Sound Field Controller Design Method based on Partial Model Matching on Frequency Domain

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Abstract: In this paper, a simple method to design an MIMO sound field control system was proposed. The control system was designed to achieve 1) noise attenuation and 2) sound equalization by utilizing feedback and feedforward controllers. The method was based on partial model matching on frequency domain which only required measured frequency response data, or impulse response data in order to tune parameters of the controller. The proposed method was applied to a normal office room and results of experiment showed effectiveness of the proposed method.

Keywords: sound field control, feedback–feedforward controller, partial model matching on frequency domain, noise reduction

1. INTRODUCTION

The objective of sound field control is to control acoustic characteristics of a room such as reverbrating time, or to attenuate noise by using acoustic devices. For example, it has been applied to attenuate noise from fans in air conditioners of rooms or from vibration of engine in a car. It also has been applied in concert halls or multi media home theater systems in order to make movies more real in acoustic sense. Owing to the development of microcomputers and digital signal processing techniques, these sound field control was achieved by using feedforward digital filters based on Head related transfer functions(HRTF)[1], inverse system filters[2], [3], [4], [5] and so on. It is able to reproduce acoustic properties of a famous concert hall by convoluting dry source signals and impulse responses of the concert hall on real time[6].

Although feedforward filters are effective, sound signals through those filters are generated by speakers and they are usually deformed via sound field of the listening room or by noise from other sources. In this case, the sound field control may fail to reproduce the desired sound field without taking the effect of those issues into account. Feedback control is effective in order to overcome those difficulties, especially for noise reduction. Therefore, two degree of freedom(2-DOF) control system[7] which has feedforward and feedback controllers is adopted for sound field control and a method to tune parameters of controllers is proposed in this paper. By using feedback controllers, details of the sound field of the listening room is required in order to show that the controlled system is stable. Usually, a sound field covers wide frequency band, and this means that it becomes rather complex to derive mathematical models of it. Hence, it is required to design controllers only by measured characteristics of the listening room, and that is what considered in this paper. Since characteristics of a sound field are well expressed on frequency domain, the design method would be effective when controllers are designed on frequency domain.

From this view point, a controller design method by using measured numerical frequency response data was proposed in order for vibration control of mechanical systems[8], [9], [10]. The method is called partial model matching on frequency domain since controllers are tuned only on specified several frequencies. The method is applicable to MIMO system and the designed control system becomes stable in the sense of partial model matching. In this paper, the method is extended for 2-DOF controller design and adopted for sound field control which achieves 1) noise attenuation and 2) sound equalization simultaneously.

In the next section, the sound filed of the listening room is modeled. The control objectives are also modeled and the design method of controllers is shown. The proposed method is adopted to an actual room in section 3. and results of experiments are shown. Finally, conclusions follow in section 4.

2. CONTROL SYSTEM

2.1. Model of a sound field control system

A stereo sound field which is considered in this paper is the simplest case in order to reproduce the desired sound field[11]. The system is considered to be a linear 2 input 2 output MIMO system and controllers are designed as linear. Let $\boldsymbol{G}(s)$ represent a 2×2 transfer function matrix as

$$\boldsymbol{G}(s) = \begin{bmatrix} g_{11}(s) & g_{12}(s) \\ g_{21}(s) & g_{22}(s) \end{bmatrix} \tag{1}$$

where $g_{ij}(s)$ are assumed to be stable transfer functions. Similarly, linear controllers are denoted by transfer function matrices $\boldsymbol{C_{FB}}(s)$ and $\boldsymbol{C_{FF}}(s)$ as follow.

$$\boldsymbol{C_{FB}}(s) = \begin{bmatrix} \frac{n_{FB_{11}}(s)}{d_{FB_{11}}(s)} & \frac{n_{FB_{12}}(s)}{d_{FB_{12}}(s)} \\ \frac{n_{FB_{21}}(s)}{d_{FB_{21}}(s)} & \frac{n_{FB_{22}}(s)}{d_{FB_{22}}(s)} \end{bmatrix} \tag{2}$$

$$\boldsymbol{C_{FF}}(s) = \begin{bmatrix} \frac{n_{FF_{11}}(s)}{d_{FF_{11}}(s)} & \frac{n_{FF_{12}}(s)}{d_{FF_{12}}(s)} \\ \frac{n_{FF_{21}}(s)}{d_{FF_{21}}(s)} & \frac{n_{FF_{22}}(s)}{d_{FF_{22}}(s)} \end{bmatrix}, \tag{3}$$

where $d_{FB_{ij}}(s)$ and $d_{FF_{ij}}(s)(i, j = 1, 2)$ are stable denominators. $n_{FB_{ij}}(s)$ and $n_{FF_{ij}}(s)(i, j = 1, 2)$ are defined as

