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AN ANALYSIS OF THE FLOW AND SOUND SOURCE OF AN ANNULAR TYPE CENTRIFUGAL FAN

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ABSTRACT

A centrifugal fan, widely used in home appliance electrical machines causes one of the serious noise problems. In general, dominant noise spectrums of the fan are discrete tones owing to tongue in the case of the blower type fan. However, in the case of centrifugal fan with annular casing, not only the discrete tone noise but also broadband noises are dominant. From the point of analysis previous researches have been focused on the generation and the reduction of discrete tones. In order to understand the generation mechanism of the discrete tone/broadband noise, a detailed unsteady flow field should be known. A DVM(Discrete Vortex Method) is used to calculate the flow field and the Lowson's equation is used to predict the acoustic pressure. It is found that the broadband noises of the annular type centrifugal fan are owing to unsteady force fluctuations related to the vortex shedding. The unsteady force fluctuation around the rotating impeller blades is due to interaction with the annular type exit in the centrifugal fan.

1. INTRODUCTION

Centrifugal fans are widely used due to their ability to achieve relatively high pressure ratios in a short axial distance compared with axial fans. Because of their widespread use, the noise generated by these machines causes one of serious problems. In general, centrifugal fan noise is often dominated by tones at BPF(blade passage frequency) and its higher harmonics. This is a consequence of the strong interaction between the periodic flow discharged radially from the impeller and the cutoff. But in a vacuum cleaner fan, the noise is dominated by not only the discrete tones of BPF but also broadband frequencies. It is generally suggested that broadband noises are generated by trailing edge/flow interaction, turbulent boundary layers and separated flow on the impeller blade and a housing.

Previous researches have been focused mainly on the generation and reduction of the BPF tones.[1,2] In order to understand the broadband noise, detailed information about the flow field around the impeller and the casing are very important.[3] The objective of this study is to understand the mechanism of broadband noise and to develop a prediction method for the unsteady flow field and the acoustic pressure field associated with the centrifugal fan. A DVM(discrete vortex method) is used to model the annular type centrifugal fan and to calculates the flow field. Lowson's method is used to predict the acoustic pressures. A single impeller and the overall fan with diffuser and casing are analyzed.

2. NUMERICAL METHODS

2-1. An Analysis of the Flow

A centrifugal fan is composed of three parts ; impeller, diffuser and casing as show in Figure 1. In figure 1 Q means the inlet flow rate, Si means the impeller blade, D means the diffuser blade and C indicates the casing. The impeller transforms the mechanical energy to the flow, the diffuser recovers the pressure through diffusing process and the casing collects and redirects the flow. We assume that the impeller rotates with a constant angular velocity and the flow is incompressible and inviscid. The impeller has NB number of blades and each blade has nc number of elements. Bound vortices are located at 1/4 point of each element and control points are taken at 3/4 point. Wake vortices are shed at the trailing edge of the impeller and the diffuser at every time step to satisfy the Kelvin's theorem. Shed vortices are convected with the local induced velocity. The inlet flow is modeled by a point source located at the center of the fan. The casing is modeled with constant source panels where the control point is taken at the center of the element. The suction in the exit to the axial direction is modeled with sink panels where the total strength is taken as the same as the inlet flow rate. Point vortices passing the exit are removed.

The induced velocity at $\vec{x}_{cj}$ is as below.

$$\vec{U}(\vec{x}_c;t) = \vec{U}_Q(\vec{x}_c;t) + \vec{U}_{bv}(\vec{x}_c;t) + \vec{U}_{wv}(\vec{x}_c;t) + \vec{U}_{sp}(\vec{x}_c;t) \tag{1}$$

The first term of right hand side represents the velocity at $\vec{x}_{cj}$ induced by the point source, the second term represents the velocity induced by bound vortices of the impeller, the third term represents the velocity induced by wake vortices and the fourth term represents the velocity induced by source panel.

