An Improved Prediction Model of Vortex Shedding Noise from Blades of Fans (Postprint)

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Abstract

The main source of the noise of an axial flow fan is the fluctuating pressure field on blade surfaces caused by the shedding of vortices at the trailing edge of blades. An analytical model to predict the vortex shedding noise generated at the trailing edge of blades of axial flow fans was proposed by Lee in 1993. In this model, for mathematical convenience, an idealized vortex street is considered. However, the agreement between the analytical results and the experimental data needs to be improved because of the simplification about the Karman vortex street in the wake of blade. In the present study, a modified model is proposed based on the prediction model by Lee. The boundary layer theory is used to analyze and calculate the boundary layer development on both the pressure and the suction sides of blades. Considering the effect of boundary layer separation on the location of noise source, the predicted overall sound pressure level compares favorably with the experimental data of an axial fan. In the calculation of A-weighted sound pressure level (LA), considering the effect of static pressure on radiate energy, the predicted broadband noise with the modified model compares favorably with the experimental data of a multiblade centrifugal fan.

Full Text

Preamble

An Improved Prediction Model of Vortex Shedding Noise from Blades of Fans

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Introduction

Noise generated by fans has become a major concern in recent years due to its impact on working and living environments. For fans, aerodynamic noise dominates over mechanical noise caused by vibration and electromagnetic noise from the motor [1]. Lighthill [2] demonstrated that vortex shedding noise from the blade wake constitutes the primary noise source.

Several prediction models based on the Karman vortex street have been widely utilized for axial and centrifugal fan noise prediction. Sharland [3] proposed that broadband noise in axial flow fans arises from three mechanisms: surface pressure fluctuations from turbulent boundary layers, vortex shedding from rigid body surfaces, and incoming flow with initial turbulence. Lee [4] developed an analytical model for predicting vortex shedding noise from axial flow fan blade wakes. Wang [5] further developed a noise model and optimized the boundary layer development algorithm. Wang [6] presented an analytical model for plenum fan noise based on centrifugal impeller characteristics. Khelladi [7] showed that in the far field, a monopolar source is equivalent to a dipolar source induced by a uniform load distribution across the entire moving surface. Sasaki [8] proposed a prediction theory for broadband noise from multiblade fans based on Karman vortex street characteristics. Capece [9] investigated unsteady flow phenomena causing differences in instantaneous rotor blade wake data. These studies consistently agree that vortex shedding noise represents the primary noise source for small fans.

In Lee’ s 1993 analytical model, an idealized vortex street was considered for mathematical convenience. However, agreement between analytical results and experimental data requires improvement due to simplifications regarding the Karman vortex street in the blade wake. The present study proposes a modified model based on Lee’ s prediction model. Boundary layer theory is employed to analyze and calculate boundary layer development on both pressure and suction sides of blades. By considering boundary layer separation effects on noise source location, the predicted overall sound pressure level shows favorable agreement with experimental data from an axial fan. For A-weighted sound pressure level (LA) calculations, incorporating static pressure effects on radiated energy yields broadband noise predictions that compare favorably with experimental data from a multiblade centrifugal fan.

Model and Method

Lee developed an analytical model for predicting vortex shedding noise generated from axial flow fan blade wakes [4]. According to stability analysis of the Karman vortex street, the distance between vortex rows can be estimated by

[4]:

$$b = d + \delta_{ss} + \delta_{sp} \quad (1)$$

where b is the distance between vortex rows, d is the trailing edge thickness, and δ_{ss} and δ_{sp} are the boundary layer thicknesses on suction and pressure surfaces, respectively.

The vortex strength can be expressed as [4]:

$$K = \pi V(\theta_{ss} + \theta_{sp}) \tanh\left(\frac{\pi s}{a}\right) \quad (2)$$

where K is the strength of each individual vortex, V represents the main flow velocity, and θ_{ss} and θ_{sp} are the momentum thicknesses of boundary layers on suction and pressure surfaces, respectively [4].