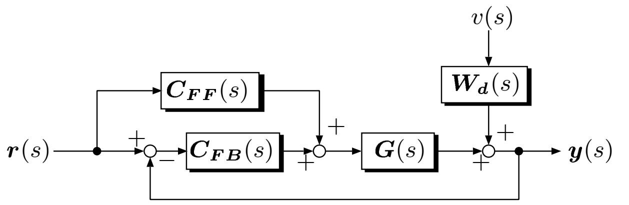

Fig. 1. Block diagram of the control system

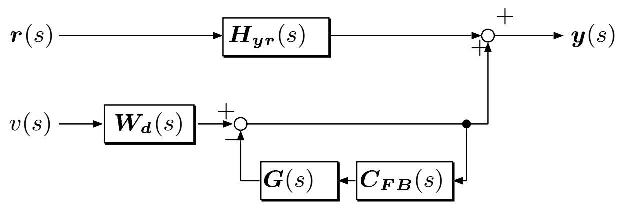

Fig. 2. Equivalent block diagram of the control system

follow.

$$n_{FB_{ij}}(s) = \sum_{l=1}^{r_{FB}} c_{FB_{ijl}} s^{r_{FB}-l} \quad (4)$$

$$n_{FF_{ij}}(s) = \sum_{l=1}^{r_{FF}} c_{FF_{ijl}} s^{r_{FF}-l}, \quad (5)$$

where $c_{FB_{ijl}}$ and $c_{FF_{ijl}}$ are parameters of controllers which are derived below and r_{FB} and r_{FF} are numbers of parameters of $c_{FB_{ijl}}$ and $c_{FF_{ijl}}$ respectively. Let $\boldsymbol{r}(s)$, $\boldsymbol{y}(s)$ and $\boldsymbol{W_d}(s)$ represent reference inputs, outputs observed by microphones and transfer functions from the source of noise to microphones respectively. Fig.1 shows the block diagram of the sound field control system.

Let $\boldsymbol{I}$ be a 2×2 unit matrix and define $\boldsymbol{H}_0(s)$ as follows.

$$\boldsymbol{H}_0(s) = \boldsymbol{I} + \boldsymbol{G}(s)\boldsymbol{C_{FB}}(s),$$

Assume that $\boldsymbol{H}_0(s)$ is not singular. Transfer function matrices from inputs $\boldsymbol{r}$ and from noise to outputs $\boldsymbol{y}$ are

$$\boldsymbol{H_{yr}}(s) = \boldsymbol{H}_0^{-1}(s)\boldsymbol{G}(s)(\boldsymbol{C_{FB}}(s) + \boldsymbol{C_{FF}}(s)) \quad (6)$$

$$\boldsymbol{H_{yv}}(s) = \boldsymbol{H}_0^{-1}(s)\boldsymbol{W_d}(s). \quad (7)$$

By using these notations,

$$\boldsymbol{y}(s) = \boldsymbol{H_{yr}}(s)\boldsymbol{r}(s) + \boldsymbol{H_{yv}}(s)v(s). \quad (8)$$

Fig.2 shows the system which is equivalent to the system of (8) in Fig.1 with the same structure of the reference system(Fig.3). The reference system consists of desired reference models which are derived below. Desired reference models are transfer functions which satisfies 1) noise reduction and 2) sound equalization in the reference system. From above, the objective of the sound field control system is to design controllers $\boldsymbol{C_{FB}}(s)$ and $\boldsymbol{C_{FF}}(s)$ such that make $\boldsymbol{H_{yr}}(s)$ and $\boldsymbol{H_{yv}}(s)$ be close to desired reference models in some sense.

Since characteristics of the system on frequency domain are important in sound field control, "distance" between the controlled system and the reference system is measured on frequency domain as cost functions. By optimizing those cost

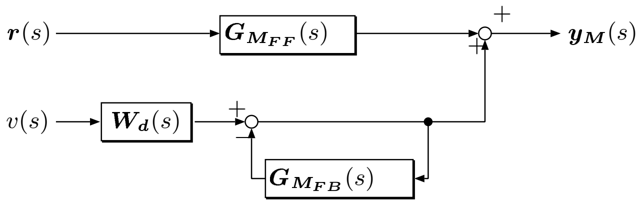

Fig. 3. Block diagram of the reference system

functions under conditions such as stability, parameters of controllers are derived.

