rangle$ is the direct-to-diffuse energy ratio (DDR) of the control sound field at $\mathbf{r}_0$.

Eq. (13) calculates the noise attenuation and predicts the quiet zones produced in a control field where both the direct sound and the diffuse sound exist. It has also included the formulations developed in reference [8]. For a large DDR indicating that only a little diffuse sound exists in the control field, Eq. (13) degenerates to

$$NR=1-2\rho(\Delta\mathbf{r},\Delta t)r_c/r_d+r_c^2/r_d^2 \quad (14)$$

Eq. (14) is the same as the noise attenuation formulation developed for the near filed of the control source where the direct sound dominates. For a small DDR, Eq. (12) degenerates to

$$NR=2-2\rho^2(\Delta\mathbf{r}) \quad (15)$$

Eq. (15) is similar to the noise attenuation formulation developed for the far-field of the control source where the diffuse sound dominates.

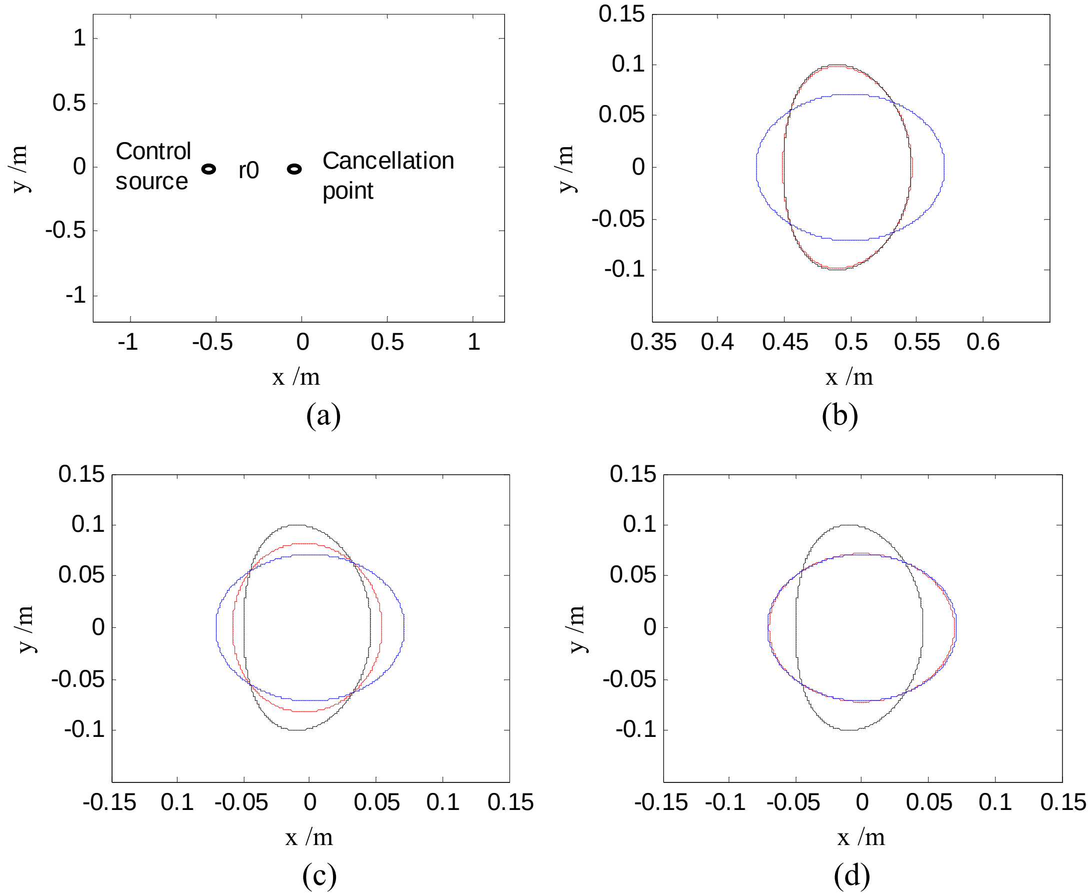

Figure 4. Quiet zones produced in control sound fields with different DDRs at the cancellation point. (a) Locations of the control source and the cancellation point (r_0=0.5 *m*); (b)/(c)/(d) Quiet zones produced in the control sound field with a high/middle/low DDR at the cancellation point. (Red lines for near-field quiet zones; black lines with circles for partially diffuse field quiet zones; green lines for far-field quiet zones.)

