

# Low Order Modeling of an Acoustic Path for Active Noise Control and Design Method of Feedback Control

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## ABSTRACT

In order to reduce the cost of active noise reduction (ANC), this research focused on feedback (FB) control, which characteristically does by not require reference signal sensors. The purpose of this paper is low order modeling of acoustic transfer characteristic (plant) between a control noise source and an error sensor arranged at the control point; and, the designing of a controller based on the results. Specifically, this entails constructing as simple an FB-ANC system as possible (a 1 input/1 output system), then investigating the noise reduction performance of that system. First, the authors employed physical modeling in order to model the plant in the lowest possible order. In addition, Smith predictor-FB-ANC is presented, and FB-ANC based on PID (Proportional Integral Derivative) control is designed for the low-order plant model obtained. Finally, the validity of these results is demonstrated with computer simulation using actual data.

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## 1. Introduction

A major issue in active noise control (ANC) is its high cost. For instance, active control of interior road noise employing feed forward (FF) control<sup>(1)-(3)</sup> requires numerous references signals (i.e., input signals) because of the complexity of input paths. Consequently, installing this system at low-cost has been considered difficult to achieve. Another issue is determining the proper placement of reference signal sensors.

As a means of resolving these issues, this research project focuses on feedback (FB) control, which characteristically does not require reference signal sensors. In FB-ANC research, the Tight Coupled Monopole (TCM) method<sup>(4), (5)</sup>, whereby error sensors (i.e., microphones) are placed near the control noise source (i.e., speakers), is well-known. However, few cases have addressed FB-ANC from the control engineering perspective of using modeling and design of a controller based on a model. In recent years, there has been some research from a control engineering perspective<sup>(6)-(10)</sup>, but FB-ANC research is still less extensive than FF-ANC research. FB-ANC offers the advantage of a simplified installation system, yet there have been few investigations of its strengths and weaknesses with respect to noise reduction. Hence, clarifying the performance of FB-ANC is an important prerequisite for further advancements in ANC technology.

Against this background, this paper has as its objectives the low-order modeling of acoustic transfer characteristics from a control sound source to an error sensor located at the control point (i.e., the “plant”), and the designing of a

controller based on those results<sup>(11)</sup>. Specifically, this entails constructing as simple an FB-ANC system as possible (single input/single output), then investigating the noise reduction performance of that system. First, physical modeling is employed in order to model the plant in the lowest possible order. Next, to create a high-precision model, gray-box modeling<sup>(12)</sup>, which uses physical modeling information and experimental data is applied. In addition, Smith predictor type FB-ANC<sup>(13), (14)</sup> is proposed, and FB-ANC based on Proportional Integral Derivative (PID) control is designed for the low-order plant model obtained. Finally, the validity of these results is demonstrated with computer simulation using actual data.

Throughout this paper, the terms “speaker” and “microphone” respectively refer to the control sound source and error sensor used in this research.

## 2. Physical Modeling of the Acoustic Path

### 2.1. Conventional Modeling Techniques

In the field of ANC, system identification is frequently used as a means of modeling. Least Mean Square (LMS) algorithm is widely employed as the identification method because of low calculations volume, Finite Impulse Response (FIR) as the identification model because of its stability<sup>(15)</sup>. Nevertheless, precise identification requires numerous FIR model parameters in order for impulse responses to converge. Designing an FB controller with an FIR model in accordance with control theory is also difficult because of the need to express the plant with an Infinite Impulse Response (IIR) model.

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On the other hand, there have been cases reported in which the plant was expressed as an IIR model to reduce the parameters in the mathematical model, and in which the Simplified Hyperstable Adaptive Recursive Filter (SHARF) method was used as the identification method<sup>(16)</sup>. The actual number of parameters, however, was determined by trial and error. Therefore, as a means of determining the number of parameters with quantitative indicators, we proposed an IIR model identification method that employs a subspace method<sup>(10)</sup>. Using this model we have expressed the plant with an IIR model having twelve parameters (i.e., a sixth-order IIR filter).

