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ACTIVE NOISE CONTROL SYSTEM FOR LARGE AXIAL FLOW FANS: CONTROL ALGORITHM FOR HIGH ORDER MODES AND INFLUENCE OF DUCT VIBRATION

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ABSTRACT
To control n transverse modes (including the plane wave) in a two-dimensional duct, a control algorithm using $4n$ sensors and n secondary sources was developed. The controller detects the forward propagating wave of each mode in a duct and minimizes them individually using the Multiple Error Filtered-x LMS algorithm. This active control system is available to be set at any position along a duct as it controls only forward propagating waves. In the case that 4 transverse modes exist in the duct, the proposed system reduces each mode by 13 to 28 dB in simulation. Also, it is found experimentally that the noise attenuation value is decreased in the case that the vibration of the duct wall is intense when using an active noise control. In this paper, the method for applying the proposed algorithm to the NZ component of axial flow fans in a large rectangular duct is explained.

1 - INTRODUCTION
In order to reduce the noise of the NZ component of large axial flow fans by using an active noise control (ANC) system, it is very important to consider the influence of high order mode propagation. In an actual power plant, for example, where the frequency of the NZ component is 193 Hz and the cross section is 8.1 m $\times$6.0 m, the NZ component propagates including not only the plane wave but also higher order propagating modes. As the noise control system for the high order propagating modes in a three-dimensional waveguide becomes very complex with the increase of the number of modes, then applying a control system for a two dimensional waveguide is practical after dividing the sectional area of the waveguide into several subsections at the controlling area.
This paper proposes an ANC system to reduce the forward propagating waves using the Multiple Error Filtered-x LMS (MEFX-LMS) algorithm [1], $4n$ sensors and n secondary sources. This system may be collocated at any position along a duct as it controls only the forward propagating waves. By a numerical simulation, its noise reduction for 4 propagation modes (including the plane wave) is $13 - 28$ dB. One more important point is the influence of vibration of duct panels on the ANC, as the actual wall is not rigid. An experiment shows that when the vibration level of the panel was increased, the noise reduction is decreased to 17.1 dB, however, it is 26.6 dB when that level is low. Finally, there is a causation of the decrease of the noise reduction of 26.6 dB with a rigid wall was decreased to 17.1 dB by a vibrating wall.

2 - NOISE REDUCTION METHOD FOR THE ACTUAL LARGE DUCT
Usually an ANC system is set at a duct outlet, however in the actual plant, the maintenance of ANC system is difficult and the duct lagging is necessary because the duct outlet is in an inaccessibly high position. Then we propose to set the ANC system near the noise source in the duct for easy maintenance and reduction of duct lagging costs. In order to set an ANC system at any position in the duct, it is important to control the forward propagating waves in the subsection of the duct. Here, we considered a system as the 0th $-$ the 3rd modes propagating in the duct. 16 sensors and 4 secondary sources are needed to detect and control the forward propagating wave of each mode.

The control block and the arrangement of sensors and sources are shown in Fig. 1. The 4 secondary sources are controlled by using reference and error signals to reduce each forward propagating mode in the subsection. The control algorithm is the MEFX-LMS algorithm and the reference and error signals are the forward propagating waves of each mode. To detect the forward propagating waves, 8 sensors are used. Sound pressure $P_m(f)$ of the m-th sensor in the subsection can be expressed as

$$P_m(f) = \sum_{n=0}^{3} \left(P_{fn} e^{-jk_{z,n} z_m} + P_{bn} e^{jk_{z,n} z_m} \right) \tag{1}$$

where f is the frequency. P_{fn} and P_{bn} are respectively the complex amplitudes of the forward and backward propagating waves of the n-th order. $k_{z,n}$ is the wave number of the n-th mode on the z axis. z_m is the z coordinate of the m-th sensor position on the duct wall. As the number of sensors is 8, the sound pressures $P_m(f)$ are given as follows:

$$\begin{bmatrix} P_1(f) \\ P_2(f) \\ \vdots \\ P_8(f) \end{bmatrix} = \begin{bmatrix} e^{-jk_{z,0}z_1} & \cdots & e^{-jk_{z,3}z_1} & e^{jk_{z,0}z_1} & \cdots & e^{jk_{z,0}z_1} \\ \vdots & & \vdots & \vdots & & \vdots \\ e^{-jk_{z,0}z_8} & \cdots & e^{-jk_{z,3}z_8} & e^{jk_{z,0}z_8} & \cdots & e^{jk_{z,0}z_8} \end{bmatrix} \begin{bmatrix} P_{f0}(f) \\ \vdots \\ P_{f3}(f) \\ P_{b0}(f) \\ \vdots \\ P_{b3}(f) \end{bmatrix} \tag{2}$$

