ABSTRACT

The rotational noise is one of the main sources of the noise emitted from a fan. This type of noise has discrete frequency components, which is so harsh that many methods have been developed to reduce the noise. Studies on the rotational noise, however, have hardly been made to clarify the generation mechanism and find concrete methods to reduce the noise.

In order to suppress the discrete frequency components of the rotational noise, the spacing of the blades were changed and the comparison was made with a fan having equally spaced blades. The discrete frequency components were found to be suppressed in the unequally spaced fan and the noise tended to have a nature of white noise if the blade spacing and blade setting angles were properly chosen.

The instantaneous pressure change on the shroud was measured in order that the pressure change was related to the discrete frequency noise was closely related to the pressure changes on the blade surfaces. By expressing the pressure change due to the blade passing in terms of the blade spacing and decomposing it into Fourier series, the noise characteristics was analyzed. The prediction result for the rotational noise emitted by an equally spaced fan was presented and found to agree well with the experiment.

1. INTRODUCTION

The noise generated by an axial fan is mainly composed of two types of noise. One is rotational noise with discrete frequency components and the other is turbulent noise with broad-band frequency components. As the former can be severely annoying, prediction and reduction of the noise is strongly needed in the engineering application such as an automotive radiator cooling fan.

Based on the sources of the noise, the rotational noise of a fan can be classified into interaction and blade passing frequency noise (BPF noise (Gutin, 1948)). While the former can be reduced by removing the asymmetric components in the fan system, BPF noise is very difficult to be reduced because it is generated by the inevitable pressure difference between the upper and lower surfaces of the blades. One of the conventional methods for reduction of BPF noise is to disperse the discrete components into a band frequency components by employing unequally spaced blades. There are few studies on the performance and noise characteristics of a fan having unequally spaced blades. Mellin (1970) obtained experimentally relationship between unequally spaced blades and the first order BPF noise component. Segawa (1982) modeled the discrete pressure changes by using delta functions and obtained the noise characteristics, which is not accurate enough to predict the rotational noise characteristics at higher order components. Dobrzynsky (1989, 1990) indicated the potential for a noise reduction of several decibels by unequally spaced propeller blades.

In order to disperse the discrete components of rotational noise and to design fans with favorable tonal characteristics, an estimation method for the rotational noise, especially the BPF noise components, is developed in this study and the calculation results is compared with the experiment.

2. EXPERIMENTAL APPARATUS AND METHOD

Figure 1 shows a schematic view of the experimental apparatus. The axial fan was located just downstream of an automotive radiator and has the inner and outer diameters of 300mm and 120mm, respectively. Experiments were made at the dimensionless flow rate and pressure head of \( \phi = 0.28 \) and \( \zeta = 0.04 \), respectively, when the rotational speed was kept constant at 2000rpm. Two kinds of five-blade axial fan were used in this study. Figure 2 shows an unequally spaced fan, it has the blade spacing which is designed so as to reduce the
first order BPF noise, exhibiting the greatest noise energy, by the method of Mellin (1970). The noise characteristics was compared between an equally and unequally spaced fan. To suppress the generation of interaction noise as much as possible, a ring-type shroud with 3mm tip clearance was employed both for the fans. The pressure distribution near the tip was obtained, five pressure transducers of 1.6mm outer diameter were installed in the shroud at the section of the fan as shown in Fig. 3. The natural frequency of the transducers was 150KHz, which was large enough for the measurement of the pressure change due to the blade passing in the flow field. The analog outputs from the pressure transducers were transmitted to a personal computer together with the data from the optical tachometer and rotary encoder as shown in Fig. 1.

By plotting the pressure change due to the blade passing, the instantaneous distribution of the pressure on the shroud was obtained within the error of blade angular position of 1 degree. Noise was measured in an anechoic chamber in which the background noise was less than 20dB (A-weighted). The microphone was installed 1m upstream of the fan on the rotational axis. The measurement conformed to JIS B 8346 (Japanese Industrial Standard), because the object of this study was to clear the mechanism of noise generation and to reduce it. The quantitative tendency of noise characteristic was ascertained not to be changed by the location of a microphone. Noise characteristics was expressed without A-weighted sound pressure level and was obtained by narrow-band analysis up to 800Hz.

