Recent progress in research on virtual sound barriers

Xiaojun Qiu[1] and Haishan Zhou[2]

[1]Centre for Audio, Acoustics and Vibration, Faculty of Engineering and IT, University of Technology Sydney, Sydney, Australia
[2]Institute of Acoustics, Nanjing University, Nanjing 210093, China

ABSTRACT
A virtual sound barrier is an active noise control system that uses arrays of loudspeakers and microphones to create a practical size of quiet zone in a noise environment just like an acoustic barrier but without blocking air and light. This technology can be used to reduce sound radiation from noise sources or to reduce noise level around one or few person heads in noisy environments in many industrial and domestic situations. This paper introduces the history, principle and design methods of the virtual sound barriers first, and then describes recent progress in research on the systems, especially the application on power transformer noise radiation control from an enclosure. The paper is concluded by the limitations and future direction discussions of the virtual sound barriers.

1. INTRODUCTION
The Virtual Sound Barrier (VSB) system is an array of acoustic sources and sensors forming an acoustic barrier, which blocks direct propagation of sound without blocking air and light (Qiu, Li & Chen 2005). It can create a quiet zone in a noise environment by using the active noise control (ANC) method, and the main mechanisms are absorption and/or reflection of the noise by the control sources. The theoretical background of the VSB can be traced back to the Huygens' Principle, quantified in one way by the Kirchhoff-Helmholtz integral equation (K-H equation), which shows that for a volume without internal sources, the sound pressure at any given location inside is completely determined by the sound pressure and its normal gradient on the boundary. Thus, if all sound pressure and its normal gradient on the boundary are reduced to zero, the sound pressure inside will be zero too (Nelson & Elliott 1992).

The idea of applying the Huygens' Principle to active sound field control was pointed out by Jessel et al. in 1968, and then various numerical studies and several experiments were undertaken to demonstrate the feasibility of the idea. However, the technology is still far from reality now (Nelson & Elliott 1992). Although there are already some research in active control of sound radiation of a primary source such as a power transformer by using near field control sources, the physical mechanism is to minimize the total power output of the primary source and control sources rather than to control the sound field based on the Huygens principle (Mangiante 1977).

The most related work to the VSB system was reported by Epain and Friot (2007), where 30 error microphones, 30 loudspeakers and a 32-channel ANC controller were used to create a spherical quiet zone with a radius of 0.3 m by applying the boundary pressure control (BPC) technique. The experiments in a quasi-anechoic environment show that noise can be efficiently cancelled everywhere inside the sphere over a wide frequency range for both pure tones and broadband noise. In the same year, the theoretical and experimental studies on a 16-channel cylindrical VSB system were conducted by Zou (2007) to verify the feasibility of the VSB system. The experiments with this cylindrical VSB system in a normal room show that the average reduction of more than 10 dB inside a cylindrical region with 0.2 m height and 0.2 m radius can be achieved up to 550 Hz. The control performance of the system is affected by the distribution of the error sensors and the control sources, and the average noise reduction inside the target region decreases with the increase of the noise frequency at about −6 dB per 100 Hz (Zou et al. 2007).

Later, Zou and Qiu (2008) studied the effects of the presence of a human head within the quiet zone surrounded by the error sensors on the performance of VSB systems. It is found that the introduction of the human head is beneficial to the VSB system in terms of performance robustness to the movements of the human head. A comparison of three cost functions of the VSB system was undertaken with numerical simulations by Zou, Qiu and Lu (2009). Considering the total noise reduction and the uniformity degree of the sound attenuation distribution in the target region, the best strategy is minimizing the sum of the total acoustic energy density at the error sensors.

In practice, the noise reduction in a target region can achieve maximal if the error sensors of the VSB system are located at these target locations. However, this sort of arrangement results in the interference of the error sensors to the human head. This problem can be solved by using the virtual sensing strategy, in which the virtual sensors are positioned in the target region and the physical sensors are positioned at the border of the target region. It is demonstrated that the introduction of the virtual sensors is feasible for developing a compact VSB system (Zou & Qiu 2009).

Because a VSB system is usually used in a room, the system may be located near reflective surfaces. It is shown that when a VSB system is near a reflective surface, its noise reduction performance fluctuates periodically around the performance curve without a nearby surface with the distance between the surface and the system. The performance of the VSB system is sensitive to the incident angle due to the reflective surface (Qiu, Zou & Rao 2009). The sound pressure in the exterior area of a VSB system usually increases due to the sound reflection mechanism of the system; however, the impact on exterior area can be reduced by using specially designed directional control sources (Rao 2011).

In this paper, the principle, design methods and applications of the VSB system are introduced, the results, advantages and limitations of a specific practical application are presented, and the future research directions of VSB systems are discussed.