Nevertheless, it is not clear to what extent the number of parameters can be reduced while still maintaining the requisite precision. Therefore, determining this will not only simplify the designing of FB controllers, but it is also an important factor in achieving low-cost ANC. We therefore use physical modeling of the plant to obtain a high-precision, low-order model<sup>(7), (17), (18)</sup>.

## 2.2. Physical Modeling

Below, physical modeling is performed separately for the three components of the plant: speaker, sound field, and microphone<sup>(18)</sup>.

### 2.2.1. Physical Modeling of the Speaker

When the speaker's mechanical system is equivalently converted into electrical circuits, the transfer function  $P_s(s)$  from voltage  $e_0$  [V], which is impressed on the circuit terminal, to vibration velocity  $v_0$  [m/s] of the diaphragm is expressed as shown below:

$$P_s(s) = \frac{As}{a_1s^2 + a_2s + C_Mr} \quad (1)$$

where

$$a_1 = \frac{8}{3} a^3 \rho r + m r + L r_M$$

$$a_2 = r \cdot r_M + C_M L + A^2$$

where

$r$  [kg/m<sup>3</sup>] = air density,  
 $a$  [m] = cone radius,  
 $m$  [kg] = diaphragm and coil mass,  
 $A$  = electrodynamic converter's force factor,  
 $r$  [W] = coil resistance,  
 $L$  [H] = coil inductance,  
 $r_M$  [Ns/m] = mechanical resistance,  
 $C_M$  [N/m] = stiffness of the diaphragm support and  
 $s$  = a complex number in Laplacian transformation.

### 2.2.2. Physical Modeling of the Sound Field

We assume that the distance from speaker to microphone can be made sufficiently short relative to the wavelength of the sound to be controlled. Because the effect of direct sounds, rather than reflected sounds, are predominant even in a closed space such as an indoor chamber, the effect of the natural acoustic mode is believed to be minimal. Therefore, a free sound field is assumed for simplicity's sake.

The speaker's radiative surface is assumed to approximate a planar piston because vibration velocity of the diaphragm is equivalent to that of air particles on the diaphragm. The result is an equation of transfer function

$P_A(s)$  from diaphragm air particle velocity  $v_0$  [m/s] to the sound pressure at a point on the central axis located distance  $l$  [m] from the planar piston. That equation is as shown below:

$$P_A(s) = \rho c \left\{ \exp\left(-\frac{l}{c}s\right) - \exp\left(-\frac{\sqrt{l^2 + a^2}}{c}s\right) \right\} \quad (2)$$

where  $c$  [m/s] is the speed of sound<sup>(18)</sup>. When the distance to the measurement point is extremely long relative to speaker radius, or in the case of a low-range frequency, the approximation

$$P_A(s) = \rho \frac{a^2}{2l} s \cdot \exp\left(-\frac{l}{c}s\right) \quad (3)$$

holds true. Equation (3) shows that the sound radiation characteristics comprise derivative element  $(\rho a^2/2l)s$  and time delay element  $\exp\{-(l/c)s\}$ . As the target of noise reduction was the 40-500 Hz frequency range, we used equations (2) and (3) to calculate changes in sound pressure over distance when 40 Hz and 500 Hz sine waves were excited. The radius is 0.0475 which is the same as the actual speaker used in the experiment. The results, shown in Fig.1, show equation (3) to be fully capable of expressing the transfer function of a sound field when the distance is 0.1 m or greater.

In addition, when the effects of the natural acoustic mode are such that they cannot be disregarded, they, too, must be incorporated into modeling.

