Influence of Karman Vortex Street on Broadband Frequency Noise Generated from a Multiblade Fan

Souichi SASAKI**, Yoshio KODAMA** and Makoto HATAKEYAMA***

In the prediction theory for a broadband frequency noise generated from a multiblade fan, the vortices in Karman vortex street were divided into \( n \) pieces. The frequency distribution of the noise was estimated so that the Strouhal number could become constant even if the wake is spread by the diffusion. From the results of the measurement of the internal flow of the fan, it was found that the noise was related to the wake characteristics of the specific location in the scroll casing where the relative flow velocity was high. The noise operating in the vicinity of the maximum efficiency point of the fan was distributed over the domain from 500 Hz to 1 250 Hz. It was experimentally proved that when the distribution of the estimated sound pressure level corresponded to the measured broadband frequency noise, no influence of the vortices on the noise in the domains of high and low frequencies existed.

Key Words: Aerodynamic Acoustics, Blade, Internal Flow, Noise, Turbomachinery, Vortex, Wake

1. Introduction

Multiblade fans are widely used in domestic air conditioners, commercial sanitary apparatuses, etc. In the case of these fans, decreasing the fan noise is one of the important technical issues for its comfortable operation. The spectra of the fan noise showed that the noise is composed of a few types of aerodynamic noise. This noise comprises not only a discrete frequency noise due to the interference with the wake and the cutoff of the scroll casing but also a broadband frequency noise distributed over the wide frequency domain(1). Depending on the design of the impeller, the sound pressure level of the total broadband frequency noise can exceed the discrete frequency noise. In such case, the broadband frequency noise becomes the major area of concern of the fan noise.

According to Sharland(2), the broadband frequency noise of an axial flow fan can be described by at least three mechanisms—the surface pressure field arising from the turbulent boundary layer, the influence of the vorticity shedding from the surface of a body and the incoming flow with the initial turbulence. On the other hand, Mugridge(3) had derived a theory of the broadband frequency noise radiation due to the turbulent boundary layer on the surface of the blades of an axial flow fan. In an experiment with the actual fan, the comparison of the predicted noise spectra with the measured radiation from the fan had suggested a reasonable validity. However, the surrounding flow of the blade of an axial flow fan differs from that of a compact centrifugal fan because the basic design of the fan is completely different. There have been few practical examples on the prediction theory of the broadband frequency noise based on the typical operation of the multiblade fan.

In the present study, a prediction theory for the broadband frequency noise generated from a multiblade fan has been derived based on the characteristics of Karman vortex street. The feature of the broadband frequency noise under operation in the vicinity of the maximum efficiency point has been estimated based on this theory.

Nomenclature

- \( B \) : number of blades
- \( b \) : blade width [mm]
- \( C \) : chord length [mm]
- \( D_1 \) : inner diameter [mm]
- \( D_2 \) : outer diameter [mm]
- \( D_w \) : width of shear layer between pressure surface side and suction surface side [mm]
2. Experimental Apparatus

The test impeller and shape of a blade are given in Fig. 1. The main dimensions are summarized in Table 1. In the subsequent section, the fan consisting of this impeller is referred to as SC99.

Figure 2 is the experimental apparatus. The dimensions of the scroll casing are listed in Table 2. The fan noise was measured in an anechoic room; the background noise in the room was below 25 dB in the A-weight measurement. The observation point of the noise is 1.0 m upstream from the bellmouth entrance along the axis of the
motor. A sound terminator is attached to the exhaust port. The terminator has a static pressure tap and a damper for controlling the flow rate. In order to intercept the electric noise from the motor, the motor is stored in an aluminum box; the interior of the box is covered with a rubber soundproof material. In the experimental results of Figs. 11 and 12, the tendency analyzed by the least squares method was represented. The internal flow was measured from MP1 to MP4 around the scroll casing in Fig. 2. The distance between the measurement point of the internal flow and center of the motor axis is 72.5 mm. The five-hole Pitot tube was used to measure the absolute flow velocity and static pressure at the impeller outlet. The experiment was performed under the condition of 1400 rpm. The operation point of the flow coefficient was determined as 0.23.

### 3. Prediction Theory of Broadband Frequency Noise

#### 3.1 Aerodynamic noise of a multiblade fan

Figure 3 is a spectra of the fan noise represented by each 1/3 octave band frequency in the A-weight measurement. A discrete frequency noise (DFN) was generated in the vicinity of the blade path frequency (BPF = 2333 Hz). On the other hand, the broadband frequency noise (BFN) was distributed in the vicinity of the frequency band at 1000 Hz. In this study, the fan noise spectra are defined as broadband frequency noise. The sound pressure level of the broadband frequency noise is almost identical to the level of the discrete frequency noise. Depending on the driving condition, the sound pressure level of the total broadband frequency noise exceeds the discrete frequency noise.

