Relationship between Internal Flow and Fan Noise of Cross Flow Fan

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Abstract
The cross flow fan has an eccentrically located vortex inside impeller. The vortex behavior has a large effect on the fan performance and fan noise. Although investigations on the internal flow of cross flow fan have been performed by many researchers, quantitative relationship between the eccentrically located vortex and fan noise is not sufficiently made clear. In our previous study, we developed a noise reduction method of cross flow fan by using a step tongue and a skew tongue. Unfortunately, a detailed mechanism of fan noise reduction is not known yet. In this paper the flow pattern and the fan noise of cross flow fan are experimentally and numerically examined.

Key words
Cross Flow Fan, Internal Flow, 3-D Numerical Simulation, Fan Noise, Rotation Noise Level

1. Introduction
In previous numerical simulation studies [1-3], cross flow fan performance and noise level were examined. In a very recent analysis, Jeon and Cho [1] have shown through unsteady CFD and unsteady aeroacoustic pressure calculations that “the interaction between the stabilizer and rotating impeller generates tonal sound. Also, it is found that the trailing edge of blade generates more acoustic pressure than the leading edge.” Meanwhile, Fukano et al. [4-6] examined the effects of impeller, casing and tongue configurations on the noise performance of cross flow fan. Hayashi et al. [7] revealed that the fan noise could be reduced by changing the blade shape. Lee et al. and Hyoung [8, 9] conducted experiments on the effects of blade and tongue configuration on noise reduction.

In our previous study, we developed a noise reduction method of cross flow fan by using a step tongue and a skew tongue [10, 11]. Unfortunately, a detailed mechanism of fan noise reduction is not known yet.

In this study, the relationship between internal flow and fan noise reduction of cross flow fan was experimentally and numerically examined. The center of the eccentrically located vortex was inferred from flow measurements. A one-hole yaw meter [12, 13] was used to obtain the pressure distributions along the inner and outer peripheries of impeller. The behavior of the eccentrically located vortex was estimated from the results of pressure distributions.

A 3-D numerical analysis was carried out and was compared with the experimental result. The position of eccentrically located vortex was mainly influenced by air flow rate. The rotation noise level was inferred to result from the change in the position of the eccentrically located vortex and the accompanying flow around the vortex. The effect of the tongue geometry on rotation noise level was ascertained by flow measurements and numerical simulation of the internal flow.

2. Experimental Apparatus and Procedure
2.1 Measurements of fan performance and fan noise
Fig.1 shows the cross flow fan used in this study. The fan had 27 blades with an outside pitch of 10 mm and 0.8 mm blade thickness. The impeller had a 90 mm outside diameter and was 127 mm in axial length. A 66.5 mm diameter hole was bored in one of the end plates to permit the insertion of the one-hole yaw meter. The tongue was flat and the casing was composed of an arc joined to a flat duct. The experimental apparatus conforms to JIS B 8330 for the configuration of air blowers, see Refs.[12, 13]. The clearance between impeller and tongue was 3 mm in the radial direction, and the clearance between leading edge of casing and impeller was fixed at 11 mm. The flow rate was controlled by changing the aperture ratio of the duct outlet. The pressure transducers, fan tachometer and the microphone were used to determine the fan performance parameters and the noise level. The rotational speed of fan was maintained at 1400 min⁻¹. A digital manometer was used to record the static and total pressures in the duct. The noise measuring method was based on the noise level measurement standard JIS B 8346 for air blowers. The A-weighting fan noise was measured by a sound level meter, the data from which were input into a PC and processed via the FFT contained in the LabVIEW8.0 software.

Fig.1 Experimental apparatus
Fan performance was characterized in terms of the following three parameters [12, 13]:

total pressure coefficient $\psi_t$,

$$\psi_t = \frac{2 p_t}{\rho u_2^2}$$

flow coefficient $\phi$,

$$\phi = \frac{Q}{D L u_2}$$

specific noise level $K$,

$$K = SPL - 10 \log_{10} Q p_t^2$$

where $p_t$ is the total pressure, $\rho$ is the density of air, $u_2$ is the rotational speed of impeller based on the outer diameter, $Q$ is the flow rate, $D$ is the diameter of impeller, $L$ is the width of impeller, SPL is the sound pressure level.