The measurement uncertainties in the variables are given below:
\[ \delta = \pm 1.16 \text{ percent}, \ \phi = \pm 1.23 \text{ percent}, \ \text{SPL} = \pm 3 \text{ dB} \]
\[ P_w = \pm 5 \text{ percent} \] (pressure on the shroud wall)

### 3. RESULTS AND DISCUSSIONS

#### 3.1 Noise characteristics of the prototype fans

Figures 4(a) and (b) show the measured noise characteristics of the equally and unequally spaced fans. The overall level of the equally spaced fan is 63.3dB (51.9dB(A) and 57.4PN(dB)) and that of the unequally spaced fan is 62.9dB (50.1dB(A) and 55.2PN(dB)), respectively. Rotational noise having discrete frequencies are dominant in the equally bladed fan but these discrete frequency components are seen to be dispersed into many frequency components in the unequally spaced fan, which generates noise reduction of more than 10dB not only in the first order rotational tone but in the higher ones except for the 3.4th rotational tone, which is equivalent to the 17th shaft order frequency noise component.

#### 3.2 Pressure measurement at blade tip

In the authors' previous study (Akaike, 1992), it has been found that significant noise is generated near the blade tip where the rotational speed is large and the pressure difference between the upper and lower surfaces is increased. Figures 5(a) and (b) show the pressure change due to the blade passing measured on the shroud by the pressure transducer located in the mid-way of the clearance for the equally and unequally spaced fans. In the fan with equally spaced blades the minimum pressure equal to Pmin is generated negatively at a certain distance from the blade tip.
constant angular pitch when a blade passes the measuring section, while in the unequally spaced fan the different values of the lowest pressure appears according to the unequally bladed spacings and has different values as shown in Fig. 5(b).

Figures 6(a) and (b) show the blade to blade pressure distribution in the equally and unequally spaced fans, respectively. The dotted line and solid lines show positive and negative pressure respectively. A space between the lines indicates pressure difference of 25Pa. In the mid-way of the blade passages pressure increases gradually toward the pressure side but decreases rapidly in the suction side. The equi-pressure curves exhibits almost the same configuration in the fan with an equally spaced blade in Fig.6(a), but the pressure difference across the blades differs with the blade pitches in the unequally spaced fan as shown in Fig. 6(b). In the suction side of the blade passage the pressure increases along the trailing surface of the blade but continues to be negative up to the exit both in the wider and narrower passages. In the pressure side, however pressure exhibits positive value only in the wide passages, causing its larger change across the blade and the BPF noise as shown below.

4. PREDICTION OF ROTATIONAL NOISE

4.1 Prediction by periodicity of pressure change

The magnitude of rotational noise components depend largely on the pressure change at the blade tip, as analyzed by Segawa (1982) using a centrifugal fan.

Let the pressure change \( P(\theta) \) due to the blade passing be simulated by a pulse wave of the equal amplitude and given dimensionlessly as a summation of delta functions as the following equation.

\[
p(l) = 0.5 \delta(\theta - \alpha(i)) + 0.5 \delta(\theta - \alpha(i+1)) + 0.5 \delta(\theta - \alpha(i+2))
\]

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Experiment

Prediction

\[ P(\theta) = \sum_{i=1}^{n} \delta(\theta - \theta_i) \]  \hspace{1cm} (1)

By decomposing the above equation into a Fourier series, the amplitude for each frequency is given by Eq. (2) and expressed in decibels by Eq. (3).

\[ a_k = \frac{1}{2} \sqrt{\left( \sum_{j=1}^{n} \cos \left( k \theta_j \right) \right)^2 + \left( \sum_{j=1}^{n} \sin \left( k \theta_j \right) \right)^2} \]  \hspace{1cm} (2)

\[ e_k = 20 \log_{10} d_k + C \]  \hspace{1cm} (3)

In Eq. (3), the last term of rhs, C is a constant to match the order level to the noise level by experiment and C = 40 was used in this stage.

