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ONE DESIGN OF A SILENCER FOR ATTENUATING FAN NOISE

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Abstract

The paper introduces one design of a silencer for attenuating fan noise and the implementation result. A denoising experiment was carried out on the intake and discharge noise of the fan which is used to cool a certain apparatus in an aircraft application. The silencer parameters are calculated in detail and comparative measurements are carried out. The experimental results show that the silencer can efficiently attenuate the noise in a wide frequency band.

1. INTRODUCTION

There is a small fan used to cool a certain apparatus by the ventilation in an airplane. Because of the fan blades' revolving at the high frequency and the asymmetrical flow caused by the restriction of flow passage, the unbearable noise have influenced the pilot's work seriously. In order to reduce the noise produced at the inlet and the outlet opening of the fan, some suitable measure must be adopted.

In this paper, a cylinder shape silencer combined microperforated structure and sound-absorbing materials is designed and is applied to attenuate the fan noise. Dissipative silencers attenuate sound by converting the acoustical energy propagating in the passage into heat caused by friction in the voids between the oscillating gas particles and the fibrous sound-absorbing materials. And microperforated plate silencers dissipate the acoustical energy by making use of the air viscosity resistance in the resonating structure. Dissipative silencers could attenuate the noise at a wide frequency band and microperforated plate silencers could efficiently attenuate the noise at the peak in the frequency domain. Hence, the combined silencer has the wider elimination frequency band and the stronger elimination ability.

2. THE ANALYSIS OF THE FAN NOISE

The experiment was carried on in the common laboratory environment. The propeller fan unit consists of an impeller (has four blades), an electric circuit board and a cover. The measure system includes the precision sound level meter—A/D—the computer. A diagram of the set up can be seen in Figure 1: Measure System.

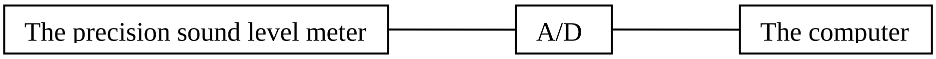

Figure 1. Measure System.

The arrangement of the measuring point should not only get the frequency spectrum characteristic of the noise but also avoid the impact of the exhausted jet stream at the outlet as far as possible. Therefore the measuring point is set at the position of 1 m slanted 45° above the outlet opening of the fan unit. The frequency spectrum of the noise at the measuring point is shown in Figure 2: Noise Frequency Spectrum of the Fan Unit.

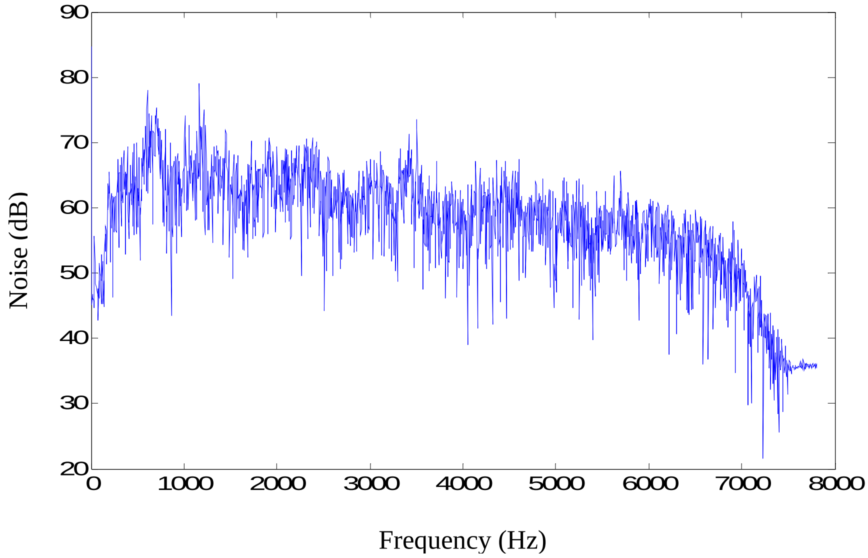

Figure 2. Noise Frequency Spectrum of the Fan Unit.

As can be seen in Figure 2, the frequency spectrum characteristic of the noise assumes the broadband, and has the marked peak. The frequency of the noise peak value is 1494.1 Hz. The noise at the measuring point is 88.7 dB in average measured by the precision sound level meter.

In the lab environment, the noise produced by the fan mainly is the aerodynamic noise that comes from the radiation of the inlet and the outlet. The aerodynamic noise consists of revolving noise, vortex noise and jet noise. In view of the outlet flow will transfer in the hosepipe used to connect the outlet and the equipment at the time of installed the fan unit in the airplane, we mainly consider revolving noise and vortex noise when we design the silencer. Moreover, because the inlet of the fan is exposed in the air, it is the main noise source.

2.1 Revolving noise

Revolving noise produced by the pressure pulse formed for the relative movement between the revolving fan blades and the air. For a point in the space, the gas pressure will rapidly fluctuates once when a blade passes by, and then produces a pressure pulse and radiates noise to the entourage.

