Noise Reduction of the Engine Cooling Fan Used in Large Bus*

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Acoustic considerations of axial flow fans become imperative due to regulations and the ubiquitous call for environment-friendly products. This paper suggests a noise reduction method for the engine cooling fan. This engineering model has been developed for the prediction of the noise spectrum of an axial flow fan. The radiated acoustic pressure is expressed as the discrete frequency noise peaks at the BPF and its harmonics and line spectrum at the broad band by noise generation mechanisms. This paper shows a comparison of the measured and calculated noise spectra of the fan for the validation of the noise prediction program. This paper also presents the characteristics of fan noise due to modify the design parameters. Accordingly, design parameter values for noise reduction of the fan were obtained.

Key Words: Fan Noise, Discrete Frequency Noise, Broadband Noise, Taguchi Method

1. Introduction

A fan is a rotary machine which delivers kinetic energy to air or gas at a constant head increase to recover system losses⁽¹⁾. Particularly, as noise regulations for environment-friendly products is strengthen at home and abroad, the need to reduce fan noise has become more and more urgent.

Many theories have been developed in the course of studying rotor noise^{(2),(3)}. Nonetheless, an exact solution to this fluid interaction still has not been found to date. In many applications, one must still rely on empirical formulae to estimate the overall SPL values of the given fan models. Such estimation is often insufficient to characterize the noise performance, because the fan noise contains both narrow band and broadband sounds, and the latter has a masking effect and carries the majority of the acoustic energy. Hence, it is desirable to be able to predict

the spectrum of the engine cooling fan⁽⁴⁾.

The objective of this study is to reduce the cooling fan noise of an express bus. For this purpose, we must develop a computer model that will allow design engineers to estimate the fan noise, given the characteristic dimensions and working conditions of the fan. Determining the total sound power level is not enough to characterize the noise performance of the axial flow fan, because it has both narrow and broadband components. Within the narrow components, discrete frequency noise predominates with axial flow fans and is periodic in character, consisting of the fundamental blade passage frequency and many higher harmonics. The components are caused by different generated mechanism. Therefore, in order to improve the noise performance of the axial flow fan, it is necessary to be able to predict the spectrum, given the fan characteristic parameters. An engineering model has been developed to estimate the noise spectrum with Wright's⁽³⁾ and Wu's^{(4),(5)} methods. To validate this engineering model, the calculated noise spectra are compared with the measured ones of a cooling fan. Then, the simulation was preformed using statistical analysis by changing the fan design parameters.

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Nomenclature

- B : the blade number of a fan
 BPF : blade Passage Frequency
 $\vec{F}$: the force exerted on the fluid by the rotating blades
 F_L : the lift force per unit length
 L : loss function
 Q : shape factor for the turbulent boundary layer noise
 R : the distance between an observer and the source
 S_r : Strouhal Number
 S_i : sum of square of a factor i
 s_0 : the period along the azimuth
 SP_j : spectrum of the radiated acoustic pressure
 V : the mean flow speed
 V_i : mean square of a factor i (S_i/f)
 a : the chord length at a particular cross section of the blade
 $\vec{n}$: the unit normal vector on the surface
 $p(r, t)$: sound pressure measured by microphone
 r : fan radius
 y_j : characteristic value
 α : the decay rate of pressure pulse
 β : the blade angle
 ϕ : the azimuthal angle
 Ψ : the angle of the inflow direction
 ∇_x : gradient taken with respect to $\vec{x}$ -coordinate
 ξ : the coefficient from ξ_b
 ξ_b : solidity of the blade surface the ratio of the blade surface area
 θ : the polar angle

2. Modeling the Noise Spectrum of an Axial Flow Fan

2.1 Modeling theory

2.1.1 Modeling of the discrete frequency noise

The discrete frequency noise predominates axial flow fans and is periodic in character, consisting of the fundamental blade-passage frequency (BPF) and many higher harmonics. The fundamental frequency of BPF can be expressed as follows.

$$BPF = (U_r \times B) / 60 \quad (1)$$

where, U_r is the number of revolutions per minute and B is a blade number.

The discrete frequency noise is primarily caused by fluctuating forces exerted on the surrounding fluid medium by the rotating blades⁽⁶⁾. And the fluctuating pressure is correlated to the lift force per unit length acting on the fan blade, and is subsequently approximated by pressure pulses that decay both space and time. To simulate the discrete frequency noise according to Wu⁽⁵⁾, a shape function is designed which

breaks the frequency into consecutive bands centered at the blade passage frequency and its harmonics. Thus the radiated acoustic pressure is expressed in terms of superposition of contributions from these pressure pulses, and the line spectrum is obtained by taking a Fourier series expansion.