2. THEORY

For a volume V without internal sources, the sound pressure at any location inside is determined completely by the sound pressure and its normal gradient on the boundary as described by the following K-H equation (Nelson & Elliott 1992)

$$p(\boldsymbol{r}) = \int_S [\mathrm{G}(\boldsymbol{r}\,|\,\boldsymbol{s})\nabla p(\boldsymbol{s}) - p(\boldsymbol{s})\nabla \mathrm{G}(\boldsymbol{r}\,|\,\boldsymbol{s})]\cdot \boldsymbol{n}\, dS \tag{1}$$

where V is surrounded by the surface S, $p(\boldsymbol{s})$ is the sound pressure at position $\boldsymbol{s}$ on the surface S, $p(\boldsymbol{r})$ is the sound pressure at position $\boldsymbol{r}$ inside the volume V and $\boldsymbol{n}$ is the unit vector normal to the surface S pointing outwards from the volume. The free space Green function $G(\boldsymbol{r}|\boldsymbol{s})$ can be expressed as (Nelson & Elliott 1992)

$$\mathrm{G}(\boldsymbol{r}\,|\,\boldsymbol{s}) = \frac{1}{4\pi\,|\,\boldsymbol{r}-\boldsymbol{s}\,|}\mathrm{e}^{-\mathrm{j}k|\boldsymbol{r}-\boldsymbol{s}|} \tag{2}$$

The K-H equation shows that the sound pressure inside the volume V can be reduced by reducing the sound pressure and normal gradient on the boundary. This is the theoretical basis of the VSB systems. Figure 1 shows an example of the VSB systems, in which an array of loudspeakers located in a three-dimension closed structure to create a quiet zone within the space surrounded by the error sensors in a noisy environment.

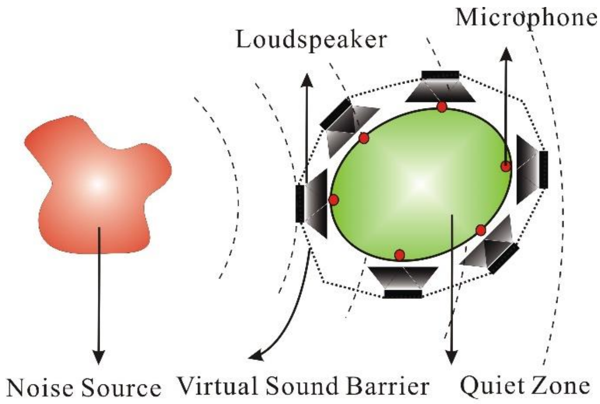

Figure 1: Schematic drawing of a VSB system

The monitoring sensors can be omitted for a steady primary noise field, where the radiation of the loudspeakers can be pre-designed in advance according to the characteristics of the primary noise field, the location of the quiet zone and the arrangement of the loudspeakers. For a general primary noise field where the position, the amplitude and the frequency of the noise source are time-varying, good noise reduction performance of the VSB systems can be achieved with an adaptive controller which receives the input signals of the monitoring sensors and adjusts the output signals to the loudspeakers.

The physical mechanisms of the VSB system in Figure 1 might be different according to the type of the control sources and the control strategies. Since the VSB system is usually far away from the primary source, the mechanism is generally absorption or reflection of the sound energy of the primary source rather than reducing the impedance seen by the primary source (except for the VSB system inside an enclosure which obtains control via modal coupling). For instance, the incident sound from the primary source to the quiet zone will be reflected back when the sound pressure of the sensors is reduced to zero in Figure 1. As a result, the total sound energy is increased after control.

3. DESIGN METHODS

There are two main design methods for the VSB systems. One is the expansion method of the primary sound field which is suitable for steady primary sound fields, and the other is the least mean square method which is applicable to time-varying primary sound fields. The VSB system of a spherical structure with a radius of r_v is used to illustrate these methods. The origin of the coordinate system is set at the centre of the spherical quiet area, and the control sources are located on the surface of the sphere. There are no any sound source or sound scattering object inside the sphere.

3.1 The Expansion Method of the Primary Sound Field

The primary sound field at point $\boldsymbol{r}$ = (r, θ, ϕ) can be expressed as (Williams 1999)

$$p_\mathrm{p}(r,\theta,\phi,k)=\begin{cases}\sum_{n=0}^{\infty}\sum_{m=-n}^{n}A_n^m(k)\mathrm{j}_n(kr)\mathrm{Y}_n^m(\theta,\phi),\ r<r_\mathrm{v}\\ \sum_{n=0}^{\infty}\sum_{m=-n}^{n}C_n^m(k)\mathrm{h}_n(kr)\mathrm{Y}_n^m(\theta,\phi),\ r>r_\mathrm{v}\end{cases} \tag{3}$$

where k is the wavenumber, $A_n^m(k)$ and $C_n^m(k)$ are the coefficients of the spherical harmonics $\mathrm{Y}_n^m(\theta,\phi)$, $\mathrm{j}_n(kr)$ is the spherical Bessel function of the first kind and the nth order, $\mathrm{h}_n(kr)$ is the spherical Hankel function of the first kind and the nth order. $\mathrm{Y}_n^m(\theta,\phi)$ is given by (Williams 1999)

$$\mathrm{Y}_n^m(\theta,\phi)=\sqrt{\frac{(2n+1)}{4\pi}\frac{(n-|m|)!}{(n+|m|)!}}\mathrm{P}_n^{|m|}(\cos\theta)\mathrm{e}^{-\mathrm{j}m\phi} \tag{4}$$

where $\mathrm{P}_n^{|m|}(\cdot)$ is the associated Legendre function of degree n and order m.