2.2 Measurement of internal flow

To measure the internal flow, the one-hole yaw meter with an 1.4 mm outside diameter, 1.0 mm inside diameter and 0.5 mm pressure tap diameter was used. Peripheral distributions of the total pressure $p_t$, dynamic pressure $p_v$, and static pressure $p_s$ along the inner and outer peripheries of the impeller were obtained to determine the internal flow pattern. The inner and outer periphery radii were $R_{in}=30.0 \text{mm}$ and $R_{out}=51.0 \text{mm}$, respectively. The yaw probe was positioned at five axial locations and then rotated in 10° increments around the axis of the impeller as shown Fig.2. The axial measurement locations were $z_1=6.4$, $z_2=31.8$, $z_3=63.5$, $z_4=95.3$, $z_5=120.7 \text{mm}$ from the end of the impeller (opposite from the motor).

To investigate the effects of inflow and outflow velocities, $w_1$ and $w_2$, shown in Fig.3 on the fan noise, these velocities were determined from the following equations [13].

$$w_1 = \frac{v_{r1}}{\sin \beta_1}$$

$$w_2 = \frac{v_{r2}}{\sin \beta_2}$$

$$\tan \beta_1 = \frac{v_{r1}}{v_{\theta 1} - u_1}$$

$$\tan \beta_2 = \frac{v_{r2}}{v_{\theta 2} - u_2}$$

where, $v_1$ and $v_2$ are the radial velocity component on the inner and outer peripheries, respectively, $v_{\theta 1}$ and $v_{\theta 2}$ are the circumferential velocities on the inner and outer peripheries, respectively. In addition, $u_1$ and $u_2$ are the rotational speeds of impeller based on the inner and outer diameters, respectively. The above-mentioned one-hole yaw meter was traversed to measure $v_{r1}$, $v_{r2}$, $v_01$ and $v_02$.

3. 3-D Numerical Simulation Method

The flow analysis software SCRYU/Tetra V7 (CRADLE Co., Ltd.) was used to analyze the internal impeller flow. Figure 4 shows the 3-D mesh used for the numerical simulation of the internal flow of the cross-flow fan. The number of mesh points in the whole flow field was 5,650,000.

In the 3-D numerical simulation, three types of boundary layer elements were used to precisely estimate the flows in the close vicinity of the blades, the end plates, the casing and the tongue. The working fluid was air at a temperature of 20°C. The atmospheric pressure was used as the inlet boundary condition of the system, and mass flow rate was used as the outlet boundary condition. The rotational speed of fan was kept at a constant value of 1400 min⁻¹. The standard $k$-$\varepsilon$ model was used as the turbulence model.

![Fig.2 Measurement position by a one-hole yaw meter](image1)

![Fig.3 Relative velocities on the inner and outer peripheries](image2)

![Fig.4 The meshes set in calculation](image3)
4. Results and Discussion of Experiments and 3-D Numerical Simulation

4.1 Fan performance and fan noise

Fig. 5 shows the total pressure coefficient $\psi_t$ and total pressure efficiency $\eta_t$ versus the flow coefficient $\phi$. The chain line drawn between $\phi = 0.3$ and $0.7$ is the results of 3-D numerical simulation. The values of total pressure coefficient, $\psi_t$, of experiment and 3-D numerical simulation agree with each other for $\phi = 0.3$ to $0.5$, while the two results become to depart with a further increase in the flow coefficient, $\phi$. This difference is associated with the fact that the static pressure obtained from the 3-D numerical simulation becomes greater than the measured value for $\phi \geq 0.7$. The reason for the increase in the numerical value of the static pressure is not clear at present and must be left for a future study. The maximum value of the total pressure efficiency is $39\%$ and it appears for $\phi = 0.5$. The noise level, SPL, and the specific noise level, $K$, shown in Fig. 6 gradually decrease as the flow rate increases. The rotation noise becomes significantly low for $\phi \geq 0.7$. Fig. 7 shows the frequency analysis data for $\phi = \phi_{\text{max}}$ and $\phi = 0.3$, where $\phi_{\text{max}}$ denotes the maximum value of $\phi$.

As the rotation noise level is extremely lower than overall noise level, as can be seen in Fig. 7 (a), the turbulence noise is dominant at this flow rate. On the other hand, Fig. 7 (b) shows that the rotation noise level and the overall noise level are comparable. Accordingly, the rotation noise plays an essential role for $\phi = 0.3$. These results collectively indicate that the rotation noise is significant in the low flow rate regime, while the turbulence noise is dominant in the high flow rate regime.