The sound pressures $P_m(f)$, the wave numbers $k_{z,n}$ and the coordinate z_m of the m-th sensor are given. Then each complex amplitude of the n-th mode is obtained by solving the equation (2). These complex amplitudes are used as the reference and error signals. The evaluation function is given as the mean square of the error signal and MEFX-LMS algorithm works to minimize the evaluation function. Even if the number of modes is greater than 4, the control method is almost the same.

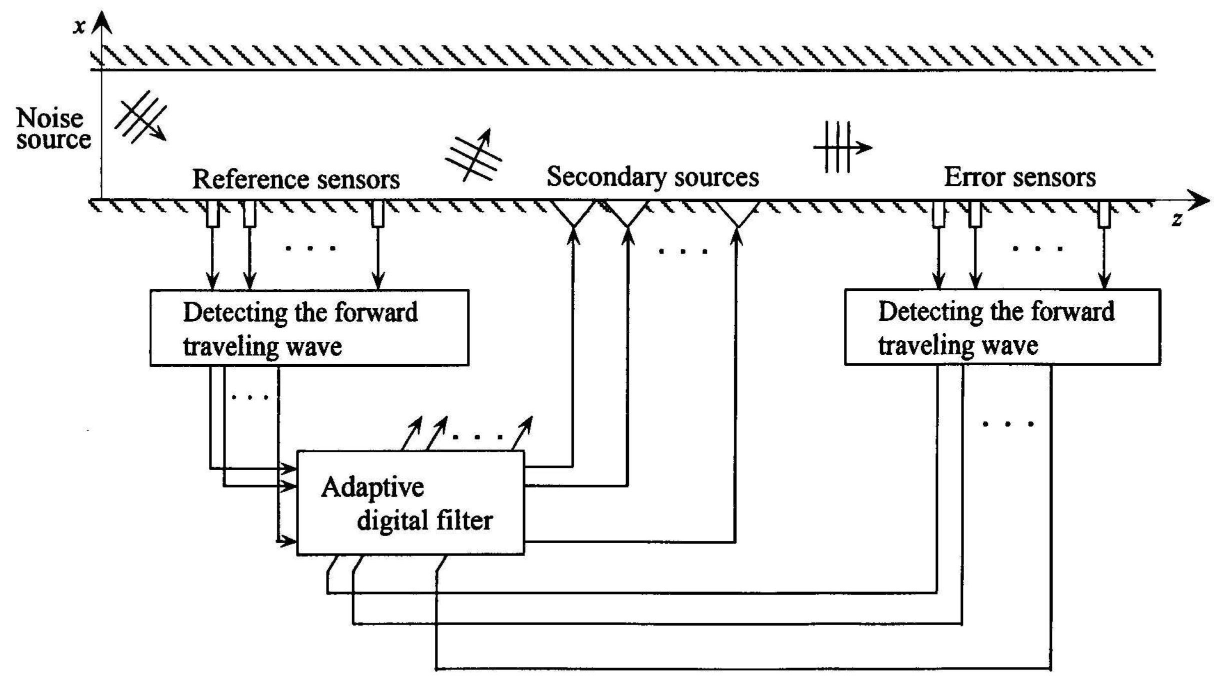

Figure 1: The control block and the arrangement of the sensors and secondary sources.