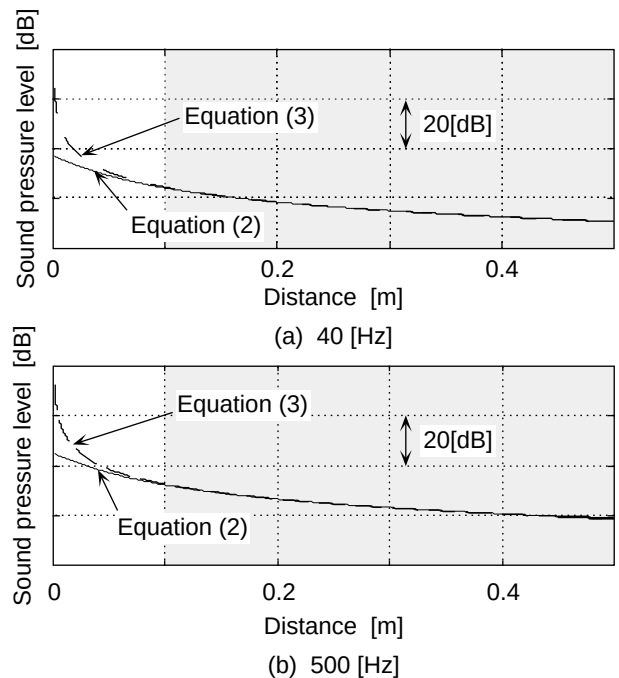


Fig.1 Distance vs sound pressure

### 2.2.3. Physical Modeling of the Microphone

This research project assumes the use of a condenser microphone, using stiffness control. As the frequency range to be controlled has flat characteristics, with no phase differences, the microphone's characteristics can be expressed in terms of proportional gain. Here, for simplicity's sake, the microphone's transfer function  $P_m(s)$  is expressed

below:

$$P_M(s) = 1 \quad (4)$$

### 2.3. Physical Model of the Acoustic Path

A physical model of the acoustic path can be expressed with the three aforementioned components. Table 1 gives the physical parameters of the speaker used. Using the physical parameters in Table 1 to calculate the transfer function of the plant for a speaker-to-microphone distance of 0.1 m, the calculations are as shown below:

$$\begin{aligned} P(s) &= P_S(s)P_A(s)P_M(s) \\ &= \frac{(\rho A a^2 / 2l) s^2}{a_1 s^2 + a_2 s + C_{MR}} \cdot \exp\left(-\frac{l}{c} s\right) \\ &= \frac{2.03 \times 10^{-2} s^2}{9.03 \times 10^{-3} s^2 + 12.43 s + 3512.6} \\ &\quad \cdot \exp(-2.94 \times 10^{-4} s) \\ &= P_C(s) \exp(-\tau s) \end{aligned} \quad (5)$$

where

$$\begin{aligned} P_C(s) &= \frac{2.03 \times 10^{-2} s^2}{9.03 \times 10^{-3} s^2 + 12.43 s + 3512.6} \\ \tau &= 2.94 \times 10^{-4} \end{aligned}$$

Thus, the plant can be expressed as the second-order lumped-parameter part  $P_C(s)$  and time delay element  $\exp(-\tau s)$ .

The next step is to validate the physical model obtained. To do this, system identification was performed by using input/output data of the plant measured in the semi-anechoic chamber. A total of 20000 samples were taken at a sampling frequency of 1378 Hz. An FIR model with 84 parameters was used for identification. Bode plots of the physical model and the results of identification are shown in Fig.2. Although the models generally agree in the gain diagram, the physical model and FIR model in the phase diagram do not agree. Let us consider the reason for this.

### 2.4. A Consideration of Phase Difference

The input/output coherence function obtained in the system identification experiment is shown in Fig.3. These

Table 1 Parameter of the loudspeaker (Panasonic WS-A10-K)

| Parameter             |        | Value                     |
|-----------------------|--------|---------------------------|
| Mass                  | $m$    | $3.1 \times 10^{-3}$ [kg] |
| Mechanical resistance | $r_m$  | 2.6 [Ns/m]                |
| Mechanical stiffness  | $C_m$  | 1351 [N/m]                |
| Coil resistance       | $r$    | 2.6 [ $\Omega$ ]          |
| Coil inductance       | $L$    | $5.4 \times 10^{-5}$ [H]  |
| Cone radius           | $a$    | 0.0475 [m]                |
| Force factor          | $A$    | 2.98 [Tm, N/A]            |
| Air density           | $\rho$ | 1.21 [kg/m <sup>3</sup> ] |
| Wave speed            | $c$    | 340 [m/s]                 |

