inter.noise 2000
The 29th International Congress and Exhibition on Noise Control Engineering
27-30 August 2000, Nice, FRANCE

I-INCE Classification: 1.1

NOISE OF PROPELLER FANS USED IN AIR-CONDITIONING UNITS

A. Guedel

CETIAT, BP 2042, 69603, Villeurbanne Cedex, France

Tel.: 04 72 44 49 74 / Fax: 04 72 44 49 49 / Email: alain.guedel@cetiat.fr

Keywords:
FAN, BROADBAND NOISE, AEROACOUSTICS, TRAILING EDGE NOISE

ABSTRACT
Fans and compressors are the two main noise sources of air-conditioning equipment. This paper deals with the noise radiated by propeller fans of external units of split systems. These fans usually have 3 to 5 large chord blades and are mounted within the unit in front of a heat exchanger. The objective of this work is to improve the knowledge on the acoustic mechanisms of this kind of fans in order to predict and reduce the noise level, especially the broadband noise level which provides the major contribution to the A-weighted OASPL since the BPF tones occur at frequencies usually much lower than 100 Hz. After a short presentation of the main aerodynamic and acoustic characteristics of these fans the paper shows results of tests performed on two 400-mm propeller fans of quite different geometry. These fans have nearly the same performance but their sound power spectra, which depend on the operating point, show some differences which are analyzed. Tests also include the determination of mean flow velocity profiles in a plane closely downstream to the blade trailing edge in order to assess the repartition of the mean load and mean flow along the fan span and try to correlate this information with acoustic performance. Furthermore the measured profiles are compared with those predicted by Fluent code. This comparison appears quite good and confirms that CFD codes are well adapted today for fan performance and mean flow prediction. Flow velocity measurement with a fixed hot film downstream the trailing edge allows to determine the blade wake width at several radiuses along the fan span. These data are used to check the validity of the sound power level prediction from the analytical model of Fukano. This prediction on both fans proves not too bad provided that the tip chord length is accounted for in the model.

1 - INTRODUCTION

Compressors and fans are the two main noise sources of air-cooling and air-conditioning equipment. Here, we are interested in the noise from fans of external units of split systems. They are propeller-type axial fans, wall-mounted facing the condenser. Their nominal pressure is low (less than 100 Pa) and in general the rotation speed does not exceed 1000 rev/min. They have 3 to 7 blades, with a chord which is more or less wide at the tip. In order to make a significant reduction in the aerodynamic noise of these fans it is important to have a better understanding of the mechanisms giving rise to the noise.
We are interested exclusively in the broadband noise radiated by the fan, which makes a much higher contribution to the A-weighted overall sound power level than the rotational noise. The main sources of broadband noise are classically (e.g. see [1], [2]):

- inflow turbulence noise
- self noise, which can be broken down into several mechanisms: turbulent trailing edge noise, blunt trailing edge (or vortex shedding) noise, laminar boundary layer instability noise, tip vortex noise and broadband noise related to flow separation

As we will see, turbulent trailing edge noise and flow separation noise are the main noise mechanisms for these fans. Inflow turbulence noise does not seem to play an important role for wide-blade propeller fans. Noise emission due to laminar boundary layer instabilities and blunt trailing edge vortex shedding results

in a high-frequency narrow band noise that is not generally observed with these fans. The contribution made by tip vortex noise has not yet been clearly established.
The purpose of this paper is to compare the experimental results obtained on two 400 mm propeller fans, enabling certain tendencies to be established as well as future avenues of research.

2 - EXPERIMENTAL RESULTS

A large number of aerodynamic and acoustic tests were carried out on two propeller fans, called *fan A* and *fan B* respectively. Fan A is a prototype fan designed by CETIAT for the project. Fan B is a commercially available fan, usually fitted on similar applications.

2.1 - Geometric characteristics of the fans

	Fan A	**Fan B**
impeller diameter (mm)	405	395
number of blades	3	5
hub/tip ratio	0.3	0.25
tip chord length (mm)	275	135
tip axial chord length (mm)	125	65
shroud width (mm)	30	87
tip clearance (mm)	10	5
blade thickness (mm)	2	2

Table 1: Geometric characteristics of the fans.

Figure 1 shows a view of Fan A taken from downstream.

Figure 1: Fan A (outlet view).

2.2 - Flow rate-pressure curves

Figure 2 compares the flow rate-pressure curves for the two fans. Fan similarity laws were used to convert the measured curves to a rotation speed of 1,000 RPM and a reference diameter of 0.5 m. The aerodynamic performances of the two fans are very similar, the best efficiency point being located around 5000 m^3/h. The curve for fan A was compared to a curve calculated using the CFD Fluent code. The comparison proved to be very good over the whole flow range.

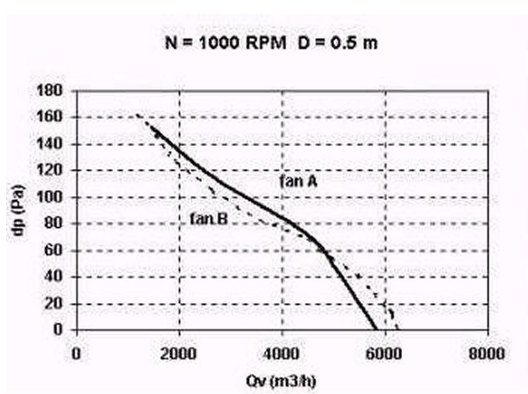

Figure 2: Fan performance curves.

