Effect of the Uneven Circumferential Blade Space on the Performance of Small Axial Flow Fan Postprint

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Abstract

A rack cooling system based on a large scale flat plate pulsating heat pipe is proposed. The heat generated from IT equipment in a closed rack is transferred by the rear door pulsating heat pipe to the chilled air passage and is avoided to release into the room. The influence of the start-up performance of the heat pipe, the load of the rack and the load dissipation to the temperature and the velocity distribution in the rack are discussed. It is found that the temperature would be lower and the temperature distribution would be more uniform in the rack when the pulsating heat pipe is in operation. Also, the effect of rack electricity load on temperature distribution is analyzed. It is indicated that higher velocity of chilled air will improve heat transfer of the rack.

Full Text

Preamble

Effect of the Uneven Circumferential Blade Space on the Performance of Small Axial Flow Fan

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This work investigates the effects of uneven circumferential blade spacing on the static characteristics and aerodynamic noise of a small axial flow fan. Blade

angle modulation is employed to design a series of unequally spaced fans with different maximum modulation angular displacements. Steady flow is simulated using Reynolds-averaged Navier-Stokes equations coupled with an RNG k-epsilon turbulence model, while unsteady flow is computed via large eddy simulation. Based on theoretical analysis, a fan with a maximum modulation angular displacement of 6° is identified as the optimal unequally spaced configuration. Static characteristic experiments are conducted in a standard wind tunnel, and aerodynamic noise measurements are performed in a semi-anechoic room. Performance comparisons between the optimal unequally spaced fan and the prototype fan reveal reasonable agreement between simulation and experimental data. The results demonstrate that the discrete noise of the optimal unequally spaced fan is lower than that of the prototype fan at the near-field monitoring point, which can be attributed to the more regular total pressure fluctuation characteristics of the optimal design.

Keywords: small axial flow fan, uneven circumferential blade space, static characteristics, discrete noise, broadband noise

Introduction

The aerodynamic noise of axial flow fans typically comprises discrete noise and broadband noise components. Discrete noise arises from the periodic interaction between rotating blades and air particles, producing a spectrum characterized by a series of discrete peaks. The discrete frequency depends on the blade count, rotational speed, and tonal characteristics, and may be more annoying than broadband noise in certain applications. Consequently, reducing discrete noise can effectively improve the noise environment of axial flow fans.

Uneven circumferential blade distribution as a passive noise reduction method was first proposed by Mellin et al. and has since been continuously developed and applied. Sun derived the acoustic radiation formula for unequally spaced fans based on Blade Loading Harmonics (BLH) theory presented by Wright, further elucidating the noise reduction mechanisms. Sun also concluded that large variations in blade spacing do not significantly affect the aerodynamic performance of axial flow fans. Cui applied this method to mine ventilation fans and compared the noise reduction effectiveness of odd versus even blade counts in tunnel environments, finding that unequally spaced fans with odd blade numbers achieved substantially greater noise reduction. Xu' s research demonstrated that unequally spaced fans not only eliminate the regular pressure destabilization caused by equally spaced blades but also attenuate resonance noise generated by blade interactions with fixed obstacles in ventilation ducts, thereby reducing discrete noise.

In the present work, uneven circumferential blade spacing is applied to a small axial flow fan for noise reduction. The application proves successful, and the research findings benefit parameter optimization and noise prediction for unequally spaced small axial flow fans.

Research Method

Design of Unequally Spaced Fan

Current design methods for unequally spaced fans can be categorized into four approaches: look-up table, conversion to two-bladed propellers, blade angle modulation, and trial calculation. This study adopts blade angle modulation to design a series of unequally spaced fans with varying maximum modulation angular displacements. This method was first proposed by Ewald et al. and subsequently improved by Duncan and Dawson. Fiagbedzi conducted a thorough mathematical analysis, while Roger provided a more physical interpretation. The method varies blade spacing according to:

$$\theta_i' = \theta_i + \Delta\theta \sin\left(\frac{2\pi n i}{Z}\right)$$

where θ_i is the angular position of the ith blade in an equally spaced fan, θ_i' is the angular position in the unequally spaced configuration, $\Delta\theta$ represents the maximum modulation angular displacement, and n denotes the modulation cycle index. Natural blade balance is achieved when $n \geq 2$.