3 - NUMERICAL SIMULATION AND EXPERIMENT

The numerical simulation model is the same as Fig. 1. The size of each of the subsectional cross sections is 0.9 m × 3.0 m and the frequency is 193 Hz (the NZ component). In the subsection of the duct, the 0th − the 3rd transverse modes exist. Then 16 sensors and 4 sources are set on the duct wall along the z-axis with a sensor spacing of 1.3 m and a secondary source spacing of 7.1 m. The NZ component is reduced by using MEFX-LMS algorithm. The sampling frequency is 4 kHz. The simulation results denote that when number of steps of the iteration increases, the reduction value becomes large to 13 − 28 dB, as shown in Fig. 2.

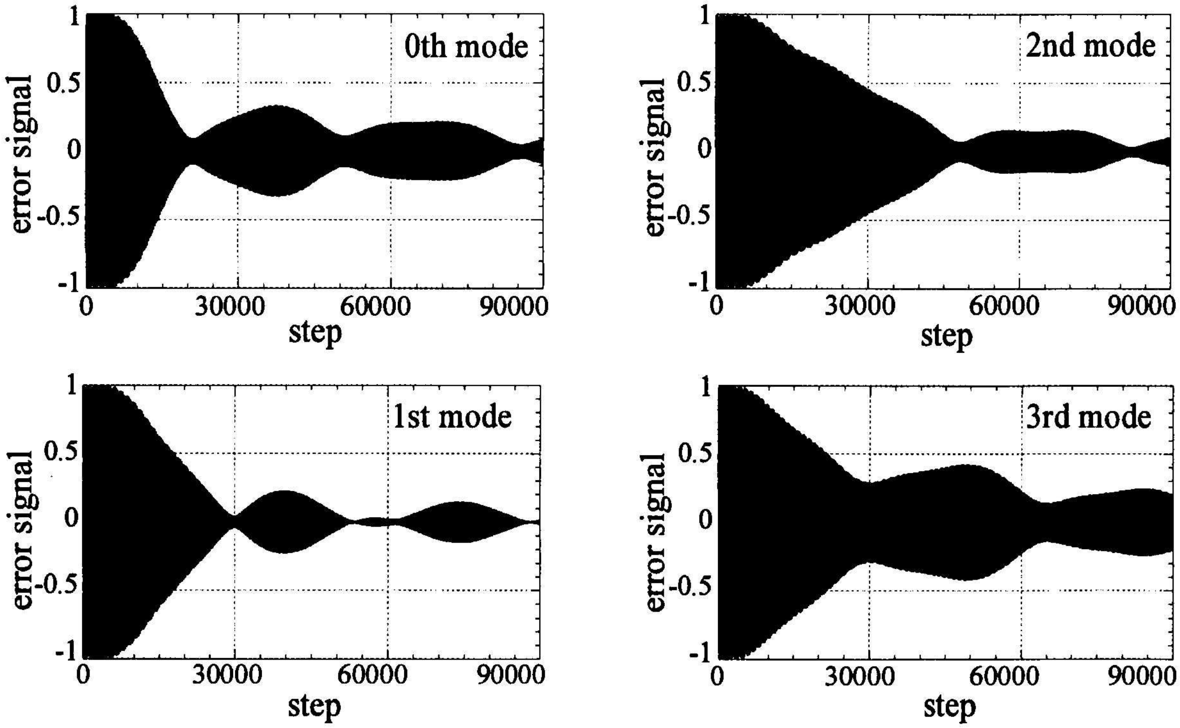

Figure 2: Error signals obtained by a numerical simulation.

To confirm the efficacy of the proposed algorithm, an experiment of a one-dimensional duct is carried out as the first step. This duct is a 1/3 scale model of which the cross section size is 0.3 m × 1.0 m and the noise frequency is 100 Hz. The primary source is a general speaker. The transverse mode in this duct is only the 0-th mode (plane mode), then 4 sensors and 1 secondary source is set as shown in Fig. 3.