2.3 - Noise spectra

The acoustic measurements on the fans were made in a reverberating room by mounting the fan on a mylar plenum (ISO 10302). In this way the total (inlet+outlet) sound power level of each fan was determined for each operating point. The evolution of the A weighted overall noise level as a function of the operating point was classic with a minimum level at the best efficiency point for both fans. The lowest sound power level at the reference conditions was 72.3 dB(A) for fan A and 73.5 dB(A) for fan B, which are relatively low levels for this type of fan.

The evolution of the spectra as a function of the operating point show similar tendencies for both fans. This evolution is shown in diagram form in Figure 3. At low flow rates the flow is separated at the hub, which results in a noise increase over the whole frequency range. Conversely, in the zone around the optimum performance the spectrum is the lowest for all the frequencies. In the flow rate on either side of the optimum performance, an increase in noise level is observed compared to the best efficiency at low frequencies (below 1 to 2 kHz depending on the fan). At higher frequencies the spectra almost merge.

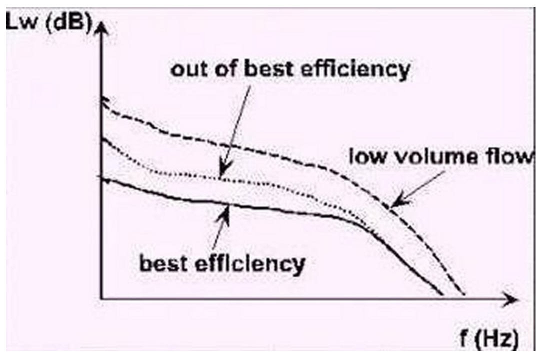

Figure 3: Evolution of the noise spectrum with operating point.

Before interpreting the results, let's compare the sound power spectra of the two fans at best efficiency. The comparison of the spectra (figure 4) reveals practically identical levels up to 1 kHz and a significant difference at high frequencies, narrow-blade fan B being noisier than wide-blade fan A. When the same spectra are represented by multiplying the frequency scale by the tip chord length ratio, the high frequency sections of the two spectra become superimposed (Figure 5).

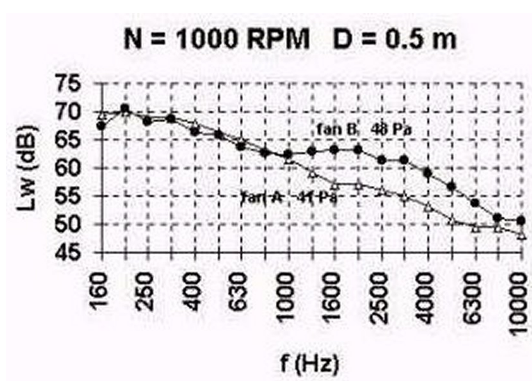

Figure 4: Measured fan spectra at best efficiency.

3 - ANALYSIS AND CONCLUSIONS

The analysis of the noise spectra reveals at least three distinct mechanisms as a function of the operating point. A large increase in noise level is observed in the low flow rate zone over the whole spectrum, which is classic for axial-flow fans and which is due to flow separation or even reverse flow at the hub due to a large radial component of the flow velocity at the fan outlet. At other flow rates, the difference at low frequencies between the spectra measured around the best efficiency point and elsewhere is probably due to what Blake [3] called broadband noise related to loading, which is produced when the angle of attack is increased and which is accompanied, by local flow separation on the blades.

At higher frequencies, here typically above 1 to 2 kHz, the main mechanism is the trailing edge noise due to the passage of the blade turbulent boundary layers past the trailing edge. This mechanism is

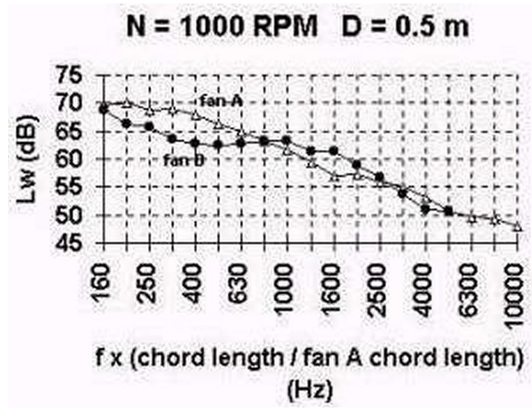

Figure 5: Fan spectra at best efficiency (frequency scaled by blade chord length).

particularly influenced, as seen here, by the blade chord length. Fukano's well known model was used to try to predict the overall sound power level linked to this phenomenon. The thickness of the blade wake immediately downstream of the impeller, which constitutes important data for the implementation of this very simple model, was measured on both fans. The confrontation between the measured and predicted overall noise levels is not bad providing the spectra are integrated above 1250 Hz for fan B and above 630 Hz only for fan A, that is to say by choosing a lower limit of integration frequency range corresponding to a constant Strouhal number based on blade chord length and tip speed. Fundamental work is on-going to predict the two latter mechanisms. The first mechanism observed in the low flow area should be avoided in all events on these fans.

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