So first, the discrete frequency noise can express the radiated acoustic pressure from an axial flow fan in the following way

$$p = -\frac{1}{4\pi} \nabla_x \cdot \left[\frac{\vec{F}}{R} \right]_\tau \quad (2)$$

where, p is the radiated acoustic pressure, ∇_x implies a gradient taken with respect to x -coordinates, R is the distance between the observer and the noise source. and $[\]_\tau$ indicates that quantities inside are to be evaluated at the retarded time τ ($=t-R/c$). $\vec{F}$ represents the force exerted on the fluid by the rotating blades.

The fluctuating force can be approximated by a pulse that decays exponentially along the chord⁽⁷⁾ and the fluctuating pressure is correlated to the lift force per unit length F_L acting on the blade surface calculated by the Blasius theorem⁽⁸⁾.

Accordingly, the radiated acoustic pressure is expressible as a superposition of pressure pulses emitted from each section of the blade surface as flowing,

$$p = \iint \frac{R_0 \xi a F_L (\sin \Psi \cos \theta \sin \phi + \cos \Psi \sin \theta)}{4\pi R (1 - e^{-2\pi \alpha r/B})} \times \sum_{k=0}^{\infty} b_k \cos(kB\phi)$$

Fig. 1 Simplified model of axial flow fan in a free field

Fig. 2 Schematic of incident flow on an airfoil

$$\times \operatorname{Re} \left[\sum_{j=0}^{\infty} a_j \left(i \frac{2\pi j B f_0}{c} - \frac{1}{R} \right) e^{-i2\pi j B f_0(t-R/c)} \right] \vec{n} \cdot d\vec{A} \quad (3)$$

where, a solidity constant, ξ is between 0.006–0.01, Ψ represents the angle of the inflow direction, α indicates the decay rate of the pressure pulses, r is the fan radius and ϕ is the azimuthally angle, $\vec{n}$ is the unit normal vector on the surface, $\vec{A}$ is the blade surface area.

And the coefficients a_j and b_k are given by Burinton⁽⁹⁾ is below;

$$a_j = \frac{(2-\delta_{0j})}{t_0} \int_0^{t_0} e^{-2\pi f_0 a r \tau} \cos \left(\frac{2\pi j \tau}{t_0} \right) d\tau \\ \approx \frac{(2-\delta_{0j}) B}{2\pi} \frac{(a r)(1-e^{-2\pi a r/B})}{(a r)^2 + (j B)^2} \quad (4)$$

$$b_k = \frac{(2-\delta_{0k})}{s_0} \int_0^{s_0} e^{-a r \phi} \cos \left(\frac{2\pi k \phi}{\phi_0} \right) d\phi \\ \approx \frac{(2-\delta_{0k}) B}{2\pi} \frac{(a r)(1-e^{-2\pi a r/B})}{(a r)^2 + (k B)^2} \quad (5)$$

To simplify the numerical computations, it is assumed that $R \gg r$, ϕ is non-zero only for the when $k=0$ and the polar angle $\theta \rightarrow 90^\circ$, the spectrum of the radiated acoustic pressure can be obtained from Eq.(3).

$$p = \sum_{j=0}^{\infty} \operatorname{Re} (S P_j e^{-i2\pi j B f_0 t}) \quad (6)$$

where $S P_j$ represents the amplitude of the j th narrow band of the noise spectrum.

$$S P_j = \int_{r_h}^{r_t} \frac{\xi B^2 F_L a r \cos \Psi \sin \theta}{8\pi^2 R_0} (1-e^{-2\pi a r/B}) \\ \times \left(i \frac{2\pi j B f_0}{c} - \frac{1}{R_0} \right) \times \frac{(2-\delta_{0j}) e^{i2\pi B f_0 \sqrt{R_0^2 + r^2}/c}}{(a r)^2 + (j B)^2} dr \quad (7)$$

where δ_{jk} is the Kronecker delta and $F_L = C_L a \rho V_0^2/2$

Equation (7) indicates that the overall shape of the spectrum is controlled by α the decay rate of the pressure pulse acting the blade surface, which is determined a curve fitting of the experimental data. And Eq.(7) chooses to focus attention on the one that influence the predominant feature of the noise spectrum of an axial flow fan in a free field which the fluctuating pressure exerted on the fluid medium by the fan blade.

2.1.2 Modeling of the broadband noise Broadband noise is due to vortex shedding from the leading and trailing edges of a rotor. This noise is broadband in character, i.e., with no discrete frequency components, which for typical axial flow fans used in road vehicles is of relatively low intensity. The broadband noise has several components, laminar boundary layer noise, turbulent boundary layer noise and vortex noise, etc. However, we focused on the turbulent boundary layer noise for the modeling of the broadband noise in this research because the turbulent boundary layer noise dominates at the interested fre-

quency region in view of experimental results so far achieved.