The entire vortex street has an advance velocity $V_i = (K/2a)\tanh(\pi s)$, and the vortex shedding frequency is given by [4]:

$$f = \frac{V_i}{2a} \quad (3)$$

The Strouhal number based on trailing edge thickness is written as [4]:

$$St = \frac{fd}{V} \quad (4)$$

[Figure 1: see original paper] shows the idealized vortex street model, where the origin o is set at the airfoil center and the noise source is considered at the blade trailing edge under the assumption that vortex shedding occurs there.

However, since the boundary layer flow on the blade suction side experiences an adverse pressure gradient at an attack angle, it easily separates from the blade surface [Figure 2: see original paper]. When boundary layer separation occurs, vortices generate immediately from the separation point. At high Reynolds numbers, vortex shedding can occur upstream at the separation point rather than at the trailing edge.

[Figure 3: see original paper] illustrates flow around an airfoil from a cascade studied at an attack angle β of 17.5 degrees [10].

In Lee' s model [4], only dynamic pressure is involved in radiated energy calculations, neglecting static pressure effects. To incorporate static pressure influence, this study proposes a pseudo-velocity V^* through dimensional analysis:

$$V^* = \sqrt{V^2 + \frac{2P_{st}}{\rho}} \quad (5)$$

which corrects the main flow velocity.

Static pressure influence on overall sound pressure level is assumed to exist only in the vortex flow domain. The critical static pressure value for calculating pseudo-velocity V^* is the minimum value in the vortex flow domain. Conversely, static pressure influence is assumed negligible in the mainstream domain. Thus, the critical static pressure value also serves as the maximum static pressure in the mainstream domain:

$$P_r = \min(P_{st,\text{vortex}}) = \max(P_{st,\text{mainstream}}) \quad (6)$$

where P_r is the critical pressure.

The corrected velocity is defined by:

$$V_{corr} = \sqrt{V^2 + \frac{2(P_{st} - P_r)}{\rho}} \quad (7)$$

The total radiated energy can be expressed as:

$$E \propto \rho V_{corr}^3 \quad (8)$$

The relationship between sound power and sound pressure p is [4]:

$$p^2 \propto \frac{W}{\rho a_0} \quad (9)$$

where l is the distance from the test point to the noise source.

Considering boundary layer separation, the noise source location may shift from the trailing edge to the separation point. Accounting for this phenomenon, an improved noise calculation equation is proposed:

$$L_p = 10 \log_{10} \left[\frac{(x_s + \xi_n)^2}{x_{te}^2} \right] + L_{p,0} \quad (10)$$

In Eq. (10), the first term x_s represents the distance from origin o (blade center) to separation point S, and the second term represents the distance between a shedding vortex and separation point S.

The sound power radiated from the incoming flow field can be expressed as [4]:

$$W = \sum_{n=1}^{\infty} \frac{\rho a_0^3 b_x b_y}{n^2} \quad (11)$$

where a_0 is the speed of sound, constants b_x and b_y are 0.19 and 0.71, respectively [11], and taking the first 50 terms from the series meets actual computing requirements [6].

Sasaki et al. conducted experiments on a multiblade centrifugal fan [8] and compared results with predictions from Lee' s classic model [4]. The agreement between theory and experiment was very poor, indicating the need for improvement in Lee' s theory for broadband noise prediction.

Considering static pressure effects in the vortex flow domain, the equation for calculating broadband noise frequency generated by vortex shedding is proposed. The vortex shedding frequency can be written as [8]:

$$f = \frac{St \cdot V_r}{d} \quad (12)$$

where St is the Strouhal number and V_r is the relative velocity at impeller outlet.

Total pressure represents the total mechanical energy of the fluid. The total pressure level at impeller outlet can characterize the noise level. Total pressure consists of static pressure and dynamic pressure:

$$P_t = P_{st} + \frac{1}{2}\rho V_a^2 \quad (13)$$

where P_t is total pressure, P_{st} is static pressure, and V_a is absolute velocity at impeller outlet.

In the flow model [8], the flow is divided into two domains: a vortex flow domain at the bellmouth side and a mainstream zone where outflow is close to the hub side [8]. Experimental results in [8] show that static pressure in the vortex flow domain is clearly larger than in the main flow domain.