It is worth noting that $\boldsymbol{C_{FB}}(s)$ and $\boldsymbol{C_{FF}}(s)$ are designed as same kind optimization problem as follows. $\boldsymbol{C_{FB}}(s)$ can be obtained by achieving

$$\boldsymbol{G}(s)\boldsymbol{C_{FB}}(s) \to \boldsymbol{G_{M_{FB}}}(s) \quad (9)$$

under some conditions. Given $\boldsymbol{C_{FB}}(s)$ and denote $\tilde{\boldsymbol{G}}(s)$ and $\tilde{\boldsymbol{G_{M_{FF}}}}(s)$ as follow.

$$\tilde{\boldsymbol{G}}(s) = \boldsymbol{H}_0^{-1}(s)\boldsymbol{G}(s)$$

$$\tilde{\boldsymbol{G_{M_{FF}}}}(s) = \boldsymbol{G_{M_{FF}}}(s) - \tilde{\boldsymbol{G}}(s)\boldsymbol{C_{FB}}(s)$$

Then, $\boldsymbol{H_{yr}}(s) \to \boldsymbol{G_{M_{FF}}}(s)$ can be rewritten as $\tilde{\boldsymbol{G}}(s)\boldsymbol{C_{FF}}(s) \to \tilde{\boldsymbol{G_{M_{FF}}}}(s)$. This is the same form of (9).

2.2. Reference Models

2.2.1 $\boldsymbol{G_{M_{FB}}}(s)$

In this paper, only one source of noise $v(s)$ is considered and transfer functions $\boldsymbol{W_d}(s)(2 \times 1)$ are assumed that they have strong power at several frequencies, or peaky on frequency domain. Although this assumption may sound unusual, noise from fans or engines has strong peaks depending on their physical dynamics. The simplest case that noise has only one peak is considered in this paper. It is also assumed that information about $\boldsymbol{W_d}(s)$ is available for the controller design. Since outputs $\boldsymbol{y}(s)$ without inputs correspond to $\boldsymbol{W_d}(s)v(s)$, $\boldsymbol{W_d}(s)$ is available by assuming $v(s)$ has uniform power on the considered band of the frequency domain.

Sensitivity functions from the noise to outputs is given as $\boldsymbol{H_0}(s)^{-1}\boldsymbol{W_d}(s)$ and noise reduction can be achieved by reducing the sensitivity. This means that $\boldsymbol{G_{M_{FB}}}(s)$ should be modeled as $(\boldsymbol{I} + \boldsymbol{G_{M_{FB}}}(s))^{-1}\boldsymbol{W_d}(s)$ attenuate the noise. Intuitively, this can be done when $\|\boldsymbol{G_{M_{FB}}}(j\omega)\| \approx k\|\boldsymbol{W_d}(j\omega)\|$ ($k \in R^+$, $\forall\omega \in R^+$) with large k. Considering stability, transfer functions of $\boldsymbol{G_{M_{FB}}}(s)$ must be minimum phase. Therefore, $\boldsymbol{G_{M_{FB}}}(s)$ is modeled as

$$\boldsymbol{G_{M_{FB}}}(s) = k \begin{bmatrix} \frac{n_{M_{FB}}(s)}{d_{M_{FB}}(s)} & 0 \\ 0 & \frac{n_{M_{FB}}(s)}{d_{M_{FB}}(s)} \end{bmatrix}, \quad (10)$$

where $n_{M_{FB}}(s)$, $d_{M_{FB}}(s)$ and k are defined adequately by information about measured $\boldsymbol{W_d}(s)$ and desired performance. i, j element of $\boldsymbol{G_{M_{FB}}}(s)$ is denoted as $g_{M_{FB}ij}(s)$.