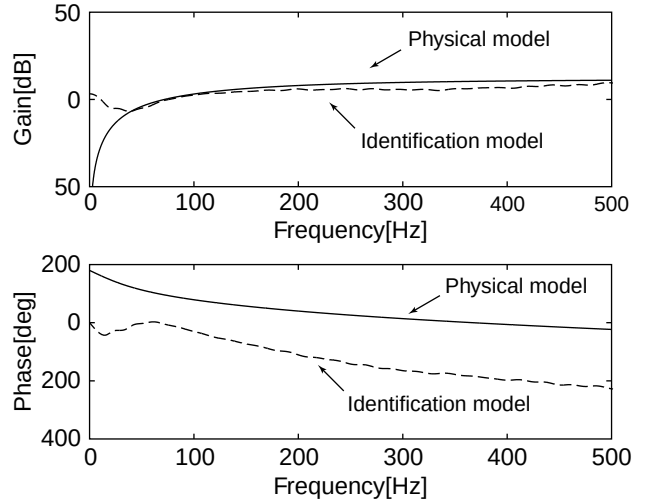


Fig.2 Comparison of Bode plot between a physical model and an identification model

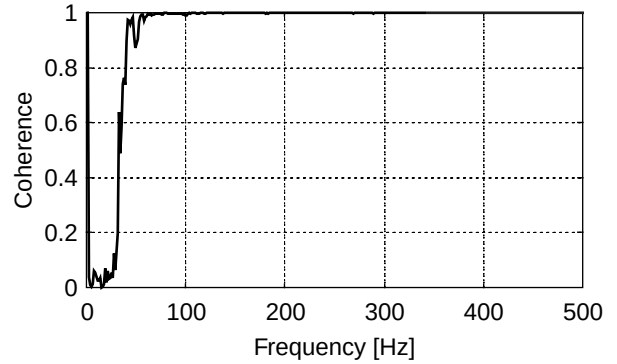


Fig.3 Coherence function

results suggest that because of the difficulty of producing noise from the speakers in the low-frequency band (below 40 Hz), coherence was extremely low. In Fig.2, phase is expressed continuously, rather than from -180 to 180 degrees because of the control system design. This means that phase has been calculated based on the phase of a band having low coherence and low identification results reliability. This results in the incorrect calculation of the absolute value for phase in the high-coherence range. This makes it impossible to determine whether the phase determined by identification or the phase calculated with the physical models is correct.

In the next section, therefore, phase is accurately estimated and gray-box modeling using the estimated phase are performed.

## 3. Gray-box Modeling of Acoustic Path

In this section, we construct a highly accurate model using gray-box modeling<sup>(12)</sup>. Here, physical model information are used for the order of the plant (structural information about the plant), and system identification data are used for adjusting parameters.

Physical modeling, a method for directly describing the dynamic characteristics of the subject in accordance with

the laws of physics, is referred to as white box modeling. System identification is referred to as black box modeling because the subject is considered a black box. Gray-box modeling is so called because it uses both of these modeling techniques to achieve greater modeling accuracy.

In estimating the phase in the coherent frequency range, the cross-correlation function is expressed as

$$\phi_{uy}(\tau) = E[u(t)y(t+\tau)] \quad (6)$$

where  $u(t)$  is input,  $y(t)$  output, and  $\tau$ —time delay. The cross-correlation function determines the time delay between input and output and so can be used to estimate the phase. Fig.4, which shows the phase estimated from the cross-correlation function, demonstrates how the estimated phase very closely approximates not the physical model, but the identification model.