The continuous monopole and dipole control sources are located on the surface of the spherical area. According to the K-H equation, the secondary sound field inside the sphere is determined by the sound pressure and its normal gradient on the boundary, which is expressed as

$$p_\mathrm{c}(\boldsymbol{r})=\int_0^{2\pi}\int_0^{\pi}\left[\frac{\partial p(\boldsymbol{r}_\mathrm{v})}{\partial \boldsymbol{n}(\boldsymbol{r}_\mathrm{v})}\mathrm{G}(\boldsymbol{r}\,|\,\boldsymbol{r}_\mathrm{v})-\mathrm{j}kp(\boldsymbol{r}_\mathrm{v})\frac{1}{\mathrm{j}k}\frac{\partial \mathrm{G}(\boldsymbol{r}\,|\,\boldsymbol{r}_\mathrm{v})}{\partial \boldsymbol{n}(\boldsymbol{r}_\mathrm{v})}\right]r_\mathrm{v}^2\sin\theta_\mathrm{v}d\theta_\mathrm{v}d\phi_\mathrm{v} \tag{5}$$

where $\boldsymbol{r}_\mathrm{v}$ = (r_v, θ_v, ϕ_v) is the coordinate of the point on the spherical surface. The source strengths of the monopole and dipole sources are given by

$$S(\boldsymbol{r}_\mathrm{v})=\frac{\partial p(\boldsymbol{r}_\mathrm{v})}{\partial \boldsymbol{n}(\boldsymbol{r}_\mathrm{v})} \tag{6}$$

$$D(\boldsymbol{r}_\mathrm{v}) = \mathrm{j}kp(\boldsymbol{r}_\mathrm{v}) \tag{7}$$

where the directivity of the dipole source is independent of the frequency due to the normalized coefficient jk of the dipole source. The objective of control is to reduce the total sound field to zero, so there is

$$p_\mathrm{t}(\boldsymbol{r}) = p_\mathrm{p}(\boldsymbol{r}) + p_\mathrm{c}(\boldsymbol{r}) = 0, \qquad |\boldsymbol{r}| \le r_\mathrm{v} \tag{8}$$

If $p(\boldsymbol{r}_\mathrm{v})$ in Eqs. (6) and (7) is set to $-p_\mathrm{p}(\boldsymbol{r}_\mathrm{v})$, the zero sound pressure inside the quiet area is achieved. Therefore, the source strengths of the control sources can be calculated by

$$S(\boldsymbol{r}_\mathrm{v}) = -k\sum_{n=0}^{\infty}\sum_{m=-n}^{n} A_n^m(k)\mathrm{j}_n'(kr_\mathrm{v})\mathrm{Y}_n^m(\theta_\mathrm{v},\phi_\mathrm{v}) \tag{9}$$

$$D(\boldsymbol{r}_\mathrm{v}) = -\mathrm{j}k\sum_{n=0}^{\infty}\sum_{m=-n}^{n} A_n^m(k)\mathrm{j}_n(kr_\mathrm{v})\mathrm{Y}_n^m(\theta_\mathrm{v},\phi_\mathrm{v}) \tag{10}$$

The continuous sources have to be discretised in practice. With N_c discrete sources, the secondary sound field is written as (Rao 2011)

$$p_\mathrm{c}(\boldsymbol{r}) = \sum_{l=1}^{N_\mathrm{c}} r_\mathrm{v}^2\beta_l\left[S(\boldsymbol{r}_l)\mathrm{G}(\boldsymbol{r}\,|\,\boldsymbol{r}_l) - D(\boldsymbol{r}_l)\frac{1}{\mathrm{j}k}\frac{\partial \mathrm{G}(\boldsymbol{r}\,|\,\boldsymbol{r}_l)}{\partial \boldsymbol{n}(\boldsymbol{r}_l)}\right] \tag{11}$$

where $\boldsymbol{r}_l = (r_\mathrm{v}, \theta_l, \phi_l)$ is the coordinate of the lth source, β_l is the weight coefficient of the spherical harmonics. When the spherical harmonics expansion is of the Nth order, the strengths of the discrete control sources are given by

$$S(\boldsymbol{r}_l) = -k\sum_{n=0}^{N}\sum_{m=-n}^{n} A_n^m(k)\mathrm{j}_n'(kr_\mathrm{v})\mathrm{Y}_n^m(\theta_l,\phi_l) \tag{12}$$

$$D(\boldsymbol{r}_l) = -\mathrm{j}k\sum_{n=0}^{N}\sum_{m=-n}^{n} A_n^m(k)\mathrm{j}_n(kr_\mathrm{v})\mathrm{Y}_n^m(\theta_l,\phi_l) \tag{13}$$