2.2.2 $\boldsymbol{G_{M_{FF}}}(s)$

In this paper, the purpose of feedforward controllers $\boldsymbol{G_{M_{FF}}}(s)$ is defined to achieve sound equalization, or $\boldsymbol{y}(t) \approx$

$\boldsymbol{r}(t)$. Therefore, the reference model $\boldsymbol{G_{M_{FF}}}(s)$ is designed to have "flat" frequency response. Since high frequency sound causes "noisy feeling", the "flat" frequency response is modeled by first order systems. Adding to this, in order to enhance binaural sound reproduction ability, non diagonal transfer functions which correspond to cross-talk terms are defined "smaller" than diagonal ones. Details of $\boldsymbol{G_{M_{FF}}}(s)$ is defined as follow.

$$\boldsymbol{G_{M_{FF}}}(s) = \begin{bmatrix} \frac{\gamma_d}{s + c_{FF}} & \frac{\gamma_n}{s + c_{FF}} \\ \frac{\gamma_n}{s + c_{FF}} & \frac{\gamma_d}{s + c_{FF}} \end{bmatrix} \quad (11)$$

where γ_d, γ_n and c_{FF} are parameters and they are defined by considering the desired performance.

2.3. Cost Functions

The frequency band considered in sound field control is usually wide i.e. from about 10Hz to about 50kHz and dynamics of the sound field is complex. Therefore, it is difficult, or impossible, to make the controlled system match the reference system exactly. However, it is not required to match the system to the reference perfectly since human's acoustic sense dose not recognize the frequency response of the sound field so precisely. Our acoustic sense dose not percept a couple dB difference of the gain, or phase of sound waves of high frequency. In this sense, it is sufficient that parameters of controllers are derived to make the system approximately close to the reference and, therefore, it is proposed to use partial model matching on frequency domain to this end. Since parameters of $\boldsymbol{C_{FF}}(s)$ and $\boldsymbol{C_{FB}}(s)$ are able to be derived by solving optimization problems which have same form, $\boldsymbol{C_{FB}}(s)$ case is shown in details in this paper.

The cost function in partial model matching is defined as the distance between the frequency response of the controlled system and that of the reference system at several specified frequencies. Number of these specified frequencies is required to be limited and they are called matching frequencies denoted as ω_{FB_q}. Denote the set consisting of matching frequencies as

$$\Omega_{FB} = [\omega_{FB_1}, \cdots, \omega_{FB_q}, \cdots, \omega_{FB_{N_{FB}}}] \quad (12)$$

where N_{FB} is the number of matching frequencies.

Define functions as

$$\begin{aligned} f(x,y) &= \begin{cases} x & (x \text{ is grater than } y \text{ [dB]}) \\ 0 & (x \text{ is smaller than } y \text{ [dB]}) \end{cases} \\ \Delta_{1_{iq}}(\omega) &= |gc_{FB_{iq}}(j\omega) - g_{M_{FB}iq}(j\omega)| \\ \Delta_{2_{iq}}(\omega) &= \left| |gc_{FB_{iq}}(j\omega)| - |g_{M_{FB}iq}(j\omega)| \right|. \end{aligned}$$

where $gc_{FB_{iq}}(s)$ represents i,q element of $\boldsymbol{G}(s)\boldsymbol{C}_{FB}(s)$. Then, the distance from elements of $\boldsymbol{G}(s)\boldsymbol{C_{FB}}(s)$ to those of $\boldsymbol{G_{M_{FB}}}(s)$ is defined as

$$J_{FB_{iq}}(\omega) = \begin{cases} f(\Delta_{1_{iq}}(\omega), \delta_{1_{iq}}) & \omega \leq \omega_0 \\ f(\Delta_{2_{iq}}(\omega), \delta_{2_{iq}}) & \omega > \omega_0 \end{cases} \quad (13)$$

where δ_{1_i} and δ_{2_i} represent widths of dead zone. Since it is enough to match the system to the reference approximately, dead zone is introduced in the cost function. $J_{FB_{ii}}$ dose not care difference in phase at high frequency(higher than ω_0) because our acoustic sense dose not precept phase at high frequency. Summarizing (13) over Ω_{FB}, the cost function J_{FB} is defined as follows.

$$J_{FB} = \sum_{\omega \in \Omega} \sum_{i,q=1,2} J_{FB_{iq}}(\omega) \quad (14)$$

Similarly, cost function, matching frequencies and its set are defined for $\boldsymbol{C_{FF}}(s)$ as J_{FF}, ω_{FF_q} and Ω_{FF} respectively.