Figure 7 shows the pressure level of each component of the rotational order for the present unequally spaced fan. Though lower levels are shown to be generated at the frequencies near 200Hz, its level increases between 300 and 400Hz, and decreases again (These quantitative tendency are shown with the wavy line in the Fig. 7). While a remarkably high level can be seen at the 17th component of the shaft order frequency tone (SOF). The changes of the noise levels almost agree with the measured discrete components as shown in Fig. 4(b), in which the BPF noise levels of frequency more than 600Hz are decreased unlike the predicted result. One of the reason for this disagreement may be the inadequate assumption of the pressure change in the above modeling.

4.2 Revised prediction method by pressure wave modeling

In the above prediction the magnitude of the pressure drop due to the blade passing was assumed unchanged. In the unequally spaced fan, however, the amplitude of the pressure wave varies largely according to the blade spacings as shown in Fig. 5(b) although the pressure waves exhibit almost similar configuration, expressed by a combination of two parabolic curves as shown in Fig. 8. Moreover, the value of negative pressure change \( P_{im(i+2)} \) is assumed to be given by a linear relation of the blade pitch \( \alpha (i+1) \) and the values of the leading and trailing low pressure changes, \( P_{im(i+1)} \) and \( P_{im(i+3)} \), respectively, as given by the following equation (see Fig. 5(b)).

\[ P_{im(i+2)} = C_1 P_{im(i+1)} + C_2 \frac{\alpha (i+1)}{\alpha_0} + C_3 P_{im(i+3)} \]  \hspace{1cm} (4)

As the contribution of the leading pitch angle \( \alpha (i) \) on \( P_{im(i+1)} \) is considered to be predominant, Equation (4) can be rewritten as

\[ P_{im(i+2)} = C_1 \alpha (i) + C_2 \frac{\alpha (i+1)}{\alpha_0} + C_3 \frac{\alpha (i+2)}{\alpha_0} \]  \hspace{1cm} (5)

Equation (5) should be valid for an equally spaced fan. Thus,

\[ C_1 + C_2 + C_3 = 1 \]  \hspace{1cm} (6)

Based on the experimental data of the pressure curves, the following combination of the coefficients are obtained by trial and error method as follows.

\[ C_1 = 0.8 \text{ to } 0.9 \]
\[ C_2 = 1 - C_1 \]
\[ C_3 = 0 \]  \hspace{1cm} (7)

![Fig. 8 Modeling of pressure change due to a solitary wave](image_url)

![Fig. 9 Comparison of pressure change between experiment and simulation](image_url)
In using the above equation, the angular pitch of $\alpha(i)$, $\alpha(i+1)$ and $\alpha(i+2)$ in Eq. (5) had better to be replaced by $\alpha_0$ when the blade space is larger than that of an equally blade fan, $\alpha_c$. In Eq. (7), the value of $C_3$ indicates that the trailing space has little effects on the pressure increase.

Using above model pressure change, the dimensionless pressure curve can be given by the following equation,

$$P_t(i+2) = f(\alpha(i), \alpha(i+1), \alpha_0)$$ (8)

where $P_t(i)$ denotes the discrete wave of the pressure drop modeled by two parabolic curves and has its amplitude of Eq. (5). The comparison of the pressure curves between the experiment and the above modeling is shown in Fig. 9, in which the pressure curve can be seen well simulated when the coefficient of $C_2$ is taken to be 0.85. Thus, the noise components can be calculated by decomposing the pressure curve of Eq. (8) into a Fourier series.

A comparison is made between the predictions by the pulse pressure model of Eq. (1) and by the present model in Eq. (8) as shown in Fig. 10, in which little differences can be seen in the low frequency zone. In the high frequency zone, however, the pressure levels, which are predicted higher in the pulse pressure model, are decreased to where they agree with the experimental results in Fig. 4.

