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The Influence of Guide Vane Blade Distribution on the Performance of Mine Axial Flow Fan

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Abstract: The noise generated by fan blades is the main noise source of axial flow fans used in mines. Because of its excessive noise, the need for noise reduction is increasingly urgent. This paper takes a certain type of mine axial fan as the research object, uses Lighthill acoustic analogy theory, FW-H acoustic wave equation and Fluent numerical simulation method to compare the performance of the fan under uniform and non-uniform distribution of the guide vane, and focuses on the influence of the guide vane layout on the aerodynamic performance, flow field characteristics and noise characteristics of the fan. The results show that, compared with the uniform distribution of the guide vane, the non-uniform distribution can reduce the sound pressure level of discrete noise in the fundamental and harmonic frequencies to different degrees, and the sound energy of the fundamental and partial harmonic frequencies can be distributed in a wider frequency band, but the total sound pressure level of the fan is basically unchanged. Selecting A reasonable distribution mode can ensure the aerodynamic performance and flow field characteristics of the fan, and play a significant noise reduction effect on the low and medium frequency noise, in which the modulation amplitude A=0.2 has the best effect, and the modulation amplitude is reduced by 13.6dB at 550Hz.

Keywords: guide vane non-uniform distribution; noise characteristics; flow field analysis; aerodynamic performance; sound pressure level

1. Introduction

As a widely used fluid machinery, axial fan plays an important role in engineering practice, but it also causes a large noise pollution, among which aerodynamic noise is the main sound source of axial fan. Aerodynamic noise can be divided into two categories according to its generation mechanism. One is discrete noise, which is generated by the pressure pulsation caused by the periodic impact of the blade on the gas particle. The other is broadband noise, which is due to the continuous change of airflow velocity from blade root to blade tip along the radius direction, and it is also affected by

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the pulsating pressure on the blade, which is a kind of broadband noise without obvious main frequency.

The aerodynamic noise of rotating blades has been studied extensively at home and abroad. According to the aeroacoustic theory of Lighthill [1], Wright [2] and Lowson [3] . derived the sound radiation formula of the rotor with the same pitch blade from different angles. In China, Sun Xiaofeng [4] derived the sound radiation formula of fans with unequal pitch based on Wright's BLH theory, successfully predicted and proved by experiment that blades with different pitch had modulation effect on the fan's passing frequency and harmonics than blades with equal pitch. Ma Jianfeng [5] et al. carried out the impeller blade unequal distance design according to the blade group self-balancing distribution scheme, and conducted numerical simulation and experimental research, which proved that the blade arrangement with unequal distance could change the phase of sound pressure, but the amplitude of sound pressure did not change significantly. Wang Gangfeng [6] et al. compared the fan performance under uniform distribution, irregular non-uniform distribution and priority number system arrangement, thus proving that the non-uniform distribution structure can weaken the peak value and frequency shift of discrete noise, but when the common ratio is too large, the flow field will be easily disturbed. Selecting the priority number system arrangement with reasonable common ratio can ensure the aerodynamic performance and flow field characteristics of the fan and play an obvious noise reduction effect.

All the above studies show that the use of non-equidistant blades has a positive effect on the noise suppression of centrifugal fans or fans. At present, the numerical studies on non-equal pitch blades are mainly focused on centrifugal fans and cross-flow fans, and there is little research on axial fans. Therefore, this paper focuses on the study of the changes of aerodynamic performance, flow field characteristics and noise characteristics of axial flow fans with non-uniform guide blades, and the results can provide theoretical basis for the optimal design of axial flow fans.

2. Model establishment

The impeller diameter of the fan is 800mm, the hub diameter is 480mm, the blade tip clearance is 2mm, the speed is 2950 r/min, the rotor blade installation angle is 38°, the number of rotor blades is 7, and the number of rear guide blades is 11, the overall structure is shown in Figure 1.

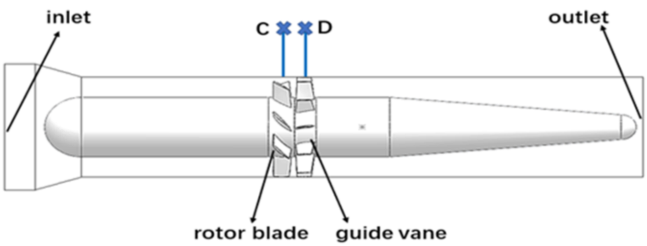

Figure 1. Mine axial fan structure schematic

Guide vane distribution adopts the sinusoidal modulation method proposed by Ewald[7] et al. The method is based on the design formula: $\boldsymbol{\varphi' = \varphi + A\sin n\varphi}$ **(1)** Where A is the modulation amplitude; n is the number of cycles of the modulated quantity, and when n > 2, the fan rotates to maintain dynamic balance; φ is the uniform blade spacing angle; φ' is the non- equidistant blade spacing angle. In this paper, four new distribution modes are proposed, corresponding to A of 0, 0.1, 0.15 and 0.2 respectively. The greater the value of A, the greater the degree of non-uniformity of the blade, where A=0 is uniform blade. Four design schemes are shown in Table 1.