3.2 The Least Mean Square Method

Assume N_c first order control sources of the VSB system are uniformly distributed on the spherical surface with a radius of r_v at the coordinate $\boldsymbol{r}_l = (r_\mathrm{v}, \theta_l, \phi_l)$. The secondary sound field can be expressed as (Rao 2011)

$$p_\mathrm{c}(\boldsymbol{r}) = \sum_{l=1}^{N_\mathrm{c}} \frac{q_l \mathrm{e}^{-\mathrm{j}k|\boldsymbol{r}-\boldsymbol{r}_l|}}{4\pi|\boldsymbol{r}-\boldsymbol{r}_l|}\left[a - (1-a)\left(1 - \frac{\mathrm{j}}{k|\boldsymbol{r}-\boldsymbol{r}_l|}\right)\cos\gamma\right] \tag{14}$$

where q_l is the strength of the lth control source. The rest part on the right-hand side of the equation is the general expression of the first order directional source, and the weight of the directivity α is between 0 and 1 (Williams 1999). When α is equal to 1, 0 and 0.5, the source will be the monopole (omnidirectional source), the dipole (figure-eight source) and the tripole (hyper-cardioid source), respectively. γ is the angle between the vector $\boldsymbol{r}$–$\boldsymbol{r}_l$ and the axis of the source.

The total sound field after control is given by

$$p_{\mathrm{t}}(\boldsymbol{r}) = p_{\mathrm{p}}(\boldsymbol{r}) + p_{\mathrm{c}}(\boldsymbol{r}) \tag{15}$$

The cost functions to be minimized can be the sum of acoustic potential energy density, the sum of acoustic kinetic energy density and the sum of the total acoustic energy density at the error sensors (Zou 2007),

$$J_{\mathrm{p}} = \sum_{n=1}^{N_{\mathrm{e}}} | p_{\mathrm{t}}(\boldsymbol{r}_n^{\mathrm{e}}) |^2 + \beta \boldsymbol{q}_{\mathrm{c}}^{\mathrm{H}} \boldsymbol{q}_{\mathrm{c}} \tag{16}$$

$$J_{\mathrm{k}} = \sum_{n=1}^{N_{\mathrm{e}}} | v_{\mathrm{t}}(\boldsymbol{r}_n^{\mathrm{e}}) |^2 + \beta \boldsymbol{q}_{\mathrm{c}}^{\mathrm{H}} \boldsymbol{q}_{\mathrm{c}} \tag{17}$$

$$J_{\mathrm{e}} = \sum_{n=1}^{N_{\mathrm{e}}} [\frac{1}{2\rho_0 c_0^2} | p_{\mathrm{t}}(\boldsymbol{r}_n^{\mathrm{e}}) |^2 + \frac{\rho_0}{2} | v_{\mathrm{t}}(\boldsymbol{r}_n^{\mathrm{e}}) |^2] + \beta \boldsymbol{q}_{\mathrm{c}}^{\mathrm{H}} \boldsymbol{q}_{\mathrm{c}} \tag{18}$$

where c_0 is the speed of sound in the air, ρ_0 is the air density, N_{e} is the number of error sensors located at $\{\mathbf{r}_n^{\mathrm{e}}$, n=1,2,… N_{e} }, $\boldsymbol{q}_{\mathrm{c}}=[q_1, q_2, ..., q_{N\mathrm{c}}]^{\mathrm{T}}$ is the vector of the control source strengths, $v_{\mathrm{t}}(\boldsymbol{r})$ is the normal component of the particle velocity. β is set to an appropriate value to restrain the control source output power. The error sensors are usually located at the boundary of the quiet zone with a radius smaller than r_{v}.

The cost functions in Eqs. (16)-(18) can be expressed in the quadratic form as (Zou 2007)

$$J = \boldsymbol{q}_{\mathrm{c}}^{\mathrm{H}} (\boldsymbol{A} + \beta \mathbf{I}) \boldsymbol{q}_{\mathrm{c}} + \boldsymbol{q}_{\mathrm{c}}^{\mathrm{H}} \boldsymbol{b} + \boldsymbol{b}^{\mathrm{H}} \boldsymbol{q}_{\mathrm{c}} + c \tag{19}$$

where $\boldsymbol{A}$ is a matrix related to the transfer functions between the pressure (and/or particle velocity) at error sensors and the control source strengths, $\boldsymbol{b}$ is a vector related to the transfer functions mentioned above and the primary sound field, c is a constant only related to the primary sound field. For example, when the cost function is the acoustic potential energy density shown in (16), the corresponding parameters are expressed as (Zou 2007)