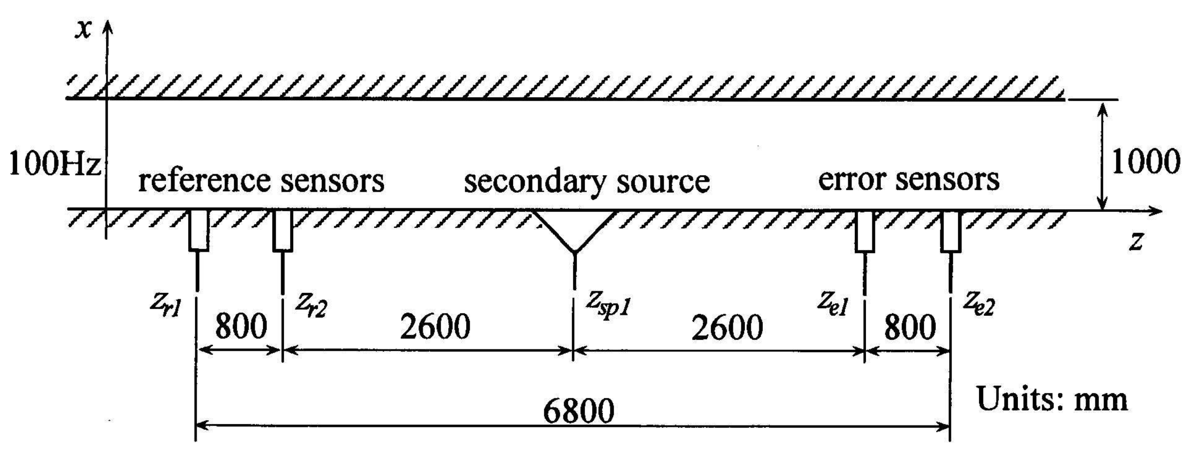

Figure 3: The arrangement of the sensors and sources for an experimental duct.

Experimental results are shown in Fig. 4. According to Fig. 4, the frequency component of 100 Hz is reduced by 26.6 dB using the proposed algorithm. The efficacy of the proposed method is proven by this simulation and experimental result.

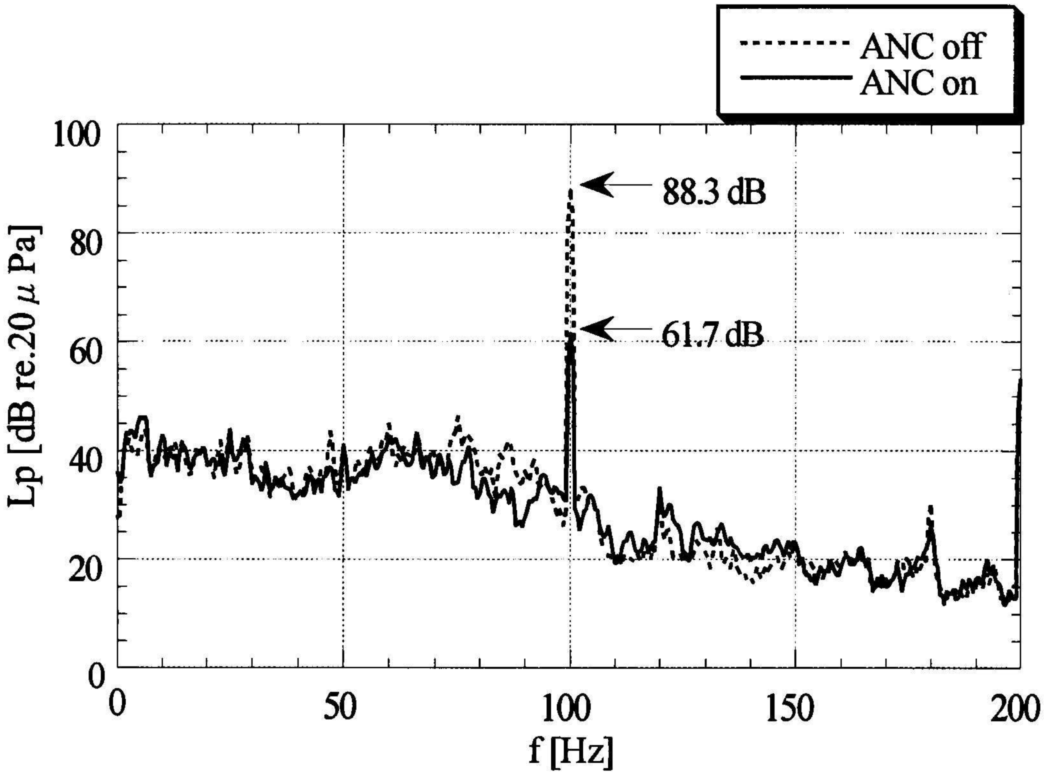

Figure 4: Auto power spectrum of sound pressure at the duct outlet.