Results and Discussions

Fukano et al. conducted experiments on an airfoil-shaped axial fan [10] and compared predicted overall sound pressure with Lee' s model and experimental data. The results show that Lee' s predicted overall sound pressure is lower than experimental values [10], as shown in [Figure 4: see original paper]. This discrepancy may be due to boundary layer separation. The prediction result using the improved model, which considers boundary layer separation effects on the suction surface, is also plotted in [Figure 4: see original paper]. The modified model shows better agreement with experiments than Lee' s original model, demonstrating that considering boundary layer separation enables more accurate sound pressure level prediction.

Sasaki et al. performed experiments on a multiblade centrifugal fan [8] and simulated the flow field using commercial software Fluent. Simulation results for fan efficiency and total pressure compare favorably with experimental data, as shown in [Figure 5: see original paper] and [Figure 6: see original paper]. In [Figure 5: see original paper], the ordinate represents efficiency η and the abscissa represents flow rate coefficient ϕ. In [Figure 6: see original paper], the ordinate represents total pressure P_t and the abscissa represents flow rate coefficient ϕ. The simulated results agree well with experimental data.

[Figure 7: see original paper] shows the distribution of relative and absolute velocities versus impeller width at the blade trailing edge (where $z = 0$ corresponds to the bellmouth side and $z = 1$ corresponds to the hub side). As mentioned, the flow is divided into two domains: main domain (hub side) and vortex domain (shroud side). Velocities in the main flow zone are higher than those in the vortex zone, consistent with experimental data [8].

If vortex shedding noise from the fan blade wake is the main noise source, the vortex shedding frequency also represents the broadband noise frequency. Thus, relative velocity is used to calculate broadband noise frequency in Eq. (12). In subsequent calculations, absolute velocity is used to calculate radiated energy in Eq. (8) because radiated energy is closely related to absolute velocity.

[Figure 8: see original paper] shows the static pressure distribution at the impeller outlet along the blade trailing edge. Static pressure in the vortex flow domain is clearly higher than in the main flow domain. In actual radiated energy calculations, only static pressure effects in the vortex zone are considered, while static pressure effects in the mainstream zone are neglected.

In [Figure 8: see original paper], static pressure is relatively high when $0 < z < 0.4$. Considering the velocity distribution along the blade trailing edge, the region $0 < z < 0.4$ is regarded as vortex flow, and the region $0.4 < z < 1$ is considered the main flow region. This partition is consistent with literature [8]. Thus, static pressure at $z = 0.4$ is taken as the critical pressure P_r in Eq. (6) for detailed radiated energy calculation.

[Figure 9: see original paper] compares broadband noise predictions from the improved model with experimental data from [8]. Results predicted by Lee’ s original model [4] are also included. The fan experiment was performed at 1400 r/min [8], generating discrete noise at the blade passing frequency of approximately 2333 Hz. Since high-frequency noise is mainly generated by harmonics, this model focuses on broadband noise from 100 to 2000 Hz.

Lee’ s model calculates broadband noise frequency that deviates from the actual broadband noise region. The improved model’ s predicted A-weighted sound pressure level (LA) is plotted in [Figure 9: see original paper], with frequency calculated using the Strouhal number from Eq. (12). In the present calculation, $St = 0.2$ is employed according to reference [8]. The prediction without static pressure effects is labeled “present V,” while the prediction including static pressure effects is labeled “present V*” in [Figure 9: see original paper]. Both calculation results agree well with experimental data in the 100 Hz–2000 Hz range. Incorporating static pressure effects yields A-weighted sound pressure level predictions that favorably accord with experiments.

Conclusions

Lee' s classic model is widely used by researchers for fan noise prediction, but it compares poorly with experiments in some cases. The present study improves this model by considering additional factors.

First, boundary layer theory is used to analyze and calculate boundary layer development on both pressure and suction sides of blades. By considering boundary layer separation effects on noise source location, a simple improved model is constructed. The predicted overall sound pressure level compares favorably with experimental data from an axial fan.

Second, in A-weighted sound pressure level (LA) calculations, an improved model is proposed that incorporates static pressure effects on radiated energy. Broadband noise predictions from the modified model compare favorably with experimental data from a multiblade centrifugal fan.

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Note: Figure translations are in progress. See original paper for figures.