$$\boldsymbol{A} = \boldsymbol{Z}_{\mathrm{c}}^{\mathrm{H}} \boldsymbol{Z}_{\mathrm{c}}, \;\; \boldsymbol{b} = \boldsymbol{Z}_{\mathrm{c}}^{\mathrm{H}} \boldsymbol{p}_{\mathrm{p}}, \;\; c = \boldsymbol{p}_{\mathrm{p}}^{\mathrm{H}} \boldsymbol{p}_{\mathrm{p}}, \tag{20}$$

$$\boldsymbol{Z}_{\mathrm{c}} = \begin{bmatrix} Z_{\mathrm{ce}}(\boldsymbol{r}_1^{\mathrm{e}} | \boldsymbol{r}_1^{\mathrm{c}}) & Z_{\mathrm{ce}}(\boldsymbol{r}_1^{\mathrm{e}} | \boldsymbol{r}_2^{\mathrm{c}}) & \ldots & Z_{\mathrm{ce}}(\boldsymbol{r}_1^{\mathrm{e}} | \boldsymbol{r}_{N_{\mathrm{c}}}^{\mathrm{c}}) \\ Z_{\mathrm{ce}}(\boldsymbol{r}_2^{\mathrm{e}} | \boldsymbol{r}_1^{\mathrm{c}}) & Z_{\mathrm{ce}}(\boldsymbol{r}_2^{\mathrm{e}} | \boldsymbol{r}_2^{\mathrm{c}}) & \ldots & Z_{\mathrm{ce}}(\boldsymbol{r}_2^{\mathrm{e}} | \boldsymbol{r}_{N_{\mathrm{c}}}^{\mathrm{c}}) \\ \ldots & \ldots & \ldots & \ldots \\ Z_{\mathrm{ce}}(\boldsymbol{r}_{N_{\mathrm{e}}}^{\mathrm{e}} | \boldsymbol{r}_1^{\mathrm{c}}) & Z_{\mathrm{ce}}(\boldsymbol{r}_{N_{\mathrm{e}}}^{\mathrm{e}} | \boldsymbol{r}_2^{\mathrm{c}}) & \ldots & Z_{\mathrm{ce}}(\boldsymbol{r}_{N_{\mathrm{e}}}^{\mathrm{e}} | \boldsymbol{r}_{N_{\mathrm{c}}}^{\mathrm{c}}) \end{bmatrix} \tag{21}$$

$$\boldsymbol{p}_{\mathrm{p}} = [p_{\mathrm{p}}(\boldsymbol{r}_1^{\mathrm{e}}) \quad p_{\mathrm{p}}(\boldsymbol{r}_2^{\mathrm{e}}) \quad \ldots \quad p_{\mathrm{p}}(\boldsymbol{r}_{N_{\mathrm{e}}}^{\mathrm{e}}) \;]^{\mathrm{T}} \tag{22}$$

where

$$Z_{\mathrm{ce}}(\boldsymbol{r}_n^{\mathrm{e}} | \boldsymbol{r}_m^{\mathrm{c}}) = \frac{\mathrm{e}^{-\mathrm{j}k|\boldsymbol{r}_n^{\mathrm{e}} - \boldsymbol{r}_m^{\mathrm{c}}|}}{4\pi |\boldsymbol{r}_n^{\mathrm{e}} - \boldsymbol{r}_m^{\mathrm{c}}|} \left[a - (1-a) \left(1 - \frac{\mathrm{j}}{k |\boldsymbol{r}_n^{\mathrm{e}} - \boldsymbol{r}_m^{\mathrm{c}}|} \right) \cos \gamma_{n,m} \right] \tag{23}$$

$$\cos \gamma_{n,m} = \frac{(\boldsymbol{r}_n^{\mathrm{e}} - \boldsymbol{r}_m^{\mathrm{c}}) \bullet \boldsymbol{r}_m^{\mathrm{c}}}{|\boldsymbol{r}_n^{\mathrm{e}} - \boldsymbol{r}_m^{\mathrm{c}}| |\boldsymbol{r}_m^{\mathrm{c}}|} \tag{24}$$

The optimal strength of control sources for all above cost functions is given by (Zou 2007)

$$\boldsymbol{q}_{\mathrm{c}} = -(\boldsymbol{A} + \beta \mathbf{I})^{-1} \boldsymbol{b} \tag{25}$$

4. APPLICATIONS

The VSB system can be applied to many industrial and domestic scenarios which have noise reduction and ventilation and/or lighting requirements. One typical application is the noise control of large power transformers. The power transformer noise consists of the fundamental frequency and its harmonics, and most of the noise components to be controlled are in the low frequency range. Sound barriers are usually built around the outdoor transformers to block the transformer noise. However, the noise control performance of the sound barrier is poor in the low frequency range due to the sound diffraction at the edge of the barrier and low transmission loss of the barrier. Some transformers are located inside enclosed rooms, suffering from poor ventilation condition and expensive building cost. In practice, for convenient maintenance and ventilation, some transformers are installed in a semi-closed space with one side opening or two side openings. Under these situations, the transformer noise is mainly transmitted out through the openings.