2.4. Constraints

Stability of feedback systems like Fig.1 is required to be guaranteed. Now, it is sufficient to consider the stability of the close loop system since feedforward controllers $\boldsymbol{C_{FF}}(s)$ are stable. Constraints based on the stability criteria by Rosenbrock[12] are defined by following [8], [9], [10]. Constraints are also computed on specified frequencies as cost functions are. Denote diagonal constraint frequency, non-diagonal constraint frequency, number of constraints and a set of constraint frequencies as ω_{d_q}, ω_{nd_q}, N_d, N_{nd} ,Ω_d and Ω_{nd} respectively. The close loop system is considered to be stable when following constraints are satisfied for $i = 1, 2$[8], [9], [10].

$$|g_{M_{FB}ii}(j\omega) - gc_{FBii}(j\omega)|^2 \leq |R_{ik_d}(j\omega)|^2, \omega \in \Omega_d \quad (15)$$

$$|1 + gc_{FBii}(j\omega)|^2 \geq \alpha_i \sum_{\substack{q=1 \\ q \neq i}}^{m} |gc_{FBqi}(j\omega)|^2, \omega \in \Omega_{nd} \quad (16)$$

where

$$R_{ik_d}(j\omega) = \zeta_i |g_{M_{FB}ii}(j\omega)|.$$

ζ_i and α_i are defined such that

$$0 < \zeta_i \leq \left| 1 + \frac{1}{g_{M_{FB}ii}(j\omega)} \right|, \qquad 1 < \alpha_i$$

From above, parameters of $\boldsymbol{C_{FB}}(s)$ are derived by minimizing cost function (14) under constraints (15) and (16).

On the other hand, there is no constraint to be satisfied by $\boldsymbol{C_{FF}}(s)$. Therefore, by using parameters of $\boldsymbol{C_{FB}}(s)$ derived above, parameters of $\boldsymbol{C_{FF}}(s)$ are able to be obtained by minimizing its cost function which is defined as J_{FB} is defined above in (14).

The optimization problem was solved by using Optimization Toolbox[14] of MATLAB.

3. EXPERIMENT

3.1. System setup

In order to validate the proposed method, the method was applied to control the sound field of our office. Fig.4 and Table1 show the experiment system and its specification. The listening room was a 6[m] square and 2.7[m] height room. In the listening room, two speakers were installed with 2.5[m] space between them. The listening point was 1.7[m] from each speaker and two microphones were placed there. Analog sound signals from microphones were sampled by the digital mixer at 44100Hz as 2ch 12bit digital audio signals and they were trasmitted to DSP which computed control inputs. Control inputs from DSP were transmited back to the digital mixer and converted to analog signals. They were sent to the amplifier and control inputs were reproduced by speakers. DSP was installed in a PC which logged all signals processed by DSP.

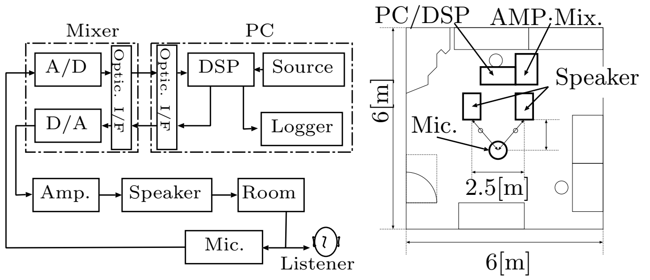

Fig. 4. Control system

Table 1. Specification

DSP	MTT DSP6067/AP2190		
Mixer	Fostex Digital Mixer VM-88		
Mic.	Primo EMU-4740	Speaker	Bose 464
Amp.	Bose 1200VI	PC	IBM-PC

The proposed method requires frequency responses of the listening room in order to design controllers. Time stretched pulse(TSP)[13] was used to measure impulse responses and frequency responses were obtained by transferring impulse responses numerically. Refer [13] in details. Fig.5 and Fig.6 show measured impulse responses and frequency responses respectively. Note that these responses not only includes responses of the listening room but also those of devices such as speakers, microphones and the digital mixer.