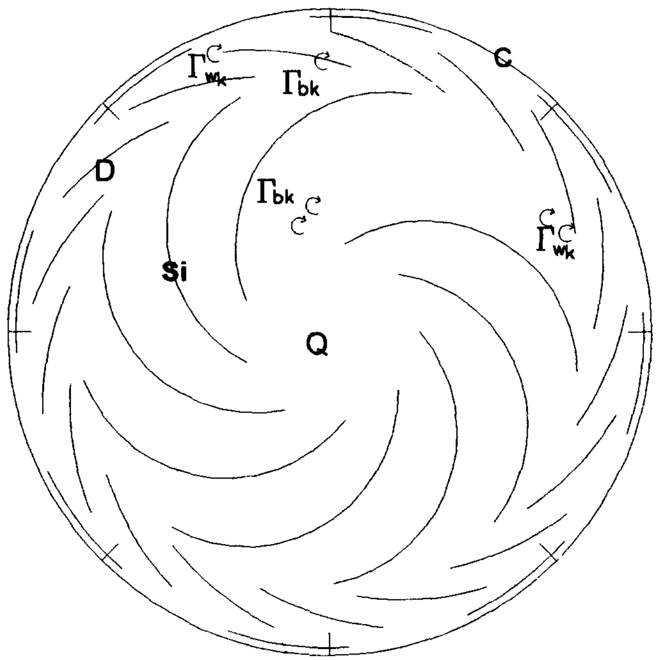

Figure 1. Configuration of the annular type centrifugal fan

The unknown strength of bound and wake vortices and source panels are calculated with the no-flow condition at the control point of each element and the Kelvin's theorem.[4]

$$\begin{aligned} g(\vec{x}_c;t)_j &= \vec{U}(\vec{x}_c;t)_j \cdot \vec{n}(\vec{x}_c)_j \\ &= \begin{cases} \Omega\,(\vec{n}(\vec{x}_c)_j \times \vec{x}_{cj}(t)), & \vec{x}_{cj}(t) \in S_i(t) \\ 0, & \vec{x}_{cj}(t) \in C \end{cases} \quad i = 1,\ldots,Z \end{aligned} \tag{2}$$

$$\begin{aligned} \left[\vec{U}_Q(\vec{x}_c;t)_j + \vec{U}_{bv}(\vec{x}_c;t)_j + \vec{U}_{wv}(\vec{x}_c;t)_j + \vec{U}_{sp}(\vec{x}_c;t)_j\right] \cdot \vec{n}(\vec{x}_c)_j \\ - g(\vec{x}_c;t)_j = 0 \end{aligned} \tag{3}$$

$$\frac{D\Gamma_m(t)}{Dt} = 0$$

$$\left[\sum_{k=1}^{nc}\Gamma_{bk}(t) + \sum_{k=1}^{nv}\Gamma_{wk}(t)\right]_m = 0 \tag{4}$$

Where, Γ_m is the total circulation of that blade. This total circulation is the sum of the circulation of bound vortices(Γ_b) and wake vortices(Γ_w).

The force of each element on the blade is calculated by the unsteady Bernoulli equation.

$$\vec{F}_{nj} = \rho\left\{\vec{U}(\vec{x}_c)\cdot\vec{\tau}_j\frac{\Gamma_{bj}}{\Delta s_j} + \frac{\partial}{\partial t}\sum_{k=1}^{J}\Gamma_{bk}\right\}\Delta s_j \tag{5}$$

Where, F is the normal force of the element and τ is the tangential vector.

2-2. An Analysis of the Noise

In 1965, Lowson derive the formulation of predicting the acoustic field generated by the moving point force from the wave equation.[5]

$$\rho - \rho_o = \left[\frac{x_i - y_i}{4\pi a_o^3 r^2(1-M_r)^2}\left\{\frac{\partial F_i}{\partial t} + \frac{F_i}{1-M_r}\frac{\partial M_r}{\partial t}\right\}\right] \tag{6}$$

Equation (6) indicates that the acoustic pressure of the moving point force is calculated using the time variation of force and the acceleration. By applying this equation to each blade elements we can predict the acoustic pressure in the free field. Therefore, the effect of the casing is not considered in this acoustic analysis. Also, the effects of the scattering, reflection and refraction are not considered. Only the behavior of the noise source can be estimated.