Generally, according to the study by Wright, the typical broadband spectrum is a broadband excrescence symmetrically placed about a peak frequency f_{pt} when measured with a suitable constant bandwidth analyzer and plotted on a logarithmic frequency scale.

The peak frequency, f_{pt} , in its simplest form can be written as Eq.(8).

$$f_{pt} = S_t (V_t/a) = S_t \{N/(\rho_a)_{\text{actual}}\} \quad (8)$$

where V_t is the blade tip speed, a is the chord length, S_t is the Strouhal Number (presumed constant with respect to changes in V_t and a) and $(\rho_a)_{\text{actual}} (=a/2\pi r_t)$ is the actual chord solidity. And r_t is fan radius. The typical value of the peak frequency Strouhal number for a fan with a typical 5% thickness to chord ratio is approximately unity and a slightly lower value of about 0.8 is found for a fan with about 10% thickness to chord ratio. Lower values of S_t are found as the blade incidence angle is increased.

An equation for the peak value of the broadband excrescence in dB can be written,

$$SPL = 10 \log \frac{\Delta f V_t^2 (e^{0.46\alpha/a^2}) T_r^2 \sin^2 \sigma}{R^2 a B} - 30 \text{ dB} \quad (9)$$

where Δf is the width of the excrescence at the 20 dB down point from the peak, α is the incidence angle, T_r is the thrust and σ indicates the elevation angle from the observer.

The thrust can be calculated due to know the lift coefficient.

$$T_r = 0.5 \rho A V_t^2 C_L \quad (10)$$

$$A = 0.33 B \cdot a \cdot r_t \quad (11)$$

$$C_L = 2 \pi \alpha \quad (12)$$

where C_L is the lift coefficient, ρ is the density of air and A indicates the effective total blade area of the rotor.

2.2 Modeling the noise spectrum

In carrying out the modeling, the blade is discredited in the radial direction. The computer program used for this modeling is written in Matlab 6.1. The input data to this program includes: the number of blades, the hub and tip radius, the hub and tip chord length and the corresponding blade angle. The output of the program includes the noise spectrum at the measurement point and the overall sound pressure value in decibels.

In order to validate the modeling of the fan noise, comparison of the calculated and measured noise spectra is performed. The fan is 0.65 m in diameter with nine blades operating at around 1720 rpm.

Figure 3 demonstrates a comparison of the calculated noise spectrum (dashed line) and the measured one. A favorable agreement between the calculated

Fig. 3 Comparison of the calculated and the measured noise spectra of the bus fan

(..... : measurement, — : prediction)

	Measured value	Calculated value
Overall value	71.57 dB	71.43 dB

and measured noise spectra was obtained. The SPL overall values of the calculated and measured noise spectra were 71.57 dB and 71.43 dB respectively showing that the overall agreement is quite good.

3. Simulation through the Changing of the Design Parameters

In this section, Taguchi method has been used to find out optimal design parameters for low noise performance. Also, the statistical analysis module was connected to a fan noise simulator for understanding the effectiveness of the design parameters. To use the statistical approach in designing and analyzing an the experiment, it is necessary that everyone involved in the experiment has a clear idea in advance of exactly what is to be studied, how the data are to be collected, and at least a qualitative understanding of how these data are to be analyzed. An outline of the recommended procedures is shown in Table 1⁽¹⁰⁾.

Control factors were selected from seven input parameters of the fan noise simulator. Table 2 shows the chosen factors and the levels for each factor. To consider the cooling performance, regions of interest for each factor were determined so that there would be selected to have small variation from the initial values: The response variable is a calculated overall value from predicted noise spectrum.

We found that interaction between the factors does not exist by pre-simulation. Accordingly, the simulation of parameter design utilized an orthogonal array design, $L_{18}(2^1 \times 3^7)$. We arranged the first column an error by pulling. After running the simulation, we obtained eighteenth response variables as shown in

Table 1 Shows the guidelines for designing an experiment

1	Recognition of and statement of the problem
2	Choice of factors and levels
3	Selection of the response variable
4	Choice of experimental design
5	Performing the experiment
6	Statistical analysis of the data
7	Conclusions and recommendations

Table 2 Factors and levels of a prototype fan

Control factors		Levels		
		0	1	2
B	Tip diameter[m]	0.637	0.65	0.663
C	Hub diameter /Tip diameter	39%	41%	43%
D	Fan blade number	8	9	10
E	Chord length at hub[m]	0.095	0.11	0.125
F	Chord length at tip[m]	0.175	0.19	0.215
G	Setting angle at tip[degree]	28.2	30	31.8
H	Setting angle at hub[degree]	42.3	45	47.7

