EFFECTS OF SLITS OF CASING ON AEROACOUSTIC NOISE RADIATED FROM AN AXIAL FAN

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Recently, reducing the aerodynamic noise of axial cooling fans used in electronic equipment has become necessary. In order to improve the performance of the small axial flow fan for electronic equipment, there is a method of installing slits channel in the casing. Although this countermeasure improves fan performance, noise may become large in some cases. In order to clarify the effects of the slits of the fan casing on the aerodynamic noise of axial fans, numerical simulations based on the incompressible Navier-Stokes equation and experiments were conducted. The numerical prediction of the aerodynamic sound was also performed based on the Curle’s equation. In the case with slits, more structured tip leakage vortices near the rotor tip on the suction surface were observed and the wake of the blades was more turbulent compared with the case without slits. These vortices are developed in the circumferential direction, thus impinging on the pressure surface of the adjacent blade and struts. As a result, the periodic pressure fluctuations occur on the struts, which lead to the acoustic radiation. Both the sound pressure spectra predicted by the present decoupled simulations and the direct aeroacoustic simulation have peaks at the blade passing frequency and its harmonic frequency. These results indicate that the slits of the casing affect the wake of the blades, the interference of the wake with the struts and aeroacoustic radiation.

Keywords: Fan noise, Large eddy simulation, Direct aeroacoustic simulation, Casing shape, Tip leakage vortex

1. Introduction

Small axial flow fans are widely used as cooling fans installed in electronic equipment such as personal computers. Recently, the downsized fan has been developed for small electronic equipment. In order to keep the cooling performance, the rotation speed needs to be increased. However, increasing aerodynamic noise radiated from small fans becomes a problem. Thus, developing the fan with good performance in both the aerodynamics and the aeroacoustics is necessary in industrial applications.

Several reports have been published by numerous researchers about the radiation mechanism of the aerodynamic noise generated from the axial fan. Jang et al. [1] studied three-dimensional flow structure near the blade tip of semi-open type propeller fan using laser Doppler velocimeter (LDV) system and the large eddy simulation (LES). Their results revealed that the tip vortex formed by the rolling-up of the suction surface boundary layer is so strong that it dominates the flow field near the tip. Fukano et al. [2] studied the relation between the relative flow field near the blade tip of the axial fan and the aerodynamic noise, in a low flow rate region. Their results revealed that the interference between the tip leakage vortex and the adjacent blade caused high pressure fluctuations on the pressure surface of the adjacent blade under the flow rate condition and it increases the sound pressure level.
However, these reports are related to large axial flow fans with a diameter of 300 mm or more, and the internal flow of the smaller fans is different. Takayama et al. [3] simulated propeller fan with a diameter of 182 mm using LES and predicted aerodynamic noise radiated from the fan using a decoupled method of incompressible flow simulations and the Curle’s equation [4]. As a result, they indicated that it is possible to capture the characteristic of the broadband noise by computations with high-density grid near the wall of the blade surface. Ito et al. [5] conducted experiments for the small axial flow fan with diameter of 85 mm. They clarified that tilting struts in the counter rotational direction increases the efficiency and reduce the aero-dynamic noise radiated from fans. Lim et al. [6] studied the characteristics of unsteady flow field and aeroacoustic noise of a small axial flow fan with diameter of 76 mm. Their results indicated that the struts of the casing are one of the most important parameters of noise generation. Since the outer shape of the small axial flow fan is often rectangular, a part of the bell mouth is not round but it has a flat part. According to the above document, the flat portion of the bell mouth is also contributed to the acoustic radiation.

Also, to improve the performance of the above-mentioned small axial flow fan, there is a method of installing slit channels in the casing. However, this countermeasure may increase the noise in some cases. In this paper, to clarify the effects of the slits of the fan casing on the aerodynamic noise of axial fans, numerical simulations based on the incompressible Navier-Stokes equations and the experiments were conducted. The aerodynamic noise analysis was calculated by the Curl’s equation [4]. The predicted sound pressure level was compared with the results of the direct aeroacoustic simulations based on the compressible Navier-Stokes equations and the experiments [7].

2. Flow Configuration

The tested axial flow fan is shown in Fig. 1(a). Specifications of the fan are summarized in Table 1. The origin of the coordinate was set to the centre of the hub end face. The streamwise, vertical and spanwise directions are the x, y, z-axis, respectively. The number of blades was \( Z_b = 5 \) and the four struts are installed in the downstream of rotor blades. The fan diameter was \( D = 40 \text{ mm} \). The hub/tip ratio was 0.45. Blade chord length at rotor tip was \( C = 22.3 \text{ mm} \). The computations were performed under the flow rates, \( Q = 0.01, 0.07, \) and \( 0.11 \text{ m}^3/\text{min} \), where \( 0.11 \text{ m}^3/\text{min} \) is corresponds to the maximum flow rate. The rotation speed was \( n = 3025 \text{ rpm} \), and the blade passing frequency (BPF) was 252 Hz. The Reynolds number based on rotor tip velocity and chord length at rotor tip, \( Re_c = \frac{V tc}{\nu} \), was \( 9.37 \times 10^4 \).