Next, we performed gray-box modeling using the structural information for the physical model along with the phase estimated with the cross-correlation function, as in Fig.4. The difference between the phases determined with the physical model and with the cross-correlation function was determined and added to the physical model as a time delay. However, because simply adding to the physical model does not assure proper expression of phase characteristics, the parameters of the lumped parameter part were also adjusted.

The transfer function of the model obtained by this gray-box modeling is expressed in the equation (7):

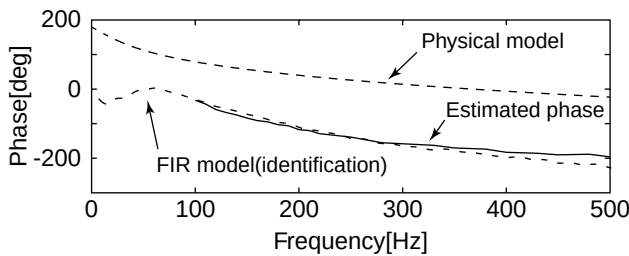


Fig.4 Estimated phase using cross-correlation function

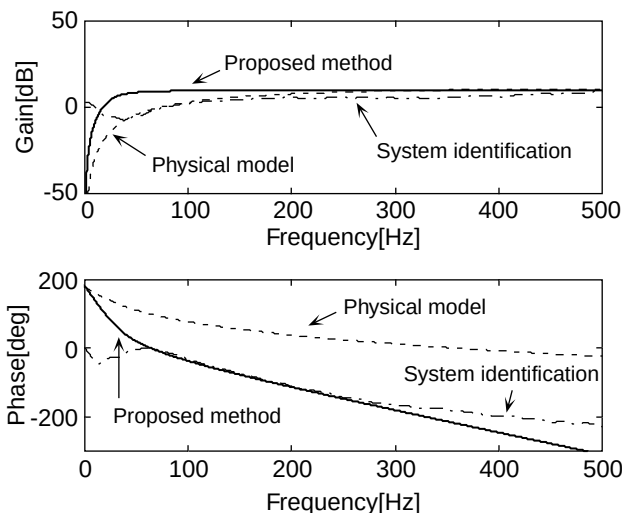


Fig.5 Comparison of Bode plot

$$P(s) = \frac{0.485s^2}{0.155s^2 + 45.1s + 4550} e^{-1.75 \times 10^{-3}s} \quad (7)$$

The Bode plots are in Fig.5.

In Fig.5, the phase corresponds up to approximately 250 Hz, beyond which it lags relative to the phase suggested by the system identification model. This is because the phases were compared primarily in the low range -- which is effective for ANC -- due to the difficulty of getting the phases to correspond throughout the 0-500 Hz range when expressing the physical model with a second-order system and time delay.

#### 4. The Proposed Smith Predictor-type FB-ANC

With equation (5), the target plant was expressed in terms of the lumped parameter part and the time delay element that occurs because of the distance delay. The fact that the plant is a time delay series entails three problems:

- 1.) The inability to express the plant with a rational polynomial expression makes it difficult to design the controller in accordance with control theory.
- 2.) Even when using Padé approximation<sup>(19)</sup> for the time delay element, a higher degree of approximation accuracy results in a higher plant order, which in turn makes for a high-order controller.
- 3.) The existence of the time delay element results in a non-minimum phase system, and the waterbed effect<sup>(20)</sup> makes broadband sound reduction difficult<sup>(9)</sup>.

Overcoming these problems requires elimination of the time delay element. To do this, we propose a type of FB-ANC using the Smith predictor<sup>(13), (14)</sup>, an effective control technique for chemical and other plants where time delay elements are predominant.

As shown in Fig.6, the Smith predictor takes the approach of designing a control system by canceling the plant's time delay element to create a closed loop system free of any time delay. In contrast, FB-ANC treats noise as a disturbance, which is added after the plant, and is considered a disturbance-driven approach because of the absence of command input. Thus, the Smith predictor configuration shown in Fig.6 cannot be directly applied to FB-ANC.