Revolving noise frequency f_1 (Hz) is relative to the fan rotational speed n (r/min) and the number of blade z. The base frequency can be calculated by Eq. (1).

$$f_1 = \frac{n \cdot z}{60} \tag{1}$$

In addition to the base frequency, there are harmonic frequencies. $f_2 = 2f_1$、 $f_3 = 2f_2$ …。 The noise intensity of base frequency is biggest. And the intensity will become weak as the order of the harmonic frequency increases.

2.2 Vortex noise

Vortex noise is the unstable flow noise. When the gas passes by the revolving blade and splits at the interface of the blade, a series of vortexes will be formed due to the air viscosity.

Vortex noise frequency f_e (Hz) can be calculated by Eq. (2).

$$f_e = K \cdot \frac{V}{D} \tag{2}$$

Where K is Strouhal number, the numerical value is 0.15~0.22. V is the relative speed between the gas and the blade, m/s; and D is the thickness of the gas incidence direction, m.

Because vortex noise frequency lies on the relative speed between the gas and the blade, and that the blade circle speed lies on the circle radius and the spot speed of the each point in the blade. Thus vortex noise produced by the fan is stochastic, and generally shows a wide band spectrum.

3. DESIGN OF SILENCER

The fan unit is installed on the equipment shelves in the cockpit. The space in the fan unit is so limited that we hardly reduce the noise in it directly. The distance from the upper airflow pipe to the outlet opening of the fan unit only has 28mm, and the distance from the lower cover of the equipment shelves to the inlet opening has 259mm. Furthermore, there are lines, pipelines and brackets around the inlet and the outlet. Although some methods reducing the fan noise had been investigated, they were found to be hard to apply to this case. No matter based on the basal analysis of the fan noise characteristic or job site situation, the silencer only allowed to install under the fan unit (the inlet opening).

The main basis for the design of the silencer is acoustic performance design of the silencer and the frequency spectrum characteristic of the source noise. The design target for this silencer is that the noise can be reduced to about 80 dB in average at the measuring point and the pressure loss must be less than 20%. As mentioned above, the fan noise is the broadband noise, covering the wide frequency range. Well then, if we want to achieve the good result, the silencer should be designed to be covered 200~6000 Hz (as can be seen in Figure 2).

There are a lot of kinds of silencers to reduce aerodynamic noise, including dissipative silencers, reactive silencers and microperforated plate silencers etc. Each kind of silencer all has its own characteristic and application respectively. For this fan unit, the stream velocity is very great, the spatial size for installed and the silencer's weight have the strict restriction. The analysis believed that, dissipative silencers could not satisfy the design target; reactive silencers are unable to be installed due to the spatial limitation; and microperforated plate silencers could not applied because of the narrow noise-reducing frequency band. After the analysis, and considering the question of the economical and practical aspect, we decided to adopt the cylinder shape silencer combined microperforated structure and sound-absorbing materials.

The silencer consists of the inner and outer sleeves. The outer sleeve is coiled by the stainless steel plate and the inner sleeve is coiled by the microperforated plate, and they are connected by two flanges. Between two sleeves, the superfine fiberglass wadding is filled in. Middle of the silencer having a strip shape of fiberglass wadding makes inner space divided into two. The application of fibrous materials in this silencer requires a protective covering. The covering consists of perforated metal (perforation rate 20%) with fiberglass cloth behind. When the noise frequency exceeds 500 Hz, the absorption coefficient of the superfine fiberglass wadding will be more than 0.8. The experimental results show that: when the filling capacity is 0.1 kg ~0.3kg, the sound-absorbing effect is had best.

A diagram of the silencer structure can be seen in Figure 3: Diagram of the Silencer Structure.

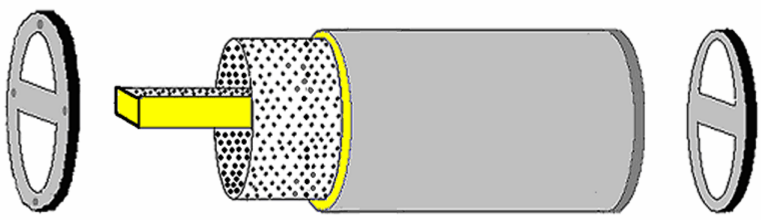

Figure 3. Diagram of the Silencer Structure.

The fan is made in USA. The inside diameter is 67mm, the outer diameter is 76mm, the height is 60mm. The normal rotational speed is 20300 r/min. The working voltage is 115V, the working current is 1.2A. The base frequency of the revolving noise calculated by Eq. (1) is 1486.7Hz.

The thickness、aperture 、perforation rate of the microperforated plate is 0.8mm、0.8mm、4% respectively.