4 - INFLUENCE OF DUCT VIBRATION

Usually the duct wall is assumed to be rigid and noise wave to be traveling from the noise source only, but if the duct wall is not rigid, the traveling wave is different. Here, the influence of the duct wall vibration is investigated experimentally. The experimental model is the same as in Fig. 3. The influence of duct vibration is considered whether the vibration generated by the primary source is isolated or not. The sound pressure along the x-axis in the duct is shown in Figs. 5 and 6. Both figures show the difference of sound pressures propagated from the primary source and the secondary source in ratio to sound pressure P_0 as the reference value. P_0 is the sound pressure on the sensor position ($x = 1{,}000$ mm). (a) is the difference of sound pressure level and (b) is the difference of phase. According to Fig. 5, the sound pressures propagated from the primary source and the secondary source are almost same, but Fig. 6(b) shows the difference of the primary and the secondary is large, especially about 17 degrees at $x = 0$. The noise control algorithm works to minimize the sound pressure at the sensor position ($x =$ 1,000). Then the reduction level of noise is decreased to 17.1 dB when duct vibration is large as in Fig. 7. It is found that minimizing the duct wall vibration is important to reduce the noise level efficiently.

5 - CONCLUSION

In this paper, an ANC algorithm for a large axial fan duct is proposed. The proposed algorithm is shown to be efficient by both numerical simulation and experiment. Also, it is found the influence of duct vibration to the noise reduction, therefore it is important that the duct vibration must be small or a new algorithm considering the duct vibration is necessary.

REFERENCES

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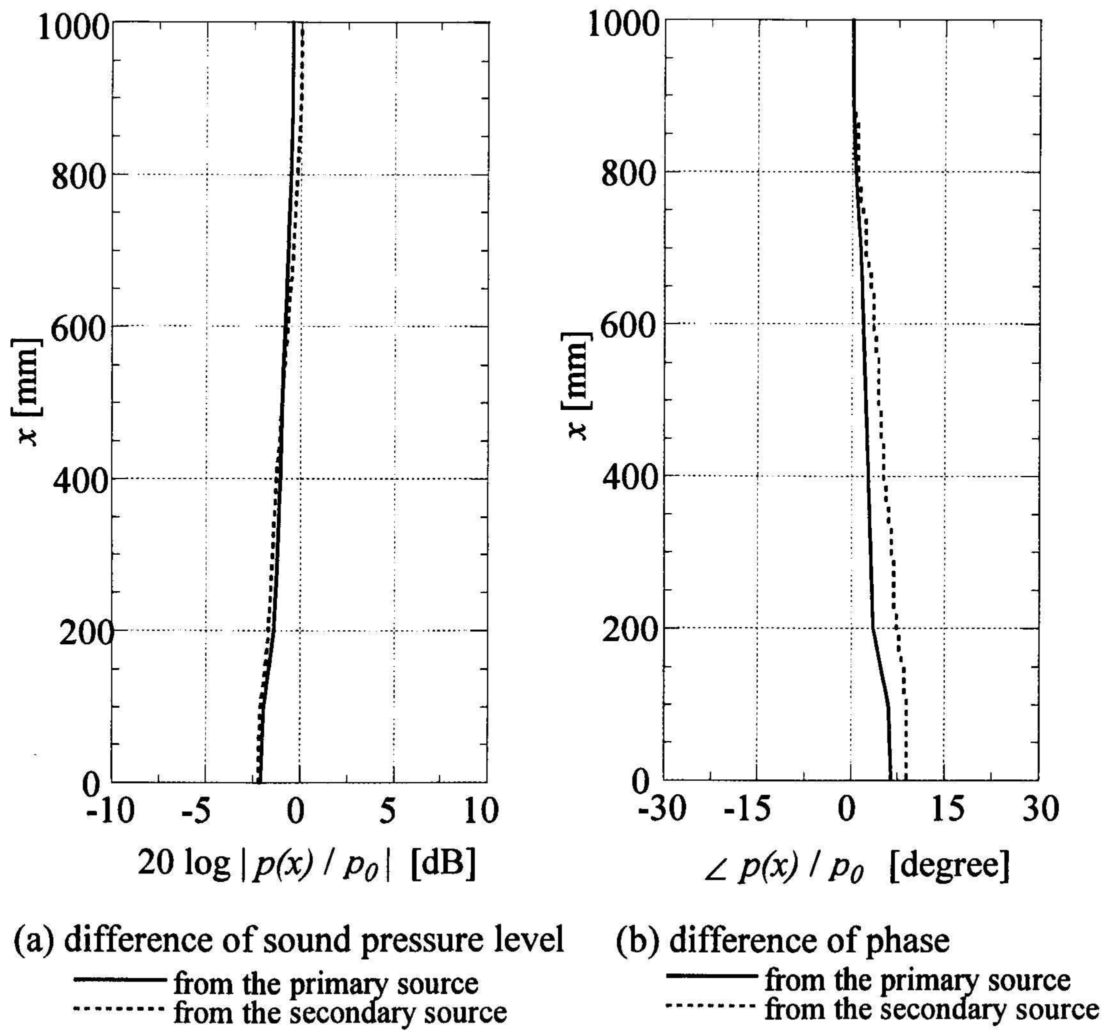