The fan casings with and without slits used in this study are shown in Figure 1(b), (c). The tip clearance between the casing inner wall and the rotor tip was 1 mm.

![Figure 1: Tested fan for analysis](image-url)
Table 1: Specifications of tested fan

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fan diameter $D$ [mm]</td>
<td>40</td>
</tr>
<tr>
<td>Hub diameter $d$ [mm]</td>
<td>18</td>
</tr>
<tr>
<td>Hub/tip ratio $d/D$</td>
<td>0.45</td>
</tr>
<tr>
<td>Rotation speed $n$ [rpm]</td>
<td>3015</td>
</tr>
<tr>
<td>Rotor tip velocity $V_t$ [m/s]</td>
<td>6.3</td>
</tr>
<tr>
<td>Chord length (at rotor tip) $C$ [mm]</td>
<td>22.3</td>
</tr>
<tr>
<td>Reynolds number $Re_c$ ($= V_t C/\nu$)</td>
<td>$9.37 \times 10^4$</td>
</tr>
<tr>
<td>Number of blades $Z_b$</td>
<td>5</td>
</tr>
<tr>
<td>Number of struts</td>
<td>4</td>
</tr>
<tr>
<td>Tip clearance [mm]</td>
<td>1</td>
</tr>
</tbody>
</table>

3. Experimental Methodologies

The performance and aerodynamic noise measurements of the axial flow fan driven by AC motors were conducted in an anechoic chamber room. The fan performance was measured for the fan attached to the front of the plenum chamber as shown in Fig. 2(a).

As shown in Fig. 2, the aeroacoustic experiments were conducted with and without a plenum chamber to clarify the effects of those conditions on the sound pressure level. The sound pressure level was also measured by the microphone installed at a distance 0.1 m far from the rotation axial of the fan as shown and the signal of the noise was analysed by using an FFT analyzer.

![Figure 2: Experimental setup to measure fan noise and performance.](image)

4. Computational Methodologies

4.1 Large Eddy Simulation

Large eddy simulations were conducted to investigate the three-dimensional structure and unsteady nature of the vortical flow in the axial flow fan. The simulations were performed using finite element analysis software (FrontFlow/blue) with governing equations of three-dimensional incompressible Navier-Stokes equations. The Dynamic Smagorinsky Model [8] was used as a sub-grid scale model. The computation of pressure and velocity fields was performed using a fractional step method along with the Crank-Nicolson implicit time integration scheme.

4.2 Computational Grids

The side views of the computational mesh and perspective view of the computational model and boundary conditions are shown in Figs. 3 and 4. The computational domain is composed of the upstream region, fan neighbourhood region and downstream region. A baffle plate with the thickness of 2 mm was assumed between the upstream and downstream regions. The spanwise and vertical
extents of the computational domain were $4D$. The streamwise extents of the upstream and downstream region were $2D$ and $4D$, respectively. The flow in the fan neighborhood region was solved in the rotational coordinate. The interaction between the rotating and stationary regions was taken into account by dynamically oversetting the grids with multiple frames of reference [9]. The values for static pressure and velocity components in the margins were transferred between corresponding neighbouring elements for each time step.

To capture the boundary layer, prism mesh were piled up in a direction perpendicular to the surface of the blades. The resolution of the prism mesh near the wall was 0.05 mm, which is corresponding to $\delta_{10}$ and satisfy $y^+ < 5$ at the first boundary layer. The total number of grid points was approximately $7.76 \times 10^7$ and $8.00 \times 10^7$ respectively in the case without and with slits.

**4.3 Boundary Conditions**

The boundary conditions of the computation are also shown in Figure 4. A uniform distribution of the velocity was chosen for the upstream boundary of the domain. The inflow was assumed to be non-turbulent. At the outlet boundary of the downstream region, the fluid traction was set to be zero in all three directions. For the surface of the duct, casing, hub, blades, and the baffle plate, no-slip boundary conditions were set. As mentioned above, the flow around the fan rotor is analyzed in the rotational frame of reference, thus a moving wall boundary was attributed to the surface of the rotating region.

**5. Comparison of Computational Methods with Experiments**

The measured and predicted performance curves of fans with and without slits tested in this study are shown in Fig. 5. The simulations were conducted at three points of a shut off point, a design point and a maximum flow rate point. The predicted static pressure rising was in good agreement with the measured value. This means that the present computations can capture the performance of the fan correctly.