#### 3.2 Flow model

A flow model of the meridional section (r–z section) in the scroll casing is given in Fig. 4. Kawaguchi et al. proved that the flow passing through the impeller of a multiblade fan forms a vortex flow to the bellmouth side under the condition of a high flow rate. In this study on the flow model, the flow has been divided into two domains. One is a domain with the vortex flow at the bellmouth side (vortex flow domain), and the other is the mainstream domain where the outflow is biased to the hub side (mainstream domain). The ratio of the vortex flow domain to the span length, i.e., the blockage factor was estimated by Eq. (1), which is based on the idea of the displacement thickness of the radial flow velocity.

$$K_b = \frac{1}{b_2} \int_0^{b_2} \left(1 - \frac{v(z)}{v_{\text{max}}}ight) dz$$

where $v_2(z)$ is the radial flow velocity at the impeller outlet. In this chapter, the relation between the wake characteristics in mainstream domain and the aerodynamic noise will be discussed. The wake characteristics mean the width of wake formed by Karman vortex street and the diffusion of the wake.

The velocity triangles at the inlet and outlet of the impeller are given in Fig. 5. The flow at the inlet side has no pre-rotation. The flow between blades forms a laminar boundary layer on the pressure surface side (PS side). On the suction surface side (SS side), the flow reattached at the leading edge also forms a laminar boundary layer and is separated again at a separation point.

The design shape of the arc blade is shown in Fig. 6. The separation point on the suction surface side has been determined to be the point where the stream line B with a deviation angle and the arc A come in contact with each other. The length of the separation domain $L$ is determined by Eq. (2).

$$L = |l_2 - l_3|$$

where $l_2$ is the vector defined from the center of the arc to the trailing edge and $l_3$ is the vector defined from the center to the separation point.

### 3.3 Width of wake

The vector of the relative flow velocity at the impeller...
boundary layer on the suction surface. The width of the shear layer on the suction surface is assumed as Eq. (4).

$$D_{SS} = 0.6L\sin\gamma_2$$  \hspace{1cm} (4)

The experimental constant in Eq. (4) has been determined by the velocity profile of Pohlhausen\(^5\). From the aforementioned estimation, the width of shear layer between the pressure surface side and the suction surface side at the trailing edge is given by Eq. (5).

$$D_w = \delta_{PS} + t + D_{SS} + \delta_{SS}$$  \hspace{1cm} (5)

where \(t\) is the thickness of blade. Karman vortex street forms the wake within a domain of the shear layer. On the other hand, the irrotational flow in the different domain becomes a jet.

Schlichting has theoretically presented a velocity defect of the wake as a Gaussian exponential function\(^6\). Then the half width of the mathematical distribution means the half width of shear layer in Eq. (5). Authors have experimentally proved that the Strouhal number satisfied the relation of 0.2 when the width of wake is estimated as the half width\(^7\). Thus, one half of Eq. (5) was given as the width of wake: \((d_2 = D_w/2)\).

### 3.4 Division of Karman vortex street

In Fig. 8, Karman vortex street in the wake is given. When the wake diffuses with the condition of maintaining a constant Strouhal number in the near field, the width of wake is divided into \(n\) pieces in the \(x\)-direction.

$$d_j = \frac{S_i w_2}{f_j}, \quad j = 2 \sim n$$  \hspace{1cm} (6)

where \(f\) is the frequency of the vortex shedding, \(S_i\) is Strouhal number \((S_i = 0.2)\) and \(j = 2\) means the outlet of the impeller. When the center frequency of \(1/3\) octave band is given to Eq. (6), the number of discrete vortices becomes \(n = 21\) within from 100 Hz to 10 kHz. The width of wake in \(j = 2\) estimated by Eq. (6) is same as the width in the aforesaid section.