4.3 Application of the present model

In order to verify the validity of the present model, it was applied to a four-blade axial fan with unequally spaced blades as shown in Fig. 11. The distribution of the blades is symmetrical with respect to the perpendicular bisectors plotted by broken lines, thus $\alpha(1)$ and $\alpha(2)$ are equal to $\alpha(3)$ and $\alpha(4)$, respectively. The prediction results of noise levels for different combinations of blade spacing angles ($\alpha(1)$, $\alpha(2)$), being (102°, 78°) and (120°, 60°) are shown in Fig. 12. In the figure fractional components such as 0.5 and 1.5th order BPF noise tones (equal to 2nd and 6th order component) are seen to be generated both for the two combinations of spacing. From the comparison between these results, it is found that the magnitude of each fractional component differs according to the distribution of the blade spacing, which suggests that the rotational noise can be controlled by changing the blade spacing.

In order to reduce the annoying noise components, discrete frequency components were dispersed into many frequency components by unequal blade spacing as shown in Fig. 4(b). Figure 13(a) and (b) show the blade space configuration and the measured noise characteristics of a prototype fan which is designed so as to have a wider band of rotational frequency components. Irrespective of the unchanged turbulent noise, the discrete frequency components are flattened, resulting in the reduction of total rotational noise (62.8 dB, 49.6 dB(A), 54.3 PN(dB), respectively). Little deterioration in the performance was caused in the unequally spaced fans compared with that for the equally spaced one.
5. GENERATION MECHANISM OF 17TH SHAFT ORDER FREQUENCY NOISE

Of the experimental and predicted frequency components for the five-blade fan shown in Figs. 4(b) and 10, the 17th shaft order frequency one is predominant. In order to discuss the mechanism, dotted line are plotted at every 21.7 degree, dividing the circumference into 17 equal sectors, in Fig. 14, and the configuration of the equally spaced five blade fan shown in Fig. 2 is pasted so that one of the blade centerlines plotted by bold lines coincides with a dotted line. Neglecting a small difference in the circumferential location, the centerlines of the blades are found to agree almost with the dotted lines, which is considered to generate a higher level of 17th order noise in the prototype fan.

In order to investigate the effects of the blade space angle on the 17th order noise component, the blade angle of $\alpha(2)$, which is taken to be equal to $\alpha(4)$ in Fig. 2, is changed from 36 to 56 degree. The measured noise pressure level of the 17th order component is seen to take its maximum for $\alpha(2)$ equal to 42 and 44 degree, which are nearly equal to the dividing angle.

Thus, the result shows that an increased level of n-th order component may be generated when the unequally spaced blades are so distributed circumferentially as to coincide with some of n-pieces of sectors.

6. CONCLUSIONS

To reduce the rotational noise generated by the rotating blade of an axial fan, the noise characteristics was analyzed both for the fans with equally and unequally spaced blades. The results are summarized as follows:

1) By using unequally space blade fan, the rotational noise components can be flattened and dispersed into more components, resulting in the decrease in the rotational noise of a fan.

2) The rotational noise generated by an unequally spaced fan was simulated by a pressure wave model and found to be well predicted over the wide range of the frequency.

3) By using this method an axial fan with desired noise characteristics will be obtained by choosing an appropriate combination of the blade spacings.
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NOMENCLATURE

B : Number of Blades on Rotor = 5
\(d_k\) : Rotational Order Amplitude for each Frequency
\(e_k\) : Decibel Value of Rotational Order Amplitude for each Frequency (dB)
BPF\(_n\) : Blade Passing Frequency (Hz) = \(nBN/60(n=1,2,\ldots)\)
SOF\(_n\) : Shaft Order Frequency (Hz) = \(nN/60(n=1,2,\ldots)\)
N : Rotor Speed = 2000 (rpm)
P\(_w\) : Mean Pressure on the Shroud Wall (Pa)
P\(_0\) : Mean Average Peak Value of Wall Pressure on Equally Blade Spacing of Rotor (minus value) (Pa)
P\(_1\) : Nondimensional Mean Wall Pressure = \(P_w/P_0\)
P\(_{1m}\) : Peak Value of Nondimensional Mean Wall Pressure
SPL : Sound Pressure Level (dB)
\(\alpha(\circ)\) : Angle between Adjacent Blades (deg.)
r, \(\theta, z\) : Radial, Rotational, Axial Direction, respectively
\(\zeta\) : Nondimensional Flow Resistance
\(\phi\) : Nondimensional Flow Rate