A planar VSB system formed by a loudspeaker array and a microphone array can be arranged on the openings of the enclosures to block the noise as shown in Figure 2. It is a hybrid active and passive noise control system, where the noise is constrained to the opening due to the large transmission loss of the walls and ceiling and active noise control is only applied on the opening.

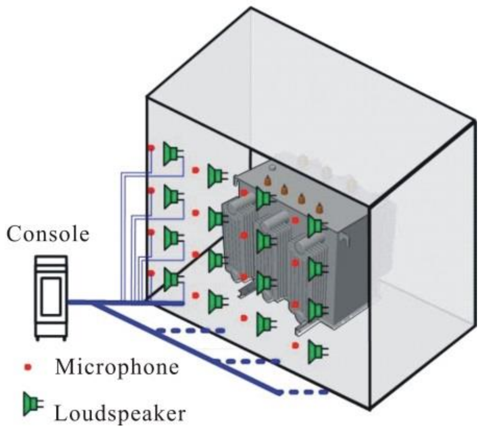

Figure 2: Schematic drawing of a planar VSB system for transformer noise control

A 15-channel VSB system was mounted at the opening of the sound insulation room in Nanjing University, as shown in Figure 3 (Wang, Tao & Qiu 2014). The opening is about 1.58×3.21 m^2, with 15 loudspeakers spaced uniformly in 3 rows and 5 columns. 15 error sensors are ahead of the loudspeakers with the distance of 0.2 m. A loudspeaker located on the floor as a primary noise source is 3 meter away to the opening. The lateral and vertical intervals between the loudspeakers are about 0.63 m and 0.72 m respectively, which results in the upper frequency limit of the effective noise reduction being about 238 Hz. Only tonal noise of 100 Hz and 200 Hz were controlled by the system. A 16-channel commercial controller embedded with the FxLMS algorithm was used in the experiment.

Figure 3: A photo of a planar VSB experiment system

The measurement points are at 1, 3, 5, 8 and 10 m away to the opening and at 1.5 m height. The sound pressure level reduction at these points is no less than 15 dB for 100 Hz and 8 dB for 200 Hz, as illustrated in Table 1. Similar results were found with the simulation results (Wang, Tao & Qiu 2015).

Table 1: Noise reduction (dB) of an experimental planar VSB system

Distance to the opening	1 m	3 m	5 m	8 m	10 m
100 Hz	18	18	17	18	21
200 Hz	8	15	8	12	8

One advantage of this system is that the noise radiation can be reduced without affecting air ventilation and light, and the other advantage is its flexibility for installation and maintenance. The limitation of the system at present is the cost. For a large opening or high frequency sound radiation, many more secondary sound sources are needed. The error sensors, the secondary sound sources, the multiple channel controller and the cabling all contribute to the cost.

5. DISCUSSIONS

Based on the existing studies on the VSB systems and other related research, the feasibility of the VSB systems has been verified by theory studies, numerical simulations and experiments. In the complex sound field where the noise comes from many different directions, the quiet zone with certain size can be generated by the system, especially in the low frequency range. However, the system is still hard to be applied in practice at present due to the narrow and low frequency range of effective noise reduction, the complexity and the high cost. Along with the increase of the upper frequency limit of effective noise reduction, the number of control channel increases dramatically with the square of the frequency. For example, using a 16-chanel VSB system, the upper frequency limit is about 500 Hz to have an average reduction of more than 10 dB inside a cylindrical space with 0.2 m height and 0.2 m radius. If the upper frequency limit increases to 4000 Hz, the required number of channels is up to 1024.

It has been shown that the above upper frequency limit can be increased or the quiet zone can be enlarged by using a cost function for minimizing the sum of the total acoustic energy density at the error sensors (Zou, Qiu & Lu 2009). In this case, the sound pressure and three orthogonal components of the particle velocities need to be measured by the microphones and the particle velocity sensors respectively. The adaptive ANC algorithm to minimize the sum of the total acoustic energy density needs to be developed. This includes approaches for secondary path modeling with 4 error signals (one sound pressure and three particle velocities at three directions) and methods for reducing algorithm complexity and memory space requirements by using the redundancy information of these 4 signals.

Particle velocity transducers are usually more expensive than microphones in practical applications, so it is preferred to use microphones in most practical systems. The problem with the sound pressure sensing strategies is that the effective noise reduction cannot be achieved at some frequencies due to the interior Dirichilet problem with zero sound pressure at the boundary [8]. Double layer sensor array is a potential compromise method. Although there are preliminary studies on the optimization of the control source directivity of the VSB system (Poletti, Abhayapala & Samarasinghe 2012), experiment research is still not available. Considering the two elements of the VSB system, a general solution with a double layer sensor array and a double layer control source array may be feasible, which is illustrated in Figure 4 (Chang & Jacobsen 2013). This solution can be further investigated both theoretically and experimentally.