Figure 5: Difference of sound pressure distribution in the duct (with vibration isolation).

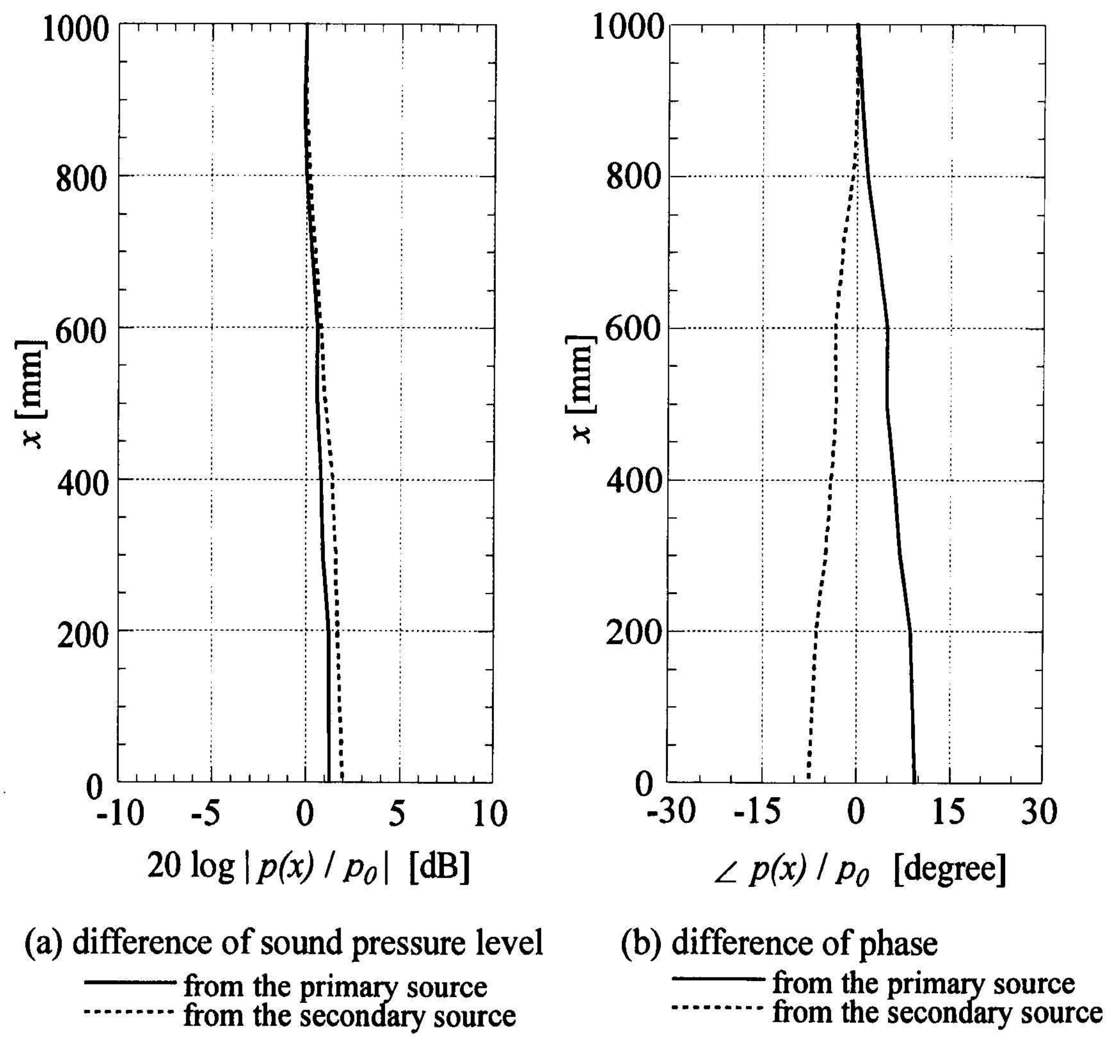

Figure 6: Difference of sound pressure distribution