3. NUMERICAL RESULTS

3-1. An Analysis of an Single Impeller

The fan used in this research is a vacuum cleaner fan having impeller, diffuser and annular casing. The impeller has 9 blades and rotates 30,000 rpm.

The inlet diameter of the impeller and angle is 0.039(m) and 41.5 degree, the outlet diameter and angle is 0.109(m) and 10.0 degree. The flow rate is 1.36~2.526($m^3/\min$). Calculated mean head is 1870(m) and the pressure is 2247mmAq. Figure 2 shows the variation of shed vortex strength at each blade for a single impeller only. The strength of each wake vortices converges into a fixed value for this impeller without the casing. Because of the constant rotating force, discrete tones at BPF are generated. Discrete spectrums are shown in figure 3.

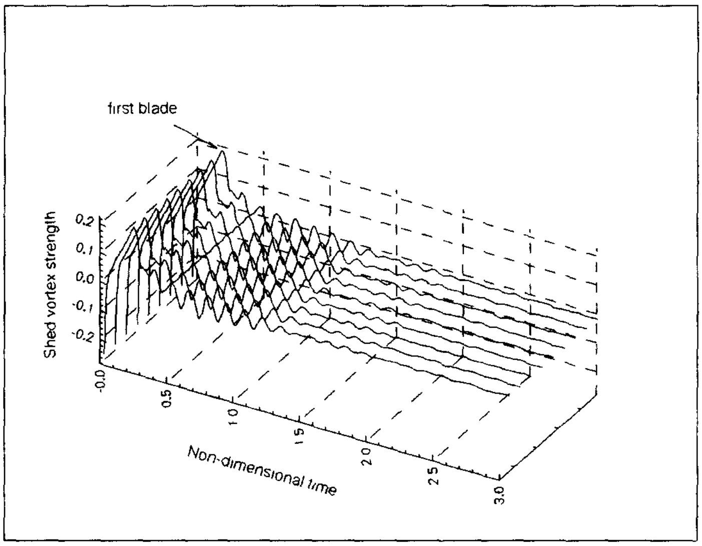

Figure. 2 The variations of shed vortex strength at each blade tip

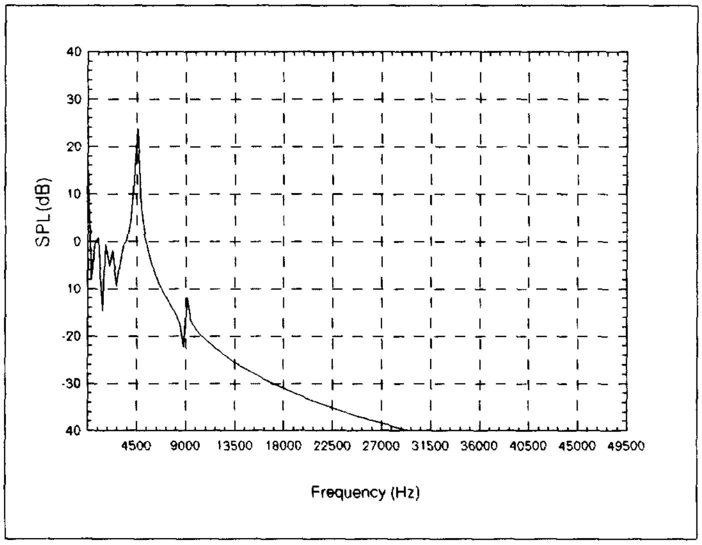

Figure. 3 Predicted Acoustic signal at 1.15m apart from the impeller

3-2. An Analysis of the Vacuum Cleaner Fan with Diffuser and Casing

The centrifugal fan used in this research contains 9 bladed impeller, 17 bladed diffuser, 8 exits and a circular casing. Wake vortices are shed at each trailing edge of the impeller and diffuser. Variations of the shed vortex strength at the impeller blades are shown in figure 4. Similar variations are observed in diffuser blades.