### 3.5 Sound pressure produced by Karman vortex street

Curle\(^8\) had given the solution of the acoustic wave equation\(^9\) on the basis of the influence on the solid surface as follows.

$$p(t) = \frac{1}{4\pi a_0} \frac{r_j}{r^2} \frac{\partial}{\partial t} \int_{S} P_i(t) dS$$  \hspace{1cm} (7)
where \( p(t) \) is the sound pressure, \( a_0 \) is speed of sound (340 m/s), \( r \) is the distance from the sound source to the observation point, and \( P_1(t) \) is the pressure acting on the area \( dS \) on the surface. If the right side of Eq. (7) is replaced with a total value such as a typical lift coefficient, the feature of the sound pressure due to individual vortices in the wake cannot be expressed. Therefore, the vortices in the wake are divided into \( n \) pieces, and then the sound pressure caused by an individual vortex has been given as Eq. (8).

\[
p_j = \frac{1}{4\pi a_0} \frac{\cos \theta}{r} \frac{\partial F_j}{\partial t}, \quad \therefore j = 2 \sim n
\]

(8)

where \( \theta \) is the angle between the sound source and observation point and \( F_j \) is the local lift caused by an individual vortex in the wake. The circulation of a vortex in the wake is estimated as liner fluctuation in Eq. (9).

\[
\Gamma_j(t) = \pi d_j w_j \sin(\omega t + \epsilon)
\]

(9)

where, \( d_j \) is width of wake in Eq. (6), \( \omega_j \) is angular frequency of vortex shedding and \( \epsilon \) is the retarded phase. The differential function of the local lift produced by the circulation is Eq. (10).

\[
\frac{\partial F_j}{\partial t} = \frac{\rho \pi \omega_j^2 d_j L_{Sj} \omega_j}{\sqrt{2}}, \quad F_j = \rho w_j \Gamma_j(t) L_{Sj},
\]

(10)

where, \( \sum \) denotes the symbol for root mean square, \( L_{Sj} \) is the spanwise correlation length of each local lift. The spanwise correlation length is almost identical to the width of wake when the feature of the individual vortex is isotropic scale(10).

The pressures of the center of the vortices maintain a state of equilibrium with the pressure on the solid surface of the adjoined blades when the vortices exist in the near field (see Fig. 8). Therefore, the distribution of sound pressure generated from the vortex is expressed as Eq. (11) by the overlap of these vortices.

\[
\bar{p} = \frac{\cos \phi \pi \omega_j^2 d_j^2}{2 \sqrt{2} \omega_t r}, \quad \therefore d = \sum_{j=2}^{n} d_j
\]

(11)

When the sound pressure produced by each vortex is overlapped, the spectra density of the sound pressure is obtained.

3.6 Broadband frequency noise of a multiblade fan

Equation (11) is the sound pressure per unit span length radiated from one blade. Thus, the sound pressure level generated from a multiblade fan becomes Eq. (12) when the previous equation is integrated with the span and circumferential direction of the impeller.

\[
\bar{p}^2 = \left\{ \frac{\cos \phi \pi \omega_j^2 d_j^2 K_0 B (1 - K_b) b_2}{2 \sqrt{2} \omega_t r} \right\}^2, \quad d = \sum_{j=2}^{n} d_j
\]

(12)

Therefore, the sound pressure level of the broadband frequency noise becomes Eq. (13).

\[
L_p = 10 \log \left( \frac{\bar{p}^2}{P_0} \right)
\]

(13)

where \( L_p \) is the sound pressure level, \( P_0 \) is the minimum sound pressure (20 \( \mu \)Pa). It is possible to predict the spectra density of the sound pressure caused by Karman vortex street.

4. Results and Discussion

4.1 Aerodynamic characteristics

The aerodynamic characteristics of fan are presented in Fig. 9. These characteristics were defined by Eq. (14).

\[
\phi = \frac{Q}{60 \pi D_2 b_2 u_2}, \quad \psi_i = \frac{2 P_i}{\mu u_2}, \quad \lambda = \frac{2 W}{\pi \rho D_2 b_2 u_2^3}, \quad \eta = \frac{\phi \psi_i}{\lambda}
\]

(14)

where \( \phi \) is flow coefficient, \( \psi_i \) is total pressure coefficient, \( P_i \) is total pressure (Pa), \( \lambda \) is power coefficient, \( W \) is input power from a motor (kW) and \( \eta \) is efficiency. The maximum efficiency was approximately 60% when the flow coefficient was 0.20. The pressure coefficient and efficiency have not still caused a deterioration of the performance at the operation point in this experiment (\( \phi = 0.23 \)).