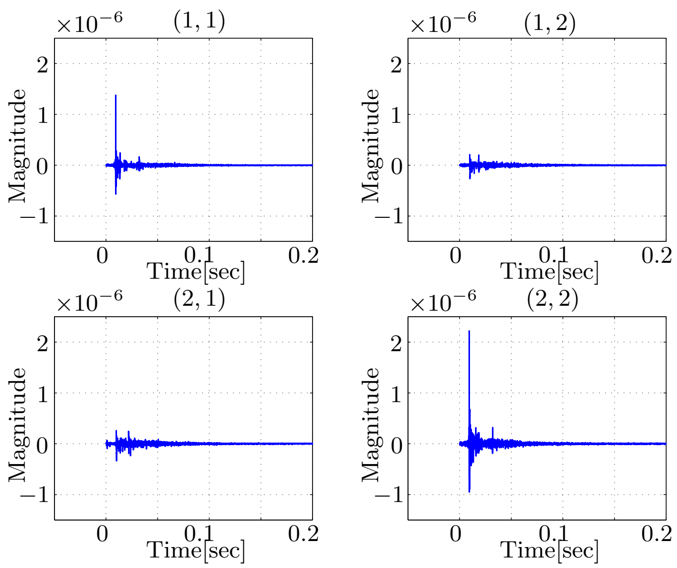

Fig. 5. Impulse responses of the listening room

Noise was modeled such that it had one discrete power at 500Hz i.e. $\frac{Y_{\max}}{2}\sin(2\pi \times 500t)$ where Y_{max} represented the maximum value that could be input to the amplifier.

Taking above information into account, reference models

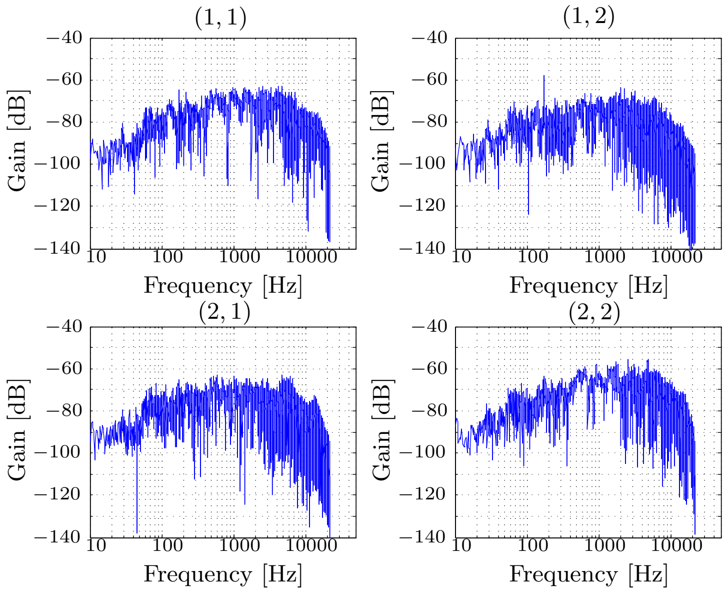

Fig. 6. Frequency responses of the listening room

were defined as follow(Fig.7 and 8)

$$\boldsymbol{G_{M_{FB}}}(s) = \begin{bmatrix} \frac{1.257\times 10s - 1.000\times 10^{-2}}{s^2+3.142s+9.870\times 10^6} & 0 \\ 0 & \frac{1.257\times 10s - 1.000\times 10^{-2}}{s^2+3.142s+9.870\times 10^6} \end{bmatrix}$$

$$\boldsymbol{G_{M_{FF}}}(s) = \begin{bmatrix} \frac{9.797\times 10}{s+1.256\times 10^4} & \frac{5.041\times 10}{s+1.256\times 10^4} \\ \frac{4.675\times 10}{s+1.2556\times 10^4} & \frac{1.229\times 10^2}{s+1.256\times 10^4} \end{bmatrix}$$

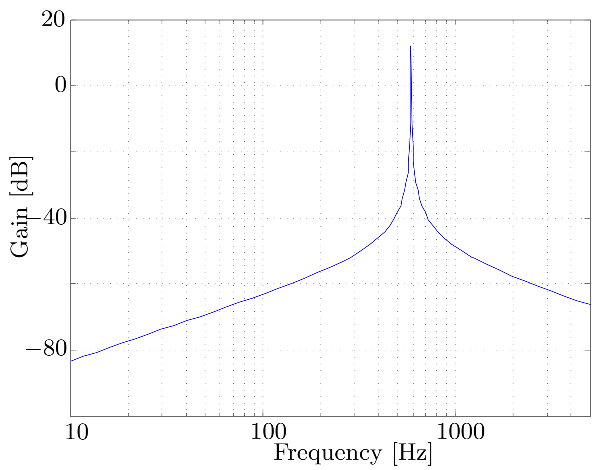

Fig. 7. Desired frequency response of feedback system

Parameters for partial model matching such as matching frequencies, constraint frequencies and so on are listed in Table2. α_i and ζ_i in (15) and (16) are defined adequately.