Table 3 Overall SPL values

Exp. No.	Overall SPL (dB)	Exp. No.	Overall SPL (dB)
1	71.6820	10	71.8468
2	70.7179	11	69.7343
3	71.1751	12	72.0854
4	71.3370	13	72.9980
5	71.9598	14	72.5595
6	71.9179	15	69.7238
7	72.3246	16	73.1442
8	73.5635	17	73.0483
9	73.0567	18	72.7232

Fig. 4 Simplified model B (Tip diameter), C (Hub diameter/Tip diameter), D (Fan blade number), E (Chord length at hub), F (Chord length at tip), G (Setting angle at tip), H (Setting angle at hub)

Table 3.

4. Estimation of the Model Parameters and Low Noise Design

To determine the influences of the selected design factors and find the optimal values, we performed on analysis of variance and a factorial effective analysis. The analysis of variance is shown in Table 4.

As can be seen in Table 4, the number of degrees of freedom of a sum of squares is equal to the number of independent elements in that sum of squares. Generally, the number of degrees of freedom for the factors is the number of level-1, for total the sum of squares it is the number of experiments-1 and for the error sum of squares, it is the difference between the total degrees of freedom and the sum of degrees of freedom of the factors. In this paper, the total sum of squares is

$$SS_T = \sum_{i=1}^{18} y_i^2 - \frac{\left(\sum_{i=1}^{18} y_i\right)^2}{N} \quad (13)$$

where y_i is the response variable and N in the number of experiments.

The mean square of factor C is found as follows,

$$SS_C = \frac{b}{n} [C_0^2 + C_1^2 + C_2^2] - \frac{\left(\sum_{i=1}^{18} y_i\right)^2}{N} \quad (14)$$

where $C_0 = \sum_{at \ C_{level0}} y_i$, $C_1 = \sum_{at \ C_{level1}} y_i$, $C_2 = \sum_{at \ C_{level2}} y_i$ and b is the number of levels of factor C . Mean squares of other factors are expressed by this formula.

The sum of squares of error is the difference between the total sum of squares and the sums of squares of factors. The mean square of each factor can be calculated as Sum of square/Degree of freedom and F ratio, F_0 , is the Mean square of factor/mean square of error.

To distinguish the effectiveness of the factors, the F -value for the factors was compared to the criterion (0.01%, 30.8). Accordingly, the null hypotheses for all factors were rejected. The result shows that they

Table 4 Analysis of variance for overall sound pressure

Source of variation	Sum of squares	Degrees of freedom	Mean square	F_0
B	9.8662	2	4.9331	6490.7886
C	0.6054	2	0.3027	398.2678
D	2.8589	2	1.4295	1880.8416
E	1.2421	2	0.6210	817.1189
F	4.3951	2	2.1976	2891.4628
G	2.0356	2	1.0178	1339.1749
H	0.1106	2	0.0553	72.7460
Error	0.0023	3	0.0008	
Total	21.1162	17		

affect the fan noise such that $B > F > D > G > E > C > H$ in sequence. This sequence is determined by the order of the sensitivity (magnitude difference/unit level), that is, the steeper slope of the factor becomes, the more sensitive. It means that the levels of each factors corresponding to the largest SN ratio become optimal values of the factors. Thus, we can obtain the minimal characteristic value satisfying the low noise performance from the above optimal parameters.

To assist in interpreting the results of this experiment, a factorial effect diagram was used. It is expressed as a graph of the average responses at each treatment combination. This graph is shown in Fig. 5. If the high level of the factor is chosen, sound pressure level of the fan due to the factor becomes lower. Figure 6 shows the comparison of the initial valued and optimal valued noise spectra of the fan. After changing the factors, the total SPL was reduced 2.72 dB.

Fig. 5 Factorial effect diagram

Fig. 6 Comparison of noise spectra of initial condition and optimal condition of the bus fan

(..... : initial, — : optimum)

	Initial condition	Optimum condition
Overall value	71.57 dB	68.85 dB

5. Conclusion

A computer model for estimating the noise performance of an axial flow fan is developed using this model, it is possible to predict the fan noise spectrum accurately by mixing two prediction methods. This engineering model was validated by comparing the results with experimental results. In order to extend this computer model to a cooling fan assembly, the effectiveness of other external factors, such as that shroud, radiator, and engine block must be based on the experimental method.

To detect the effectiveness of the design parameters, statistical analysis was used. Of the selected factors, fan radius, number of blades and the chord length sequence at the tip primarily affected fan noise. Finally, the total SPL was reduced 2.72 dB with the selected optimal set of design parameters.

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