Table 1. Four kinds of non-uniform distribution guide vane adjacent blade angle

Blade number	1	2	3	4	5	6	7	8	9	10	11
1(A=0)	32.73	32.73	32.73	32.73	32.73	32.73	32.73	32.73	32.73	32.73	32.73
2(A=0.10)	37.93	31.84	26.77	28.66	35.29	38.92	35.31	28.66	26.77	31.82	38.03
3(A=0.15)	40.56	31.29	23.79	26.6	36.59	42.03	36.62	26.61	23.79	31.36	40.66
4(A=0.20)	43.20	30.94	20.8	24.52	37.92	45.13	37.95	24.55	20.79	30.91	43.2

3. Numerical simulation

Tetrahedral mesh is used for the moving and guiding impeller, and structured mesh is used for the flow collector and diffuser. The grid independence of the fan model with uniform guide vanes (A = 0) was verified by seven grid division schemes. Figure 2. shows the change of the total pressure efficiency of the fan with the number of grids. When the total number of grids reaches 3.5 million, the total pressure efficiency does not increase with the increase of the number of grids, but keeps fluctuating in a small range. Therefore, the number of grids used in the whole computing domain is about 3.5 million.

Firstly, Fluent software was used for the Steady calculation of three-dimensional full flow field. The solver was steady and the turbulence model was RNG k-ε. The wall surface is non-slip boundary, and the pressure-velocity coupling adopts SIMPLEC algorithm. The inlet boundary is set to the pressure inlet, and the outlet boundary is set to the pressure outlet. Multiple reference frame model is used to solve the interference problem between static and static blades. Convergence is considered when k, ε and velocity residuals in each direction are less than 10^{-4}. After the convergence of steady calculation, Proudman is introduced to improve the broadband noise model and the power distribution of fan aerodynamic noise is obtained by using the broadband noise source model. A combination of large eddy simulation and FW-H model was used for noise simulation. The steady state calculation was used as the initial field, the pressure correction was adopted by PISO algorithm, the turbulent kinetic energy was set as PRESTO format, and the sliding grid was used to calculate the dynamic and static interfaces and moving blade regions. The time domain characteristics of pressure pulsation are obtained by setting monitoring points, and the time domain characteristics of pressure are transformed into the spectral characteristics of sound pressure by fast Fourier transform.

4. Analysis of Results

4.1 Aerodynamic performance analysis

The aerodynamic performance of the fan can be studied by analyzing the performance parameters such as wind pressure, flow rate and efficiency. In order to show the aerodynamic performance of the fan more directly, the full pressure flow curve (P-Q) and the full pressure efficiency flow curve (η-Q) can be drawn. By changing the flow rate of the inlet, the performance parameter value of the fan under the flow rate can be obtained, and the corresponding curve can be drawn Figure 3. shows the comparison of the aerodynamic performance of the fan under different schemes.