However, we have made it possible to apply the Smith predictor to FB-ANC by doing the following. The design specifications of FB-ANC are intended to stabilize a closed loop system and reduce sensitivity function  $S(s)$  in the frequency band to control (i.e., the transfer function from

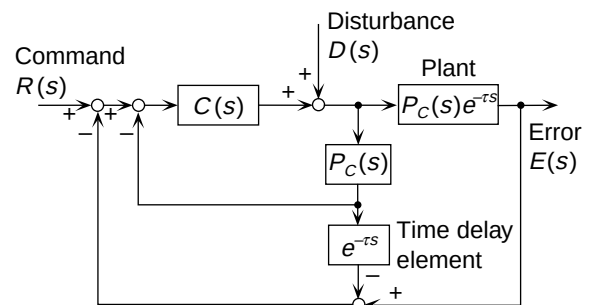


Fig.6 The Smith predictor

noise  $D(s)$  to error  $E(s)$ ). With the configuration in Fig.6, the sensitivity function when disturbance is added after the plant is determined with this equation:

$$S(s) = \frac{1 + P_C(s)C(s)\{1 - \exp(-\tau s)\}}{1 + P_C(s)C(s)} \quad (8)$$

Thus, the plant's time delay element is reflected in the sensitivity function. To prevent the time delay element from being reflected in the sensitivity function, the authors have proposed a Smith predictor-type FB-ANC configuration, shown in Fig.7, in which the sensitivity function is:

$$S(s) = \frac{1}{1 + P_C(s)C(s)} \quad (9)$$

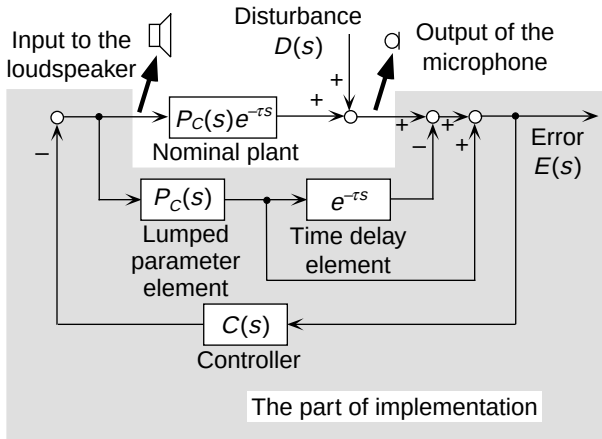


Fig.7 The proposed Smith predictor type FB-ANC for a nominal plant

With this configuration, design of the controller to the plant's lumped parameter part  $P_C(s)$  is possible with equation (9).

The Smith predictor is interpretable within the Internal Model Control (IMC) framework whose application in FB-ANC we have proposed previously<sup>(9),(21)</sup>. The Smith predictor is characterized by its facilitating of control system design using a  $P_C(s)$  from which the plant's time delay element has been removed.

## 5. The PID Controller Design Method

This section discusses the structure of, and design method for, controllers. With respect to controller structure, focus is placed on PID control<sup>(22)</sup> as a way of achieving low-cost ANC. Also known as three-term control, PID control is characterized by a simple structure comprising proportional control, integral control, and derivative control. PID control is used quite extensively by the industry, but no cases can be found of its application in ANC. Therefore, this section discusses the design of FB-ANC that uses a PID controller.

The design method used by the authors is a PID controller design method that was proposed by Saeki and which enables the systematic determination of the three design parameters of PID control<sup>(23)</sup>.