The predicted sound pressure spectra with slits at the maximum flow rate were compared with the results of measurement and direct aeroacoustic simulations as shown in Fig. 6. The measured sound pressure spectra with a plenum chamber present that the sound pressure level becomes intense particularly at low frequencies around the blade passing frequency (BPF). This is possibly due to the acoustic effects of the chamber. Also, the spectrum by the direct aeroacoustic simulations has a similar floor level, where the simulations were performed for the fan in the duct like the present computational domain.

Also, the sound pressure level near the BPF ($nZ_0$) and the floor level predicted by the present simulation are close to the measured data without a chamber. This is because the present decoupled
method is based on the Curle’s equation and the acoustic radiation into the free space is assumed. The predicted result shows that intense tonal sound occurs at the blade passing frequency and its harmonic frequencies particularly of $4nZ_b$. This feature is consistent with the measured data.

6. Results and Discussion

6.1 Effects of slits on flow and sound

Figure 7 shows the comparison of the predicted sound spectra for the cases with and without slits. The predicted tonal sound at the frequency of $nZ_b$ and $4nZ_b$ for the case with slits becomes more intense compared with the case without slits. These sound pressure spectra are based on the aerodynamic force on the fan and struts. Therefore, it is indicated that the slits affect the pressure fluctuations on the fan and struts.

6.2 Vortical structures

Figure 8 shows the iso-surface of the second invariant at $d/T_{rev} = 12.86$, where the wake of the blade interferes with the strut and the pressure near the strut becomes lowest. The contour was colored with the vorticity magnitude. In the case without slits, the tip leakage vortex is developed toward the pressure surface of the adjacent blade. On the other hand, in the case with slits, the tip leakage vortex is convected to the trailing edge of the adjacent blade. Also, the vortices interfere with the flow by the flow from the struts. As a result, the tip leakage vortices become smaller vortices, and the more fine-scale vortices appear in the wake. These fine vortices are collided with the struts, therefore, the pressure fluctuation occurs extensively on the struts as discussed in the next section.
6.3 Pressure fluctuations

The RMS of the pressure fluctuations on the fan and casing surface are shown in Fig. 9. The local intense pressure fluctuations at the end of the struts and inner wall of the casing were observed. The time histories of the static pressure at five points (10 - 90% between the boss and the casing) on a strut as shown in Fig. 10 are shown in Fig. 11. It found that the static pressure was decreased at the time for the interference of the wake with the strut of $t/T_{rev} = 12.86$. Possibly, the above-mentioned leakage vortices affect these pressure fluctuations. In the case with slits, the pressure fluctuations at the points 50% and 70% was particularly more intense compared with that of the case without slits.

Figure 12 shows the time history of the pressure at four sampling points between boss of the fan and struts also as shown in Fig. 10. The pressure fluctuations around all the struts have the local maximum at approximately the same time such as $t/T_{rev} = 12.76$. This local maximum periodically occurs at the frequency of $nZ_b$ as shown in Fig. 12. Also, the level of the fluctuations is higher for the case with slits compared with the case without slits. This result is consistent with the higher sound pressure level at $nZ_b$ for the case with slits.
The positions of the struts and blades are shown in Fig. 13, where the static pressure distributions at cylindrical surface (60%) are shown. The wakes from all the blades pass between the struts and do not intensely interfere with the struts. Also, the subsequent time of $t/T_{\text{rev}} = 12.86$, the wakes interfere with the struts near point A. As a result, the pressure becomes relatively lower as mentioned in the previous subsection.

Figure 11: Time histories of the pressure at five points shown in Fig. 10 on the surface of the strut

Figure 12: Time history of the pressure at four points between the boss of fan and struts (60% $D$)

Figure 13: Static pressure distribution at cylindrical surface (60%)
Also, it is found that the fluctuations of static pressure at the points A and D were more intense than the points B and C. This trend was shown in both the cases with and without slits. It is because the wakes are relatively close to points B and C at this time of $t/T_{rev} = 12.76$ compared with other struts. As a result, the intensity of the pressure fluctuations on the struts differs between the struts although the struts are symmetrically arranged along the rotational axis. These phenomena affect the difference in the tonal sound pressure level at the BPF and the harmonic frequencies.

7. Conclusions

In order to clarify the effects of the slits of the casing on aeroacoustic noise radiated from an axial fan, unsteady incompressible flow simulations were performed. The tonal sound pressure level at the blade passing frequency becomes more intense for the case with slits compared with the case without slits. In the case with slits, the tip leakage vortex is convected to the trailing edge of the adjacent blade and fine vortices were formed between the fan and the struts. The fine vortices are collided with the struts, causing the pressure fluctuations on the struts. At the time for the interference of the wake of the blade with the strut, the static pressure around the strut becomes decreased. Meanwhile, the pressure fluctuations around all the struts have a local maximum at the time when all the wakes pass between the struts. These pressure fluctuations are related with the difference in the tonal sound pressure level at the blade passing frequency and the harmonic frequencies.

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REFERENCES