in the duct (without vibration isolation).

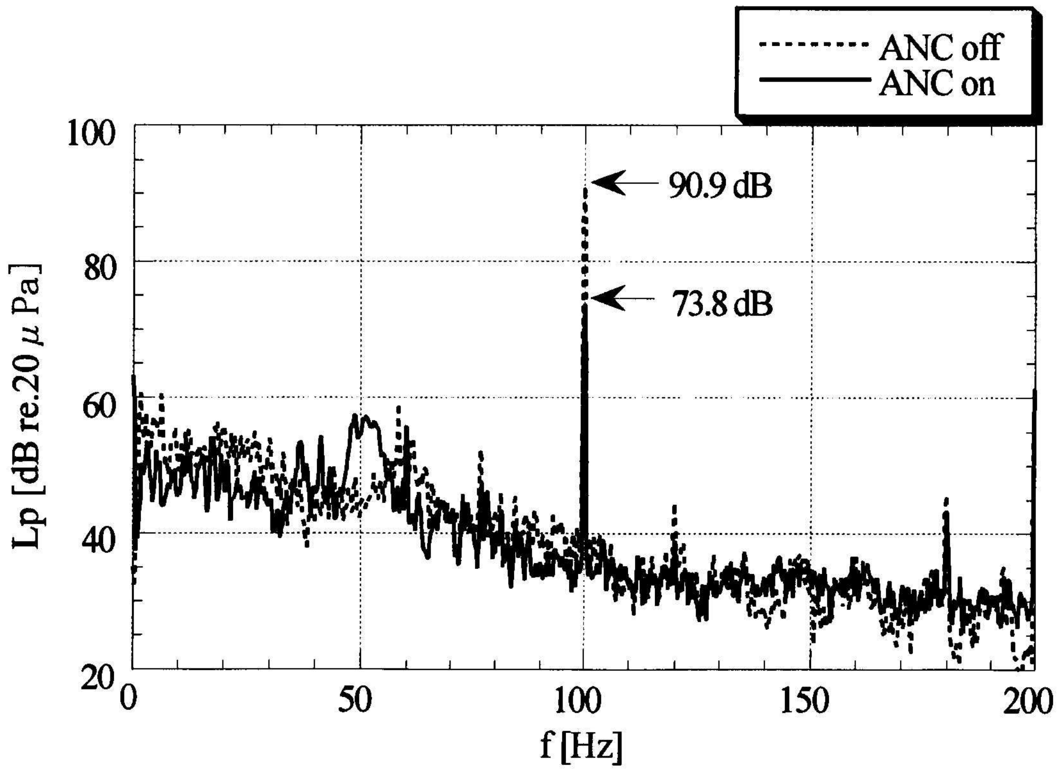

Figure 7: Auto power spectrum of sound pressure at the duct outlet (without vibration isolation).