As illustrated in [Figure 1: see original paper], a coordinate system is established to determine each blade' s angular position, with the fan center as the origin and the angular position of the zth blade referenced as 0°. Equation (1) indicates that larger $\Delta\theta$ values produce more uneven circumferential blade distributions, making performance differences between equally and unequally spaced fans more pronounced. Therefore, selecting a reasonable $\Delta\theta$ requires careful consideration of both static characteristics and aerodynamic noise performance.

Simulation for Static Characteristics

Steady flow simulation employs Reynolds-averaged Navier-Stokes equations coupled with an RNG k-epsilon turbulence model. Standard wall functions handle near-wall regions, while the SIMPLE algorithm manages pressure-velocity coupling. Second-order upwind differencing discretizes the governing equations. The computational domain ([Figure 2: see original paper]) comprises inlet, outlet, rotating fluid, and casing regions. The inlet region is a hemisphere with radius $r = 85$ mm, while the outlet region is a cylinder with radius $R = 170$ mm extending 500 mm to ensure fully developed flow. Mass flow rate is controlled at the inlet, and atmospheric pressure is specified at the outlet.

Simulation for Aerodynamic Noise

Large eddy simulation (LES) and the Ffowcs Williams-Hawkings (FW-H) noise model predict unsteady flow and aerodynamic noise. The steady simulation solution at rated conditions ($Q = 0.01$ kg/s) serves as the LES initial condition. The finite volume method discretizes the filtered Navier-Stokes equations, with the PISO algorithm handling pressure-velocity coupling and second-order central differencing applied to the momentum equation. Dynamic grid methods manage interface interactions between rotating and stationary regions.

Selection of Optimal Unequally Spaced Fan

The prototype fan is shown in [Figure 3: see original paper] with main parameters listed in . Blade angular positions for the prototype fan are given in . Based on this prototype, a series of unequally spaced fans with different $\Delta\theta$ values were designed using blade angle modulation.

[Figure 4: see original paper] illustrates static pressure variation with $\Delta\theta$ at rated conditions, with the solid black line representing least-squares fitting. Static pressure initially increases then decreases as $\Delta\theta$ increases. A 4% static pressure drop relative to the prototype fan is considered acceptable, establishing $\Delta\theta \leq 22°$ as the allowable range.

[Figure 5: see original paper] shows efficiency variation with $\Delta\theta$, also fitted using least-squares methods. Efficiency generally decreases with increasing $\Delta\theta$. A 4% efficiency drop is deemed acceptable, yielding $\Delta\theta \leq 8°$ as the permissible range.

Aerodynamic noise was evaluated using near-field (1 mm) and far-field (1 m) monitoring points on the positive rotational axis. [Figure 6: see original paper] presents SPL variation with $\Delta\theta$ at both locations. While all unequally spaced fans reduce near-field SPL compared to the prototype, far-field SPL shows minimal variation. Notably, near-field SPL does not decrease monotonically with $\Delta\theta$ but fluctuates within a certain range.

Considering both static characteristics and aerodynamic noise, the fan with $\Delta\theta = 6°$ is selected as the optimal unequally spaced configuration. Using Equation (1), the design equation becomes:

$$\theta_i' = \theta_i + 6° \sin\left(\frac{4\pi i}{Z}\right)$$

Blade angular positions for the optimal design are listed in . Center of mass verification confirms the optimal fan remains balanced on the rotational axis, eliminating eccentricity-induced vibration. The blade spacing variations show the maximum circumferential spacing change occurs between blades 2 and 3, with a 15.9% variation rate—consistent with Wright' s recommendation of maintaining variations below 20%.