4.2 Analysis of internal flow

In Fig. 10, the relative flow velocity and the width of wake are shown. The data at MP4 has been employed as
Table 3 Summary of the wake characteristics of the mainstream domain and vortex flow domain

<table>
<thead>
<tr>
<th>Domain Description</th>
<th>( w_2 \text{ m/s} )</th>
<th>( d_2 \text{ mm} )</th>
<th>( f_2 \text{ Hz} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mainstream Domain (( z / b_2 &lt; 0.6 ))</td>
<td>6.821</td>
<td>1.346</td>
<td>1014</td>
</tr>
<tr>
<td>Vortex Flow Domain (( z / b_2 &gt; 0.6))</td>
<td>2.693</td>
<td>0.921</td>
<td>582</td>
</tr>
</tbody>
</table>

Fig. 11 Distribution of the wake characteristics around the scroll casing

Table 4 Summary of the wake characteristics around the scroll casing

<table>
<thead>
<tr>
<th>( \theta ) deg.</th>
<th>( w_2 \text{ m/s} )</th>
<th>( d_2 \text{ mm} )</th>
<th>( f_2 \text{ Hz} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>MP4 ( \phi = 0.23 )</td>
<td>6.373</td>
<td>1.404</td>
<td>908</td>
</tr>
<tr>
<td>MP3 ( \phi = 0.23 )</td>
<td>4.880</td>
<td>1.004</td>
<td>972</td>
</tr>
<tr>
<td>MP2 ( \phi = 0.23 )</td>
<td>4.271</td>
<td>0.971</td>
<td>880</td>
</tr>
<tr>
<td>MP1 ( \phi = 0.23 )</td>
<td>3.877</td>
<td>0.885</td>
<td>876</td>
</tr>
</tbody>
</table>

Fig. 12 Relationship between the flow coefficient and sound pressure level

Fig. 13 Spectra distribution of the fan noise with different flow coefficients

A measurement point for the analysis of the wake characteristics. The mainstream domain implied a biased flow to the hub side, and the relative flow velocity was high within the domain. These wake characteristics in each domain are summarized in Table 3. The frequency of vortex shedding was estimated by the Strouhal number in Eq. (6). The sound pressure level in the mainstream domain was approximately 27 dB higher than that in the vortex flow domain when it was estimated by Eq. (11). This indicates that the influence of the vortex flow domain on the fan noise is minimal.

The relative flow velocity and the width of the wake distributed around the scroll casing are shown in Fig. 11. In the case of the scroll casing in Fig. 2, \( \theta \) on the horizontal axis is the angle in the counter-clockwise rotation from the base criterion of the angle made at 12 o’clock. These wake characteristics are averaged in the mainstream domain. The relative flow velocity and the width of the wake attain a maximum value at MP4. The wake characteristics are summarized for each measurement point in Table 4. The sound pressure level at MP4 estimated by Eq. (11) with the measured wake characteristics was 9.9 dB higher than the level of MP3. The results of this experiment show that the flow in the scroll casing was biased to the vicinity of MP4, and the broadband frequency noise is related to the wake characteristics in the mainstream domain where the relative flow velocity was high.

4.3 Broadband frequency noise

The relation between the fan noise and flow coefficient is shown in Fig. 12. When the flow coefficient was lower than 0.20, the fan noise was almost at a constant level. The relative flow velocity is low in the domain of the low flow coefficient. Therefore, it is considered that the fan noise in the domain is different from the noise generated by the flow model in the previous chapter. The fan noise tended to increase in proportion to the sixth power of the flow coefficient when the flow coefficient was greater than 0.20.

In Fig. 13, several spectra of the sound pressure level with different flow coefficients are shown. In the case of \( \phi = 0.20 \), the broadband frequency noise was distributed from 500 Hz to 1250 Hz. However, the discrete frequency noise did not exist at the blade pass frequency \((\text{BPF} = 2333 \text{ Hz})\). The noise at 1700 Hz is a mechanical vibra-
tion noise. In the case of $\phi = 0.23$ and $\phi = 0.25$, the discrete frequency noise and the broadband frequency noise were generated. The sound pressure of the broadband frequency noise was proportional to the minus second power of the frequency even when the flow coefficient was different. Therefore, the feature of the frequency is regarded as a wake characteristic that diffuse with the condition of maintaining a constant Strouhal number because the sound pressure is in inverse proportion to the frequency. Table 5 is a summary of each flow coefficient of the total broadband frequency noise distributed from 500 Hz to 1 250 Hz and the fan noise. The ratio of the broadband frequency noise to the fan noise was 53.7% when the fan was operated at $\phi = 0.23$, and the ratio was high in the low flow rate.