As for a monopole sound source with broadband spectrum nature, the noise reduction (NR) expression in Eq. (12) could be extended to be

$$NR = \int_{\omega_1}^{\omega_2} \langle |p(\mathbf{r},t)|^2 \rangle d\omega \Big/ \int_{\omega_1}^{\omega_2} \langle |p_p(\mathbf{r},t)|^2 \rangle d\omega$$
$$= \frac{\int_{\omega_1}^{\omega_2} \left(\left(1 - 2\rho(\Delta\mathbf{r},\Delta t,\omega) r_c / r_d + r_c^2 / r_d^2 \right) + \left(2 - 2\rho^2(\Delta\mathbf{r},\omega)\right) A(\omega) \right) S(\omega) d\omega}{\int_{\omega_1}^{\omega_2} \left(1 + A(\omega)\right) S(\omega) d\omega} \tag{16}$$

where $S(\omega)$ denotes the power spectral density of the signal, and the parameters ρ and A are frequency dependent for the signal with broadband spectrum nature.

Figure 4 presents the contours of the 10 dB quiet zones created in control sound fields with different DDRs at the cancellation point. The primary sound field is a perfect pure tone diffuse sound field with the excitation frequency of 300 Hz, and the control sound field is created by a monopole sound source located 0.5 meters away from the cancellation point, as shown in Figure 4(a). The near-field contours and the far-field contours are created according to the formulations developed by Rafaely [8], while the partially diffuse field contours are created according to Eq. (13). Figure 4(b) shows that at the cancellation point in the control sound field with a large DDR of about 24.0, the quiet zone nearly coincides with that in the near-field of the control source. Figure 4(c) shows that in the partially diffuse field with a medium DDR of 1.0, the quiet zone is larger than the near-field quiet zone, but smaller than the far-field quiet zone. Figure 4(d) shows that in the partially diffuse field with a small DDR of 0.04, the quite zone nearly coincides with that created in the far-field. The results suggest that the diffuse sound of the control source might help to create quiet zones with more profitable spatial extensions.

4. A Simple Experiment

A simple local ANC experiment is carried out in a large room to compare the performance of local ANC in a semi-enclosed booth with that directly in the large room.

In the experiments, the error sensor is placed 30 centimetres laterally away from the control source. The sound field is excited with a pure tone noise with the frequency of 300 Hz.

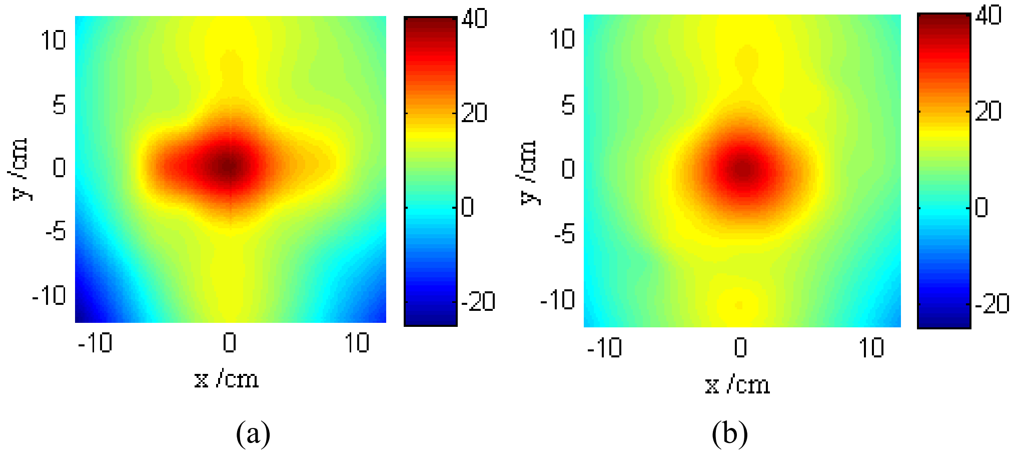

Figure 5. Noise reduction achieved in different control sound fields. (a) Noise reduction achieved in a semi-enclosed booth;(b) Noise reduction achieved in large room.

Noise reduction achieved at different locations near the error sensor position is presents in Figure 5. It is shown that the on axial line spatial extent of the quiet zone created in the booth is larger than that directly created in the large room. The results may indicate that the reflection sound of the control source could help for local ANC to create more profitable quiet zones.

5. Conclusions

In this paper, local ANC in a small semi-enclosed space is numerically studied. It is shown that the sound field created in the semi-enclosed space is more like a partially diffused field. A theoretical model is developed to predict the noise attenuation achieved by local ANC in a control field where both the direct sound and the diffuse sound of control sources exist, and a simple experiment is also carried out. Results suggest that the diffuse sound of the control sources may help for local ANC to create quiet zones with more profitable spatial extensions.

Acknowledgement

This work is supported by the National Natural Science Foundation of China (Grand No. 514091 99).

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