Then, controllers were derived by usind the proposed method. In order to implent the derived controllers, they

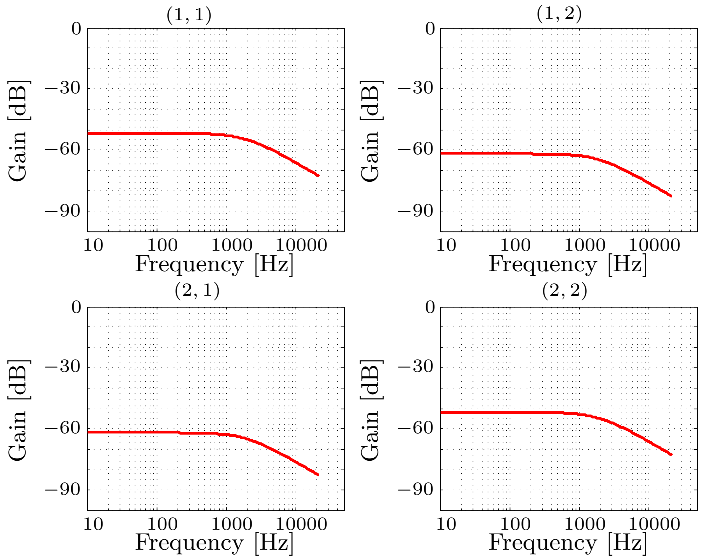

Fig. 8. Desired frequency response of feedforward system

were discretized and obtained as follow.

$$C_{FB}(z) = \frac{1}{0.9999z^{-2} - 1.995z^{-1} + 1.000} \times$$

$$\begin{bmatrix} 0.06103z^{-2} - 0.1191z^{-1} + 0.05810 & -0.01424z^{-2} + 0.029481z^{-1} - 0.01524 \\ 0.009719z^{-2} - 0.01543z^{-1} + 0.005717 & 0.04137z^{-2} - 0.08398z^{-1} + 0.04261 \end{bmatrix}$$

$$C_{FF}(z) = \frac{1}{-0.7504z^{-3} + 2.497z^{-2} - 2.745z^{-1} + 1.000} \times$$

$$\begin{bmatrix} -0.6981z^{-3} + 2.327z^{-2} - 2.563z^{-1} + 0.9354 & -0.01196z^{-3} + 0.04016z^{-2} - 0.04469z^{-1} + 0.01649 \\ -0.01196z^{-3} - 0.02053z^{-2} + 0.01887z^{-1} - 0.005404 & 0.007060z^{-3} - 0.02053z^{-2} - 2.604z^{-1} + 0.9491 \end{bmatrix}$$

3.2. Results

By using above system, experimental results are shown in this subsection.

At the first experiment, noise reduction was tested by letting $\boldsymbol{r}(t) = \boldsymbol{0}$. Using feedback controllers derived above, time responses of outputs are shown in Fig.9. During the first 1.5 second, controllers were off and then they were turned on.

Table 2. Parameters (at the experiment)

Ω_{FF}	[31.64133.5498.4500.1501.8562.42371 10000] $[Hz]$		
Ω_{FB}, Ω_d and Ω_{nd}	$[160.2, 500.0, 1000]$ $[Hz]$		
$\delta_{1_{iq}}$	3 [dB]	$\delta_{2_{iq}}$	3 [dB]
ω_0	100[Hz]		

The figure shows good performance of noise reduction and the noise was surpressed to smaller than $\frac{1}{5}$ of its original magnitude. Although it is difficult to vanish the noise perfectly only by using output feedback control, the performance of feedback control system is considered to be sufficient since authors listened the tone throught the experiment and found that the tone was surpressed siginificantly soon after controllers were activated. This result will be shown at the presentation.