The contour line of the sound pressure level is shown in Fig. 14. In this figure, the horizontal axis is the frequency and the vertical axis is the flow coefficient. In the domain higher than the maximum efficiency point, the discrete frequency noise and the broadband frequency noise in the vicinity of 1 000 Hz increased. The exponential characteristics of the sound pressure corresponding to the flow coefficient in each 1/3 octave band frequency from 400 Hz to 1 250 Hz is summarized in Table 6. The data of flow coefficient that was greater than the maximum efficiency point ($\phi = 0.20$) shown in Fig. 14 were used for this analysis. The variable $S$ in the table is the exponential component of the flow coefficient in Fig. 12. The variable $S$ became approximately six within the domain from 400 Hz to 1 250 Hz. Since the fan noise increased in proportion to the sixth power of the flow coefficient, the sound source may be a dipoles type.

In the aforementioned analysis of the internal flow of the fan, it was experimentally proved that the fan noise was due to the influence of Karman vortex street on the mainstream domain in the vicinity of MP4. However, in this experiment, it was not sufficient to analyze the influence on the circumferential extent of the biased flow to the vicinity of MP4. In the following estimation, the biased flow of 24 degrees to the circumferential direction of the impeller was given to the measuring wake characteristics at MP4 ($K_{\theta} = 24/360$). It will become necessary to decide these reasonable parameters in the prediction of the actual fan noise as a future work.

In Fig. 15, the measured spectra of the broadband frequency noise are compared with the spectra of the estimation. The sound pressure level of the fan noise was measured by the A-weight measurement, and the spectra was analyzed for the each 1/3 octave band frequency. The estimated sound pressure level is recognized as having a reasonable validity from 500 Hz to 1 250 Hz. The measured sound pressure level in the low frequency domain is smaller than the estimated level. It is thought that the pressures of the vortices no longer influence the sound pressure on the observation point because the enlarged vortex in the low frequency domain is distant from the blade. The measured level in the high frequency domain was also smaller than the estimation. It has been considered that this is due to the fluctuation of the small scale vortices in the boundary layer. On the other hand, the level at the blade pass frequency ($\text{BPF} = 2 333 \ Hz$) exceeded the estimation. The fan noise has to be estimated by a flow model different from that used in this study because this is an in-

Table 5
Summary of the composite sound pressure level of the broadband frequency noise and fan noise

<table>
<thead>
<tr>
<th>$\phi$</th>
<th>BFN $L_4$ dB</th>
<th>Fan Noise $L_4$ dB</th>
<th>$p^2$(BFN) / $p^2$(Fan Noise)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.20</td>
<td>40.7</td>
<td>42.3</td>
<td>0.692</td>
</tr>
<tr>
<td>0.23</td>
<td>42.5</td>
<td>45.2</td>
<td>0.537</td>
</tr>
<tr>
<td>0.25</td>
<td>43.6</td>
<td>47.6</td>
<td>0.398</td>
</tr>
</tbody>
</table>

Table 6
Relationship between the frequency in each 1/3 octave band and the exponential characteristic of the sound pressure level on the flow coefficient

<table>
<thead>
<tr>
<th>Frequency $f$ Hz</th>
<th>Exponent $S$</th>
<th>Frequency $f$ Hz</th>
<th>Exponent $S$</th>
</tr>
</thead>
<tbody>
<tr>
<td>400</td>
<td>6.2</td>
<td>800</td>
<td>6.2</td>
</tr>
<tr>
<td>500</td>
<td>7.1</td>
<td>1000</td>
<td>6.3</td>
</tr>
<tr>
<td>630</td>
<td>6.6</td>
<td>1250</td>
<td>5.5</td>
</tr>
</tbody>
</table>

Fig. 14 Contour line of the sound pressure level of SC99

Fig. 15 Comparison of the measured broadband frequency noise of SC99 with the estimated spectra
terference noise between the cutoff of the scroll casing and the wake.

5. Conclusions

We have proposed a prediction theory of the broadband frequency noise generated by a multiblade fan. The noise and internal flow of the actual fan under operation in the vicinity of the maximum efficiency point have been measured. The following results were obtained based on this theory.

(1) The influence of the vortex flow at the bellmouth side on the broadband frequency noise of the fan was minimal.

(2) The broadband frequency noise was related to the wake vortices of the mainstream domain in the scroll casing where the relative flow velocity was high.

(3) The sound pressure of the fan noise was proportional to the sixth power of the flow coefficient from 500 Hz to 1250 Hz, and the spectra of the sound pressure became a distribution that was inversely proportional to the second power of the frequency even when the operation point was different.

(4) The influence of Karman vortex street on the broadband frequency noise in the high and the low frequency domains ceased to exist when the estimated noise corresponded with the measured fan noise.

References