Stated simply, the actual design process begins by using the following equation to determine one of the PID control parameters  $k_i$  from the control constraints of the sensitivity function in the low frequency band.

$$C(s) = k_p + k_d \frac{s}{1 + T_s} + k_i \frac{1}{s} \quad (10)$$

Next, the following parameter sets concerning  $k_p, k_d$ , which satisfy the nominal performance conditions (i.e., noise reduction as nominal performance) and robust stability conditions (i.e., stability conditions that account for uncertainty in the plant model), are solved for each frequency.

$$K_S(\omega) \equiv \{[k_p, k_d] \mid |S(j\omega)| < \alpha(\omega)/|W_S(j\omega)|\} \quad (11)$$

$$K_T(\omega) \equiv \{[k_p, k_d] \mid |T(j\omega)| < \beta(\omega)/|W_T(j\omega)|\} \quad (12)$$

The elements of the set of the intersection of the sets in equations (11) and (12)

$$K(\omega) \equiv K_S(\omega) \cap K_T(\omega) \quad (13)$$

are now the PID control parameters that satisfy the design specifications. From the elements of the set in equation (13), the parameters are selected and the robust PID controller is determined.

## 6. Verification of Control Performance

### 6.1. Validity of the Proposed Smith Predictor-type FB-ANC

To determine the validity of the proposed Smith predictor-type FB-ANC, we used computer simulation to predict controlled sound pressure. The experiment shown in Fig.8 simulated road noise. Noise inside the vehicle was measured and used as noise data for simulation, which was performed with a discrete time series and a sampling frequency of 10 240 Hz.

The plant used was the physical model discussed in section 2. The PID controller designed for the plant's lumped parameter portion  $P_C(s)$  was used as controller  $C(s)$ , shown in Fig.7. Given  $a=0.58$  and  $b=0.75$ , the parameters were selected from parameter set  $K$ , which satisfies the nominal performance conditions and robust stability conditions, and the PID controller was determined with the following equation.

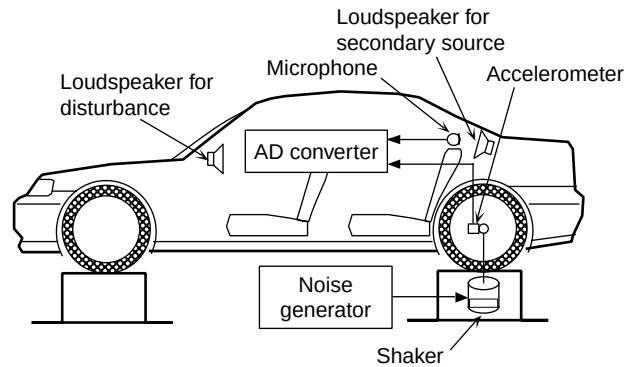


Fig.8 Data acquisition experiment simulating road noise in the cabin

$$C(s) = 0.2125 + \frac{s}{1 + s/(2\pi \cdot 500)} (1 \times 10^{-6}) + \frac{900}{s} \quad (14)$$

Fig.9 shows the sensitivity function, which was used as an indicator for predicting noise reduction performance during control; Fig.10, the results of sound pressure prediction simulation. A noise reduction of 4 to 7 dB is predicted from Fig.9. In actuality, the results of simulation indicate a 4 to 7 dB noise reduction in the broad band from 40 to 500 Hz, as Fig.10 shows. There was a marked reduction of 7 dB in the 100-200 Hz band. An equivalent noise reduction effect was also verified when the FB-ANC composition was applied to lumped parameter portion  $P_c(s)$  of the nominal plant. This verified the validity of the PID controller designed using the physical model, and the validity of the proposed Smith predictor-type FB-ANC configuration.