Comparison of Static Characteristics

Simulation Results and Analysis

Static characteristics represent fundamental performance indicators for small axial flow fans. [Figure 7: see original paper] compares static characteristic curves between the optimal unequally spaced fan and prototype fan, including static pressure ([FIGURE:7(a)]) and efficiency ([FIGURE:7(b)]) curves. Both fans exhibit similar P-Q curve trends: static pressure decreases sharply, then increases slowly, and finally decreases rapidly with increasing flow rate. Static pressure values remain close across different flow rates, with stable operation ranging from 0.005 kg/s to 0.012 kg/s. Beyond this range, static pressure changes dramatically.

The -Q curves ([FIGURE:7(b)]) also show similar trends, with efficiency increasing gradually before decreasing rapidly. Maximum efficiency exceeds 22% for both designs, with values remaining close across a range of flow rates. The prototype fan achieves maximum efficiency at $Q = 0.011$ kg/s, while the optimal design reaches it at $Q = 0.012$ kg/s.

[Figure 8: see original paper] presents static pressure contours on impeller surfaces at rated conditions. Comparing suction sides reveals similar pressure values and distributions, with negative pressure zones of -40 Pa (mid-blade) and -50 Pa (near leading edge) appearing on most blades. Pressure side comparisons show similar static pressure distributions, with only a few blades exhibiting zones above 20 Pa. The minimal differences in static pressure fields explain the similar P-Q curves.

Wind Tunnel Experiment

Wind tunnel experiments validate simulation reliability. [Figure 9: see original paper] shows the experimental setup, with test fans illustrated in [Figure 10: see original paper]. [Figure 11: see original paper] compares prototype fan P-Q curves between simulation and experiment, showing good agreement, particularly in the stable operation range of 0.005-0.011 kg/s, thus confirming simulation reliability.

[Figure 12: see original paper] presents experimental P-Q curves for both fans, revealing nearly identical trends. Static pressure decreases sharply, increases slowly, then decreases rapidly with flow rate. Minimal differences exist between the two designs in the stable operation range (0.005-0.011 kg/s). Combined simulation and experimental results demonstrate that the optimal unequally spaced fan produces only minor changes to the prototype's static characteristics.

Comparison of Aerodynamic Noise

Simulation Results

[Figure 13: see original paper] shows SPL variation over time at both monitoring points. Before 6 rotational periods (T), SPL at both locations is unstable, exhibiting an overall downward trend. After 6T, SPL stabilizes, indicating the internal flow field reaches a relatively steady state. At the near-field point, the optimal design achieves over 2 dB noise reduction compared to the prototype, while far-field noise reduction is negligible. Since discrete noise attenuates faster than broadband noise, near-field noise is dominated by discrete components while far-field noise is broadband-dominated. This suggests uneven circumferential blade spacing significantly reduces discrete noise but has minimal effect on broadband noise.

[Figure 14: see original paper] presents SPL distributions along the downstream axial direction. The optimal design shows lower SPL at all axial positions, with the maximum difference of 2.3 dB occurring at the near-field monitoring point. The SPL difference decreases with increasing axial distance, stabilizing at 0.4 dB beyond 600 mm. The rapid attenuation of discrete noise relative to broadband noise indicates that discrete components are completely attenuated beyond 600 mm, where broadband noise dominates. This further confirms that uneven blade spacing reduces discrete noise effectively while having little impact on broadband noise.

[Figure 15: see original paper] shows Power Spectral Density (PSD) distributions at the near-field monitoring point. The prototype fan exhibits a 200 Hz peak of 0.11 W/Hz, which the optimal design splits into two lower peaks of 0.05 W/Hz and 0.025 W/Hz. This indicates the optimal design does not reduce total acoustic energy at 200 Hz but redistributes it across the frequency domain. PSD values for both fans approach zero above 600 Hz, confirming that noise reduction concentrates at 200 Hz and primarily affects discrete noise.