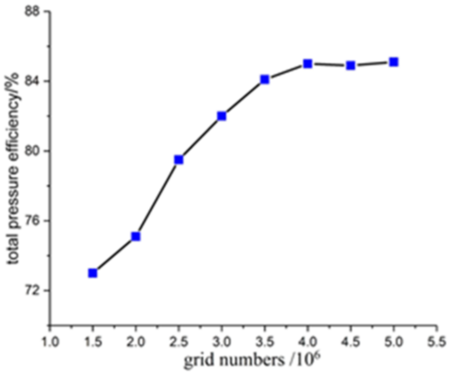

Figure 2. Grid independence verification

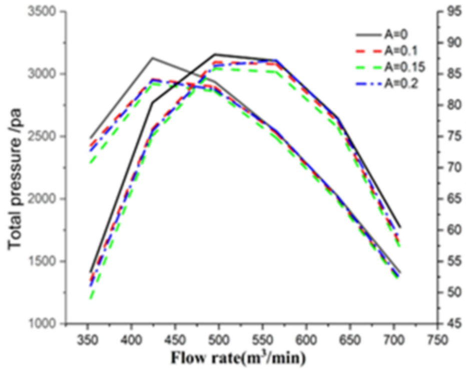

Figure 3. Comparison of fan aerodynamic performance

According to the analysis, the total pressure and efficiency of the three schemes with non-uniform guide vane distribution are reduced to different degrees compared with those with uniform guide vane distribution, indicating that the aerodynamic performance of the three schemes is worse than that of the fans with uniform guide vane distribution. However, in general, the range of changes in total pressure and efficiency is limited, and the difference is only between 2%-5%. Especially in the working area, the efficiency changes very little, so to a large extent, the choice of blade distribution mode will not significantly reduce the aerodynamic performance of the fan[4].

4.2 Flow field characteristic analysis

The flow field characteristics of the fan can be expressed by the pressure distribution, which includes static pressure, dynamic pressure and total pressure. The cloud diagram of static pressure distribution is shown in Figure 4.

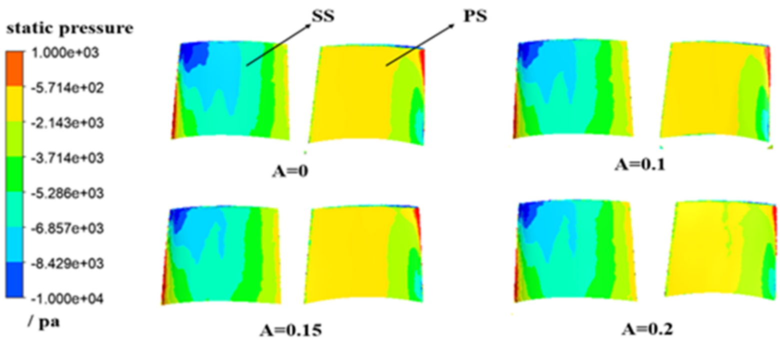

Figure 4. Comparison of static pressure cloud image of 4 kinds of fan blades

The analysis of Figure 4 shows that the static pressure of the pressure surface (PS) is higher than that of the suction surface (SS). In the suction surface, the four kinds of fan blades have the same distribution characteristics. The static pressure value appears negative at the leading edge of the blade tip and the middle area of the blade, while the static pressure value gradually increases at the trailing edge of the blade, which is related to the convex structure of the blade. In the pressure surface, the four kinds of fan blades all have large static pressure values at the leading edge of the blade tip, while the static pressure values at the center of the blade are relatively low and evenly distributed. This is because the center of the blade is concave to the inside. When the air passes through the leading edge of the blade in the process of blade rotation, the static pressure in the center area decreases due to the concave structure. Compared with the four schemes, it is found that the static pressure value of the three non-uniformly distributed blades decreases, and the modulation amplitude A=0.2 is the most obvious.

4.3 Noise characteristic analysis

In order to better predict the noise distribution of mine axial fan, Proudman was introduced to improve the broadband noise model after the steady-state calculation convergence, and the sound pressure cloud image of the fan was obtained, as shown in Figure 5. As can be seen from Figure 5, the sound pressure level in the impeller region of the fan is large, and the maximum value is located in the tip region of the two-stage impeller blade. Therefore, the two-stage impeller region of the fan should be taken as the key area of noise research.

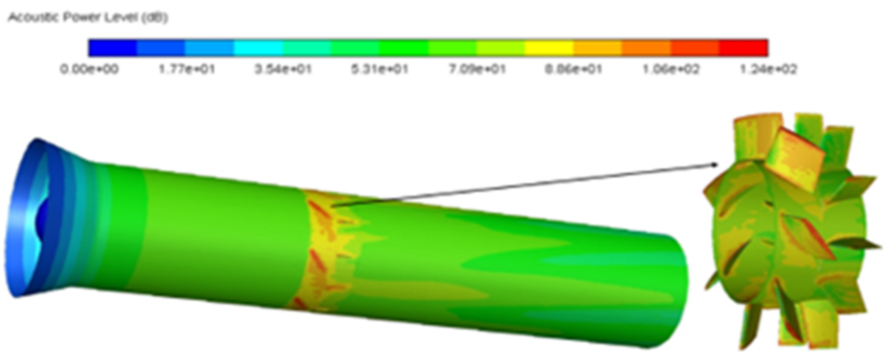

Figure 5. Cloud image of sound pressure level distribution of aerodynamic noise

The combination of LES and FW-H is used to analyze the aerodynamic noise in fluent software. By solving the transient flow field through LES, the pressure pulsation on the blade surface can be obtained, and then the FW-H equation can be solved to obtain the time domain signal at the monitoring point, and the sound pressure level spectrum curve can be obtained by fast Fourier transform. Pneumatic noise monitoring points set outside the fan are shown in Figure 1. A monitoring point is set in the axial middle position of the wind-driven vane and the rear guide vane and 0.5m away from the fan shell. Along the direction of air flow, the monitoring points for aerodynamic noise are named C and D respectively.