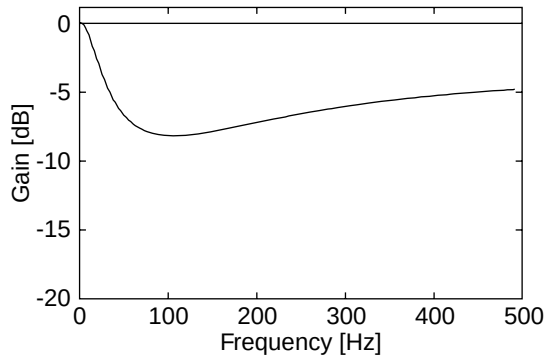


Fig.9 Sensitivity function for a physical model

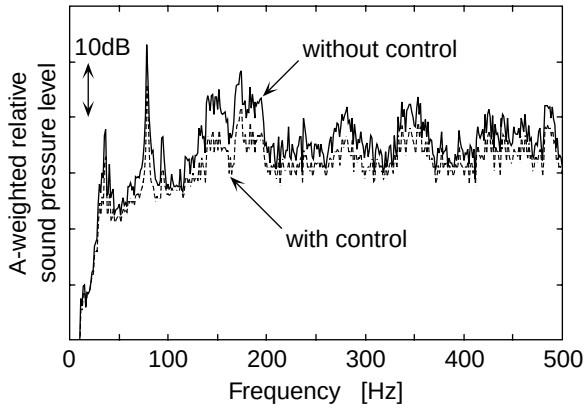


Fig.10 Sound pressure level for a physical model

## 6.2. Controller Based on a Gray-box Model

Next, the control performance of the controller designed with the gray-box model is compared with that of the controller designed with a physical model as described in the previous section. The transfer function of the PID controller designed with the gray-box model is represented by the following equation.

$$C(s) = 0.15 + \frac{s}{1 + s/(2\pi \cdot 500)} (1 \times 10^{-5}) + \frac{700}{s} \quad (15)$$

The sensitivity function is shown in Fig.11; the results of the simulation of sound pressure prediction, in Fig.12. These results demonstrate that noise reduction in the 100-200 Hz band is superior to that of the controller designed with the physical model. This confirms the validity of plant models obtained with gray-box modeling with respect to noise reduction performance.

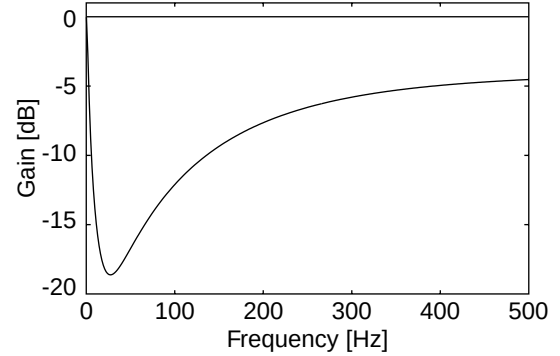


Fig.11 Sensitivity function for a gray-box model

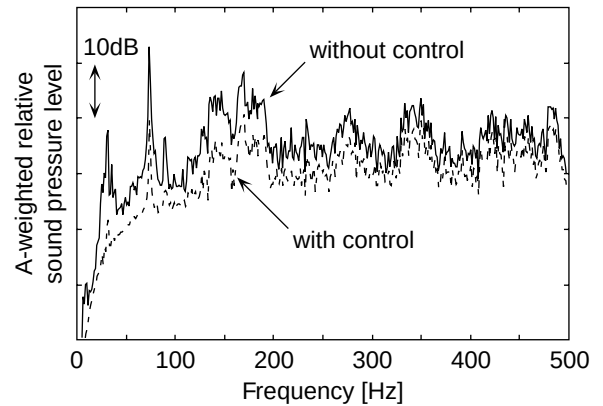


Fig.12 Sound pressure level for a gray-box model

## 7. Conclusion

Focusing on FB-ANC as a means of constructing low-cost ANC systems, this paper yields the following conclusions through the low-order modeling of a plant, and by designing a simple-structure controller with the results of said modeling:

- 1.) By using gray-box modeling as a means of high-precision low-order modeling, the plant is expressed with a second-order series and a time delay element.
- 2.) Smith predictor-type FB-ANC system eliminating time delay is proposed for controller design.
- 3.) By applying PID control to ANC, a simple, robust PID controller taking into account the uncertainty of time delay is designed. The validity of the plant model obtained and the controller designed is verified by predictive simulation of control performance using signals obtained from an actual vehicle.

We next plan to verify the proposed method's validity by implementing in an actual vehicle.

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