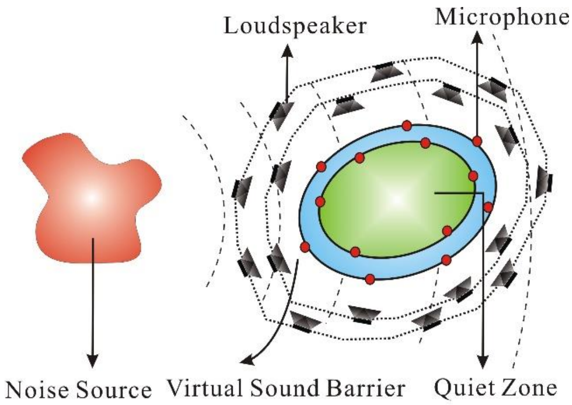

Figure 4: Schematic drawing of a double layer VSB system

The error sensors of the VSB system are expected to be omitted for compact practical systems. The main difficulty is how to ensure the effective noise control in different primary sound fields. A potential method is using a feedforward control system with fixed control parameters, as shown in Figure 5. The reference sensor array is arranged outside the control source array, and the control signal of the speakers are determined by some mapping equations which include the relationships between the sound field information on the surface of the reference sensor array and the sound field information inside the VSB system and the relationships between the input of control source and the sound field information inside the VSB system. All these relationships, as well as the robustness of the mapping equations to various primary sound fields, can be investigated in the future.

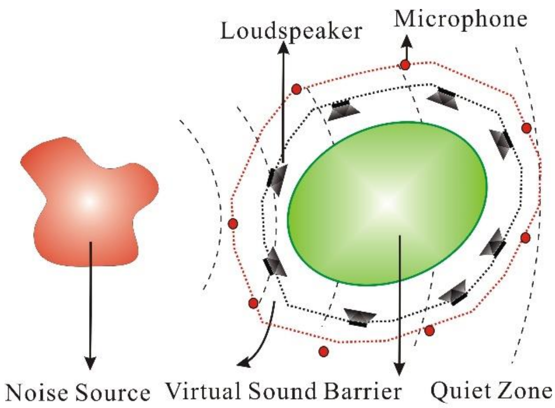

Figure 5: Schematic drawing of a mapping VSB system

A hybrid active and passive control method might improve the performance of a VSB system by increasing the noise reduction inside the quiet zone or increasing the upper frequency limit of effective noise control or reducing the channel numbers (Pan & Qiu 2008). One kind of the hybrid VSB systems is composed of some active control sources and several passive sound insulation boards, which are arranged at intervals, as shown in Figure 6.

The physical mechanisms of this hybrid VSB system are not clear, and the effect of the intervals, the size and the arrangement of the boards on the performance of the system can be analyzed. The VSB system located in or close to a designed acoustic structure such as a corner is another kind of hybrid system as shown in Figure 7. Future studies can be focused on the physical mechanisms of the systems, the effect of the acoustic characteristics of wall surfaces on the control performance, the optimization methods of acoustic parameters and the shape of the structures.