In the spectrum diagram, the discrete noise is represented as the noise corresponding to the discrete peak value, and its frequency value is mostly an integer multiple of the rotation frequency. The formula is : **f = nzi / 60 (2)** Where n is the speed of the impeller (r/min); z is the number of blades; i indicates the harmonic sequence number. When i = 1, the frequency is the fundamental frequency, when i = 2,3... The frequency is harmonic frequency. From the spectrum diagram of fan aerodynamic noise with uniform blade distribution (A=0), it can be seen that the first peak of fan aerodynamic noise appears near the base frequency of 350 Hz, while the second peak appears near 550Hz, which may be due to the passage frequency generated by the airflow passing through the rear guide vane. Subsequently, high order harmonic frequencies such as 680 Hz and 1060 Hz were found to peak. Figure 6. and Figure 7. are the spectral graphs of aerodynamic noise at monitoring points C and D under four schemes respectively. Table 2 shows the comparison of total sound pressure levels under uniform and non-uniform distribution of guide blades, and Table 3 shows the comparison of sound pressure levels at fundamental and harmonic frequencies under uniform and non-uniform distribution of guide blades. In summary, it can be concluded that the non-uniform distribution of guide blades can reduce the sound pressure level of the fan at the fundamental frequency and harmonic frequency to different degrees, but the total sound pressure level remains basically unchanged.

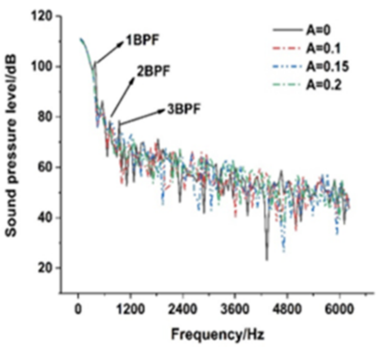

Figure 6. The spectrum comparison diagram of 4 schemes at C

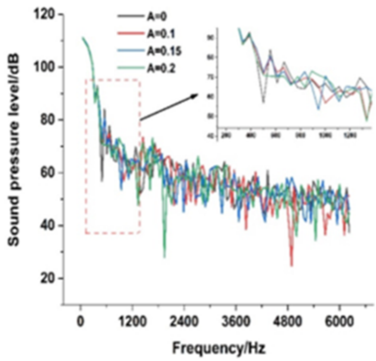

Figure 7. The spectrum comparison diagram of 4 schemes at D

Table 2. Comparison of total sound pressure levels of 4 schemes

	1 (A=0)	2 (A=0.1)	3 (A=0.15)	4 (A=0.2)
Sound pressure level/dB(A)	112.64	112.87	112.65	113.74

Table 3. Comparison of sound pressure levels at the fundamental frequency and harmonic frequency of the fan under 4 schemes

Frequency	345Hz	550 Hz	680 Hz	1060 Hz	1390 Hz
A=0	91.3	84.1	77.7	66.8	65.2
A=0.1	91.0	79.9	74.8	60.5	61.7
A=0.15	91.2	80.7	72.9	64.3	62.3
A=0.2	92.2	70.5	70.5	66.4	67.8

5. Conclusions

In this paper, the effects of four blade distribution modes on the aerodynamic performance, flow field characteristics and noise characteristics of the fan are studied by numerical simulation, and the following conclusions are reached:

(1) For the three types of non-uniformly distributed guide vane structures, their aerodynamic performance is lower than that of uniformly distributed guide vane structures, but the difference is only in the range of 2%-5%. Therefore, it can be inferred that the selection of blade distribution mode within a certain overstep will not worsen the aerodynamic performance of the fan.

(2) Flow field analysis shows that the non-uniformly distributed guide vane structure has obvious improvement on the static pressure distribution of the fan flow field.

(3) The noise spectrum analysis shows that the non-uniform distribution of guide blades can reduce the sound pressure level on the fundamental frequency and harmonic frequency to varying degrees, but the total sound pressure level is basically unchanged.

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