[Figure 16: see original paper] presents 1/3 octave band SPL at the far-field monitoring point. The highest SPL band (200–250 Hz) reaches 40 dB for the prototype. The optimal design shifts this peak to a higher frequency range (300–360 Hz) without changing its magnitude. SPL values remain similar across other frequency bands, demonstrating minimal broadband noise reduction at far-field locations.

Analysis of Flow Field Characteristics

Uneven circumferential blade spacing significantly reduces discrete noise while having minimal effect on broadband noise. This section explains these phenomena through flow field characteristics.

[Figure 17: see original paper] compares total pressure fluctuations at the near-field monitoring point between 6T and 9T, when both fans reach relatively stable states. Neither fan exhibits obvious periodic variation, with approximately four peaks appearing per rotational period—consistent with the 200 Hz PSD peak observed earlier. Compared to the prototype, the optimal design shows more regular, equal-amplitude oscillations with smaller extreme differences.

[Figure 18: see original paper] shows total pressure fluctuations 500 mm from the impeller hub on the rotational axis. At this location, both fans exhibit irregular fluctuations with very small magnitudes, indicating noise is not influenced by total pressure fluctuations at far-field positions.

[Figure 20: see original paper] presents vorticity contours at section Z = 6 mm (near the trailing edge, [Figure 19: see original paper]) at different moments (7T, 7T+3T/5, and 8T). Comparisons reveal nearly identical vorticity magnitude and distribution near the blade trailing edge for both fans at all moments. Maximum vorticity exceeds 2.4×10^4 s ^{1}, located between 1/3 and 1/2 span on the suction side, while minimum vorticity between adjacent blades approaches 2.0×10^3 s ^{1}. The similar vorticity variations during rotation explain why broadband noise differences between the two fans are minimal at far-field monitoring points.

Noise Experiment

Noise experiments validate simulation results in a semi-anechoic room (5 m × 4 m × 3.5 m) with 21.5 dB background noise. The measurement setup is shown in [Figure 21: see original paper].

[Figure 22: see original paper] presents measured SPL along the downstream axial direction. The maximum SPL difference (exceeding 3 dB) occurs at the near-field measuring point (1 mm from the impeller hub). SPL decreases rapidly with increasing axial distance before stabilizing, with both fans showing nearly identical values at the far-field point (1 m from the hub). Experimental and simulation differences arise primarily from the absence of casing in experimental fans, though the data remain comparable. These results further confirm that uneven circumferential blade spacing significantly reduces discrete noise while having minimal effect on broadband noise.

Conclusions

This paper investigates the effects of uneven circumferential blade spacing on small axial flow fan performance, examining static characteristics and aerodynamic noise. Based on the prototype fan, blade angle modulation designs a series of unequally spaced fans with varying $\Delta\theta$. The conclusions are summarized as follows:

(1) Simulation results indicate that static pressure initially increases then decreases with increasing $\Delta\theta$, while efficiency generally decreases. The unequally spaced fan with $\Delta\theta = 6°$ provides optimal noise performance while meeting static characteristic requirements.

(2) Both simulation and experiment demonstrate that the optimal unequally spaced fan produces lower discrete noise than the prototype at near-field monitoring points, while broadband noise remains essentially unchanged at far-field locations.

(3) At near-field points, discrete noise dominates. The optimal design exhibits more regular total pressure fluctuations compared to the prototype. However, vorticity magnitude and distribution near blade trailing edges are nearly identical for both fans at different moments. Therefore, total pressure fluctuations explain the reduced discrete noise of the optimal design.

(4) At far-field points, broadband noise dominates, and vorticity-induced noise significantly exceeds that from total pressure fluctuations. Since vorticity variations are similar for both fans during rotation, their broadband noise levels are comparable.

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