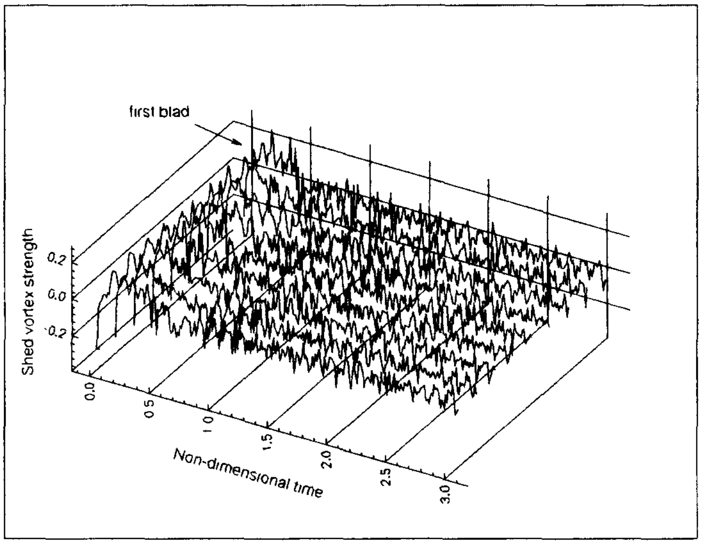

Figure. 4 Variations of shed vortex strength at the impeller

The unsteady fluctuating force is the dominant sound source in this case. Each blade has different pattern of the unsteady force fluctuation in this configuration having different numbers of diffuser blades, impeller blades and the exit. Figure 5 shows the predicted acoustic pressure at the point of 1.15(m) apart from the fan. It is clearly shown that the dominant noise characteristics of the fan are broadband. This is due to the highly fluctuating unsteady force not correlated to the rotating speed compared with the steady rotating force. Unsteady force fluctuations around the impeller blades is due to the unsteady vortex shedding from the blade interacting with the diffuser blades and the exits the exit of finite length on the wall of the casing. in the axial direction.

4. CONCLUSIONS

The prediction method to identifying the noise source from a vacuum cleaner centrifugal fan noise has been developed. To predict the sound field, unsteady flow field and unsteady force fluctuations are calculated by the DVM. From the results we found that the broadband noise of an annular type centrifugal fan is due to the unsteady force fluctuation around the impeller blades related to the vortex shedding from the blade. Unsteady forces associated with the shed vortices are generated by interaction with diffusers and exits on the wall of the casing.

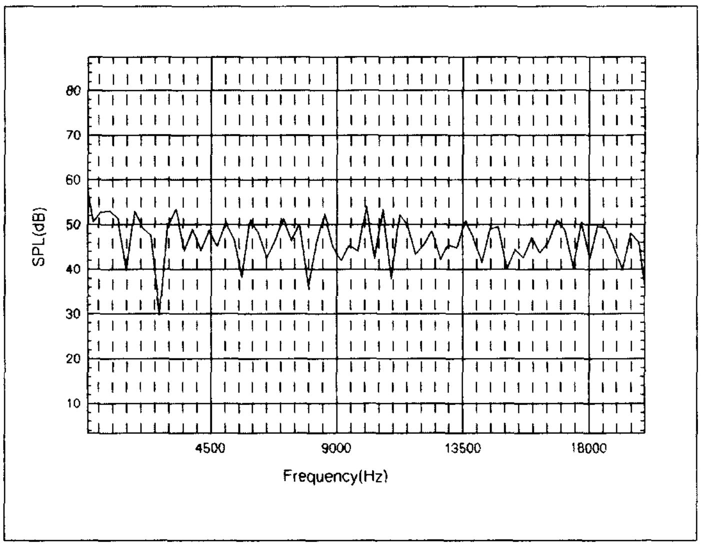

Figure. 5 Predicted acoustic signal at 1.15m apart from the fan

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