The resonance structure frequency formula is given by Eq. (3).

$$f_r = \frac{c}{2\pi}\sqrt{\frac{p}{L_k D}} \tag{3}$$

Where D is the thickness (cm) of air gap behind the microperforated plate, p is the perforation rate, L_k is the effective length (cm) of the aperture. $L_k = t + 0.8d$, t is the thickness (cm) of the microperforated plate; d is the aperture (cm).

The frequency of the noise peak value is 1494.1 Hz. The thickness of air gap behind the microperforated plate can be calculated by Eq. (3). D = 30mm.

Then we can design the silencer's size. The silencer's length is 200mm; the inside diameter of inner sleeve is 68mm. So we can calculate that the inside diameter of outer sleeve is 106mm.The strip shape's size is 200mm×65mm×20mm.

4. EXPERIMENTAL RESULTS

Using this combined silencer, we have carried out the experiment in the laboratory. The frequency spectrum of the noise with the silencer at the measuring point is shown in Figure 4: Noise Frequency Spectrum of the Fan Unit with the Silencer.

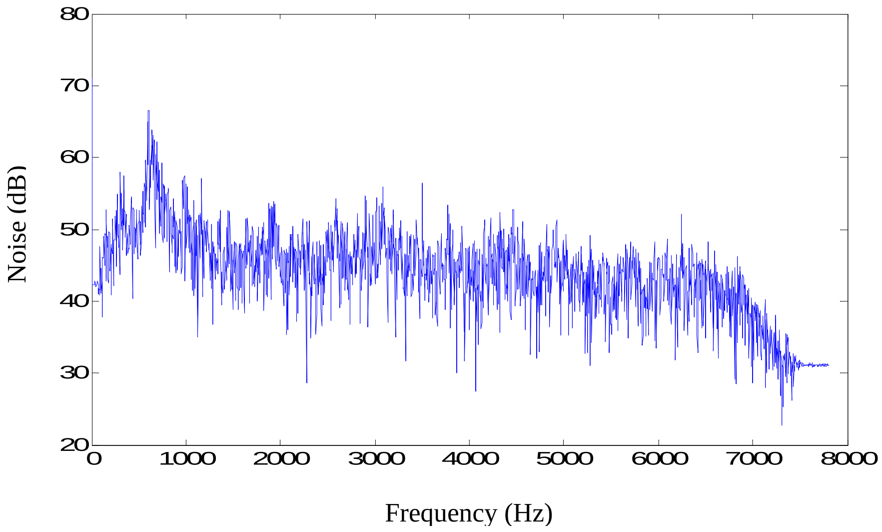

Figure 4. Noise Frequency Spectrum of the Fan Unit with the Silencer.

The comparison of 1/3 octave spectrum of the fan unit with the silencer and without the silencer is shown in Figure 5: Comparison of 1/3 Octave Spectrum of the Fan Unit with the Silencer and without the Silencer.

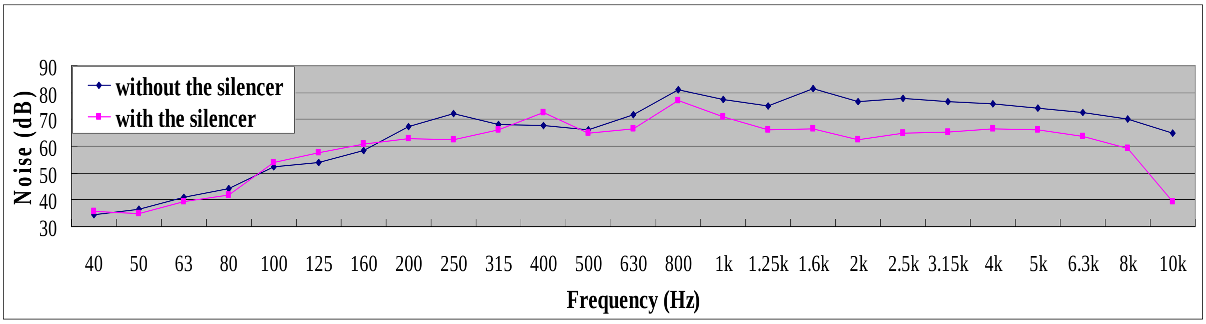

Figure 5. Comparison of 1/3 Octave Spectrum of the Fan Unit with the Silencer and without the Silencer.

As revealed in Fig. 4 and Fig. 5, in the wide frequency range, especially between 1000~6000 Hz, The effect of noise reduction with the silencer is very well. The results are consistent with the theoretical analysis and computation.

The measuring results indicated that the noise reduced about 8 dB in average at the measuring point.

5. CONCLUSIONS

The silencer has a simple structure, light weight, small volume, relatively little effect to the flow (the pressure loss is far less than 20%) and can run in the long time at the high speed. Withal, we can control the noise elimination frequency band and the noise elimination volume in a certain range by chose the structure parameters. Therefore, the silencer is suited to the specific condition of the fan unit.

The experimental results show that the silencer can efficiently attenuate the noise in a wide frequency band and reduce about 8 dB around the fan in the 1m scope.

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