4.2 Position of eccentrically located vortex center

The vortex center is located on the angular location of $\theta = \theta_{\text{VC}} \ [10, 13]$ where the total pressure, $p_t$, exhibits its minimum value. Fig. 8 shows the distribution of vortex center along the impeller axis obtained from 3-D numerical simulation in the high flow rate regime. A good agreement can be seen between the numerical and measured values of angular displacement of vortex center. The vortex center distribution for $\phi = 0.7$ is almost flat as shown in Fig. 8, whereas that for $\phi = 0.3$ is not flat as shown in Fig. 9. In the case that the turbulence noise is the main noise source, the vortex center is located near the edge of the tongue. Meanwhile, when the rotation noise is the main noise source, the vortex center departed from the edge of the tongue.
4.3 Relative velocity vector along the impeller

Figs. 10 and 11 show the relative velocity vectors, \( w_1 \) and \( w_2 \), obtained from experiments and 3-D numerical simulation for \( \phi = 0.7 \) and 0.3, respectively. The simulated results based on the 3-D numerical simulation are in good agreement with the experimental ones in the two cases. Definite difference cannot be seen between the results shown in Figs. 10 and 11. Consequently, it is difficult at present to discuss the effects of the relative velocity vectors on noise generation.

5. Development of Noise Reduction Method

5.1 Nine configurations of tongues

In our previous study [11], the fan noise could be reduced by alteration of the tongue shape. Unfortunately, a detailed mechanism of the fan noise reduction is not known yet. In this section, we mention about noise reduction caused by the alteration of tongue shapes. The nine different dividing tongue geometries investigated in this study are shown in Fig. 14. These geometries can be summarized as follows: Tongue A affords the reference condition (default tongue position, taken as the reference datum). The shaded regions in Fig. 14 show those portions of the dividing tongue geometry that are at the default reference position. Tongues B and C are both still flat tongues, but positioned one-half
blade pitch above or below the datum. Tongues D, E, F and G have a half pitch offset over a half of the tongue width. The tongue D is a mirror image of the tongue E. The tongues F and G are rotation of the tongues E and D about the tongue centerline, respectively.

5.2 Experimental results and discussion

5.2.1 Fan performance

The fan performance is essentially unaffected, or very slightly improved, by modifications to the dividing tongue by comparison of the performance of the modified tongues (step tongue D, skew tongue I) with that of flat tongue A in Fig.15. Meanwhile, Fig.16 shows a relatively significant decrease in the SPL and $K$, especially for tongues D and I. The SPL are reduced except for high flow rate ($\phi \geq 0.7$).

5.2.2 Fan noise performance

The spectral analysis of the acoustic signals from dividing tongues D and I for $\phi = 0.3$ are shown in Fig.17 (a) and Fig.17 (b), respectively. Tongues D and I result in almost 17 dB noise reduction at the blade passing frequency relative to that of tongue A. The overall noise level for tongue D was 59.7 dB and that for tongue I was 63.1 dB. There was some reduction in SPL for the other tongues as well. The skew tongue I reduced not only the rotation noise level but also second rotation noise level as shown Fig.17 (b).

The relationship between the overall noise level and the rotation noise level for the nine different tongues are shown in Fig.18. Fig.18 (a) shows the overall noise level for $\phi = \phi_{\text{max}}$ differs more than 17 dB from rotation noise level. At this flow rate, turbulence noise (sometimes called the broadband noise [1, 8]) is the main noise source. Meanwhile, Fig.18 (b) shows the noise data for a low flow rate of $\phi = 0.3$. The rotation noise levels of flat tongues A, B and C are nearly equal to the overall noise level. However, the rotation noise levels of the tongues E to I are vastly reduced. Accordingly, in this case the rotation noise is the main noise source. The alteration of tongue shapes therefore is very effective.
6. Conclusions

Experimental and numerical investigations were carried out to reveal the relationship between the flow pattern and the fan noise of cross-flow fans. The main findings obtained in this study can be summarized as follows:

(1) The characteristics of internal flow inside impeller obtained from 3-D numerical analysis are in good agreement with those experimentally measured under every experimental condition.

(2) The rotation noise is the main noise source in the low flow rate regime, while the turbulence noise is the main noise source in the high flow rate regime. Change in the main noise source is closely associated with the shift of the center location of the eccentric vortex.

(3) In the case that the rotation noise is the main noise source, the alteration of tongue shapes is very effective for reducing the noise level.

Nomenclature

- $D$: diameter of impeller, m
- $K$: specific noise level, dB
- $L$: impeller width, m
- $p_t$: total pressure, Pa
- $p_s$: static pressure, Pa
- $Q$: flow rate, m$^3$/s
- $\text{SPL}$: sound pressure level, dB
- $u$: rotational velocity, m/s
- $x$: axial coordinate
- $y$: vertical coordinate
- $z$: horizontal coordinate
- $\rho$: density, kg/m$^3$
- $\phi$: flow coefficient
- $\phi_t$: total pressure coefficient

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