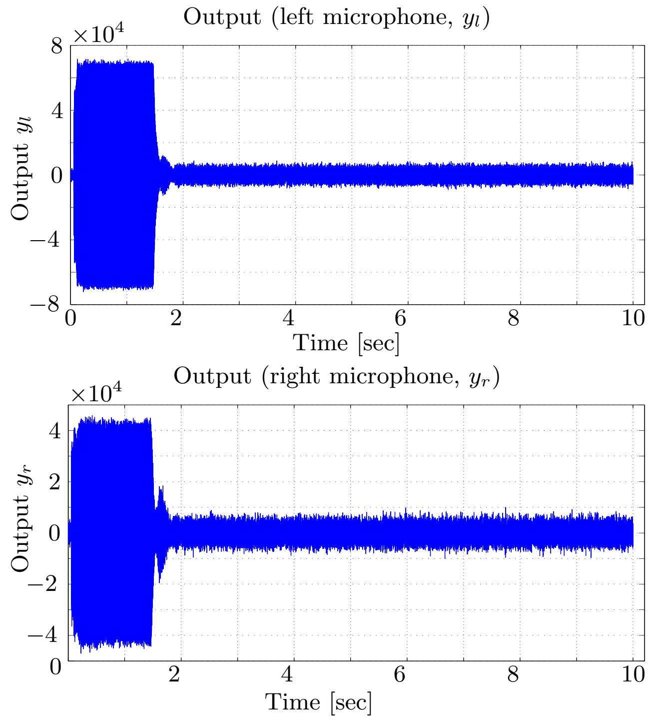

Fig. 9. Noise reduction

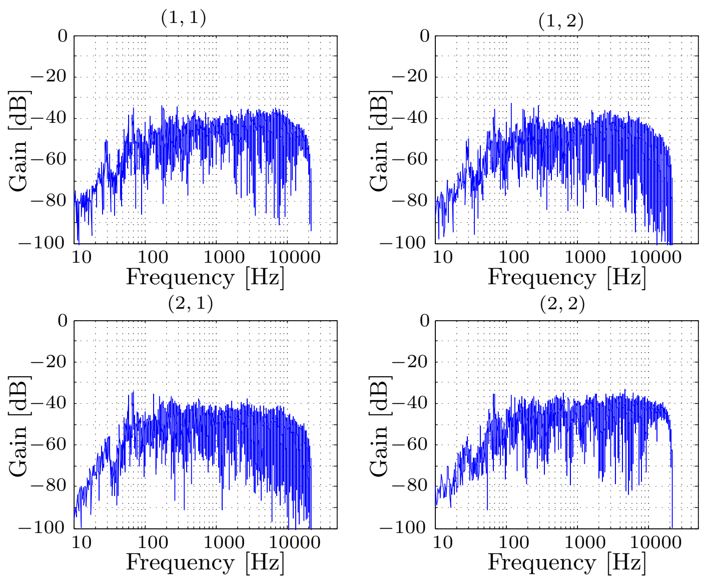

Fig. 10. Frequency response of the controlled room

Second experiment was aimed to show the performance of sound equalization. Feedback and feedforward controllers were activated and TSP signals were used to measure frequency responses of the controlled listening room. Fig.10 shows the result. The figure shows that controlled responses got "flat" than the original frequency responses(Fig.6). Diagonal responses were larger than nondiagonal responses as they were designed in the reference model. From above, it can be said that the proposed method worked well in sound field control in order to achieve 1) noise reduction and 2) sound equalization.

For further research, comparing frequency responses and the desired frequency responses in Fig.8, slight difference can be found in high and low frequency although they mathced siginificantly well in the middle of the considered frequency band. Reasons of mismatches are considered as follow.

- In low frequency, the amplifier and speakers do not have enough ability to reproduce control inputs. Decaying time of the listening room was rather short, i.e. less than 0.1 second, and it meant that frequency responses were not able to be well identified in low frequency(lower than 10Hz).

- In high frequency, the controlled frequency responses did not decay as desireds responses were. Although the relative degree of the desired model was 1, that of feedfoward controllers was 0 in order to compensate the effect by feedback controllers. Difference of relative degree might cause mismatch in high frequency.

4. CONCLUSION

In this paper, the method to design the sound field controller which achieved noise reduction and sound equalization simultaneously was proposed by extending partial model matching method to 2-DOF feedback and feedforward control system. The method is practical since it only requires numerical measured frequency response of the target room. The proposed method was applied to sound field control of an office room. For simplicity, noise interfering the sound field was assumed to have only one frequency component. Results of experiments showed that the derived control system was able to attenuate noise sufficiently and that frequency responses of the sound field were controlled to match the desired reference model. Therefore, the validity of the proposed method was shown through experiments.

It is not shown in this paper but it is worth noting that authors found that noise was almost perfectly reduced when a stereo music signal was supplied as reference input. Owing to feedforward and feedback controllers, authors were not able to percieve that the noise was reduced and that the original music signal was not distorted. Further studies including psycological tests are required to show the performance of the controlled system.

In future studies, noise which has more general characteristics is required to be considered. Matching performance is also required to be enhanced as mentioned in the exeperiment section. To this end, it is thought to be effective to use high order controllers or subband techniques.

In this paper, the cost function was designed by considering characteristics of human's perception of listening. In order to achieve high performance in sound field control with respect to human's perception, charecteristics of human's listening ability such as masking is required to be taken into account in future works.

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