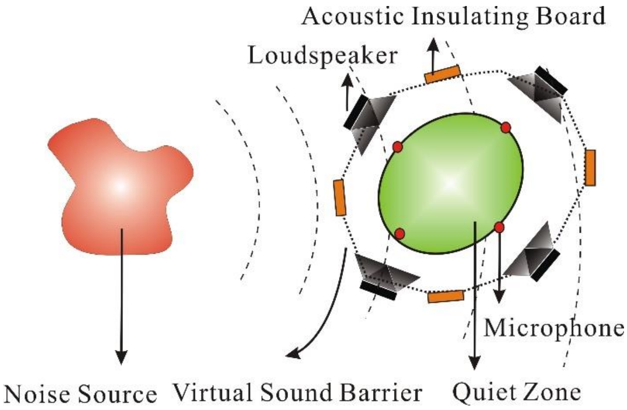

Figure 6: Schematic drawing of a partition hybrid VSB system

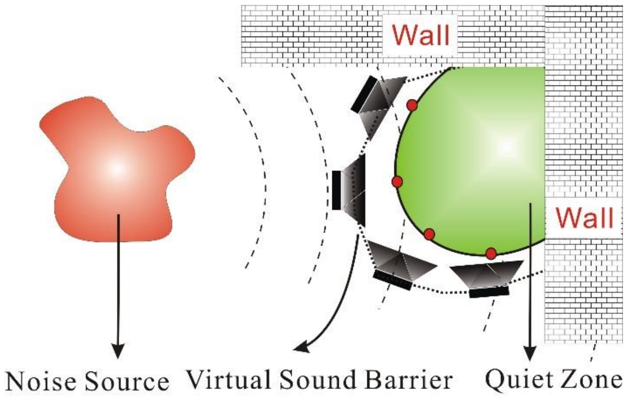

Figure 7: Schematic drawing of a corner hybrid VSB system

VSB systems are completed ANC systems, which use loudspeaker and microphone arrays and signal processing methods to control the sound field. The hybrid VSB systems introduce different acoustic elements at the acoustic boundary, and the sound field control is obtained by integrating functions of passive elements with different impedance and adaptive active elements. From the algorithm aspects, there are also many problems can be explored such as the stability condition of non-ideal and insufficient length secondary path model and its influence on the adaptive control algorithms. To further overcome the detrimental effect of imperfect secondary path model, the control algorithms without secondary path modeling are of particular interest. To reduce the complexity and cost of practical VSB systems, it is also important to investigate the implementation of feedforward, feedback and/or integrated control structures as well as the centralized and decentralized control strategies.

6. CONCLUSIONS

The progress, principle, design methods and some applications of the VSB systems are introduced. Although the feasibility of the VSB systems has been verified by existing research, it is still far from practical applications due to the narrow and/or low frequency range of effective noise reduction, the complexity and the high cost of the systems. To solve these problems, future research directions for the VSB systems are discussed, which include the VSB systems with active sound energy control, double layer VSB systems, and active and passive hybrid VSB systems.

If the upper frequency limit of VSB systems can be increased, it has potential to be applied to many practical noise control scenarios that have ventilation and/or lighting constrains.

ACKNOWLEDGEMENTS

This work was supported under Australian Research Council's Linkage Projects funding scheme (LP140100987) and National Nature Science Foundation of China (Projects 11474163).

REFERENCES

Chang, JH & Jacobsen, F 2013, ‘Experimental validation of sound field control with a circular double-layer array of loudspeakers’, *The Journal of the Acoustical Society of America*, vol. 133(4), pp. 2046-2054.

Epain, N & Friot, E 2007, ‘Active control of sound inside a sphere via control of the acoustic pressure at the boundary surface’, *Journal of Sound and Vibration*, vol. 299(3), pp. 587-604.

Mangiante,G 1977, ‘Active sound absorption’, *The Journal of the Acoustical Society of America*, vol. 61(6), pp. 1516-1523.

Nelson, PA & Elliott, SJ 1992, *Active control of sound*, London: Academic Press.

Pan, J & Qiu, X 2008, ‘Performance of an active control system near a reflecting surface’, *Australian Journal of Mechanical Engineering*, vol. 5(1), pp. 35-42.

Poletti, MA, Abhayapala, TD & Samarasinghe, P 2012, ‘Interior and exterior sound field control using two dimensional higher-order variable-directivity sources’, *The Journal of the Acoustical Society of America*, vol. 131(5), pp. 3814-3823.

Qiu, X, Li, N & Chen, G 2005, ‘Feasibility study of developing practical virtual sound barrier system’, *Proceedings of the 12th International Congress on Sound and Vibration*, Lisbon, Portugal.

Qiu, X, Zou, H & Rao, W 2009, ‘Performance of a virtual sound barrier near a reflective surface’, *Proceedings of the 7th International Symposium on Active control of Sound and Vibration* (Active 2009), pp. 20-22, Ottawa, Canada.

Rao, W 2011, ‘A study of effects of boundary and directional sound sources on virtual sound barrier systems’, Master Degree Thesis, Nanjing University, Nanjing.

Wang, S, Tao, J & Qiu, X 2014, ‘Active control of transformer noise radiated outside a three-dimensional building with one opening’, *Proceedings of the Twenty-first International Congress on Sound and Vibration* (ICSV21), Beijing, China.

Wang, S, Tao, J & Qiu, X 2015, ‘Performance of a planar virtual sound barrier at the baffled opening of a rectangular cavity’, *Journal of the Acoustical Society of America*, 138(5), pp. 2836-2847

Williams, E 1999, *Fourier acoustics: sound radiation and nearfield acoustical holography*, New York: Academic.

Zou, H 2007, ‘A study on virtual sound barrier system in frequency domain’, Doctor Degree Thesis, Nanjing University, Nanjing.

Zou, H & Qiu, X 2008, ‘Performance analysis of the virtual sound barrier system with a diffracting sphere’*, Applied Acoustics*, vol. 69(10), pp. 875-883.

Zou, H & Qiu, X 2009, ‘Optimization of the locations of the virtual sensors in the virtual sound barrier system’, *Journal of Nanjing University*, vol. 45(1), pp. 57-66.

Zou, H, Qiu, X & Lu, J 2009, ‘A study of cost functions for the virtual sound barrier system’, *Proceedings of 38th International Congress & Exhibition on Noise Control Engineering* (inter-noise 2009), Ottawa, Canada.

Zou, H, Qiu, X, Lu, J & Niu, F 2007, ‘A preliminary experimental study on virtual sound barrier system’*, Journal of Sound and Vibration*, vol. 307(1), pp. 379-385.