Research on Performance Optimization of Multi-Stage Centrifugal Fan

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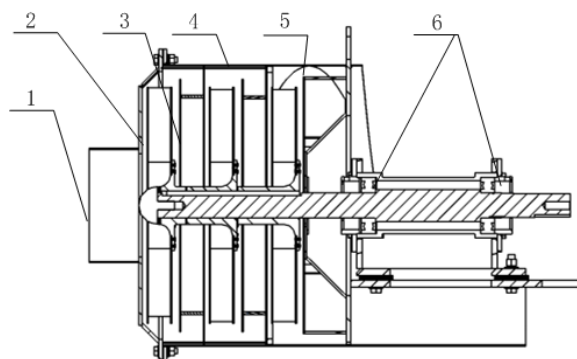
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Abstract. The optimization of fan performance under changed conditions has important significance for the practical application of the fan. In view of the low efficiency of the original fan, the stable working flow span of the flow pressure curve is narrow, and it is difficult to meet the performance requirement after the working point change. Without changing the radial and axial dimensions of the fan, the performance of the original fan is optimized so that it increases from the total pressure 16153pa to 17271pa when operating flow 1900m3/h, meets the optimized target of total pressure 17000pa, the performance increases by 6.9%, and makes the stable working range of its pressure curve widen.

1. Introduction

With the continuous expansion of fans in the metal mines, coal industry, steel industry and electric power industry, fan energy consumption continues to increase [1]. It is of great significance to study how to improve the performance of fans and reduce the power consumption of fans for energy conservation [2]. Multi-stage centrifugal fan shown in figure 1, with a small flow of large pressure, wide operating conditions, low noise, economical and practical, and other notable features [3].

1. Air inlet, 2. Rotary impeller, 3. Returner, 4. Volute, 5. Air outlet, 6. Bearing

Figure 1. Multi-stage centrifugal fan structure

The current research on rotating machinery design methods mainly focuses on single-stage application of centrifugal fans. For example, Xi G proposed the application of approximate model

method in turbomachinery aerodynamic optimization design [4]; Li J B based on this, an optimal design method for controlling the relative mean velocity of centrifugal impeller runners was proposed [5]; Zhao Y B et al. calculated the internal flow field of 2d numerical centrifugal fans [6]; Therefore, the research on performance optimization of multi-stage centrifugal fan under different working conditions is of practical significance. however, the research on multi-stage centrifugal fans is relatively rare.

2. The original fan structure and performance improvement analysis

Table 1. Statistics of original fan test results

Test point	Flow (m^3/h)	Total pressure (Pa)	Motor power (KW)	Efficiency
1	0.21	18599.04	6.37	0.02%
2	637.20	19288.10	8.79	44.38%
3	1268.65	19114.23	11.90	64.61%
4	1793.15	16971.75	14.68	64.64%
5	1925.62	15853.29	15.54	60.76%

The air inlet performance test was performed on the fan, and the test data was converted to the calculation of the standard conditions. The results are shown in Table 1. The analysis of the test results of the fan can obtain that the stable working range of the original fan pressure curve is narrow, and the operating pressure is 1900m^3/h, the full pressure is 16153pa, the efficiency is low, and the pressure value near the operating point changes significantly, which is not conducive to The actual work of the fan. Therefore, it is necessary to optimize the design of the existing fan according to the basic requirement that the pressure curve near the working point is steady and the work efficiency is high.

The radial and axial volume of the original fan is small, and the optimization space is narrow. According to the process requirements, the fan optimization cannot change the radial and axial dimensions of the fan.

3. Determination of optimization plan

In this design process, the rotation speed and the transmission ratio of the motor remain unchanged, and the radial and axial dimensions of the fan are unchanged. In the optimization process, the number of blades and inlet and outlet angles of the moving impeller and the returner are mainly considered.

Using the hydraulic calculation equation (1) to estimate the fan fluid loss.The pressure loss coefficient can refer to the single-stage centrifugal fan of the same speed, and then obtain the overall performance of the fan.

$$\Delta p_i = \xi_i \rho \frac{c_i^2}{2} \tag{1}$$

Due to the fact that the radial dimension of the fan has not changed,Therefore, when the fan operating conditions change,The main reason for the pressure drop of the fan is the impact loss caused by the fan under non-working conditions.The method to reduce the impact loss of the fan is to reduce the attack angle i when the air flow enters the blade from the impeller. the difference between the air flow angle $\beta_{1'}$ and the blade inlet angle β_1 is as small as possible.

The number of blades of the original fan rotating impeller and the returner was 9 and 6 respectively. In order to reduce the vibration, the number of the two groups of blades was changed to be a prime number, and the number of blades of the returner was initially changed to 7 pieces. According to equation (2), the average velocity of the air flow in each channel of the fan is obtained.

$$c_i = \frac{Q_i}{S_i K} \tag{2}$$

By the equation (2) estimating the blade inlet flow passage axis average speed is 25.61m / s. Since the circumferential velocity u_1 of the blade at the inlet is 48.96 m/s. According to equation (3), the airflow angle β_1 is 27.61°. When the air flows into the blade, since the blade thickness such that the flow area decreases, the speed increases, and the inlet velocity of the flow channel can be calculated as 26.84 m/s by equation (5). The speed is iterated to equation (3), and the airflow angle at the inlet of the blade can get again be 28.73°. The difference between the result and the geometric angle of the air inlet can be obtained. If the airflow angle of attack is small enough, the iteration can be stopped.

$$\beta_i^{'} = \arctan \frac{c_i^{'}}{u_i} \tag{3}$$

$$\tau_i = 1 - \frac{Z \times \delta_i}{\pi \times D_i \times \sin \beta_{i A}}$$
$$= 1 - \frac{9 \times 0.0015}{\pi \times 0.187 \times \sin 30^0} = 0.9540 \tag{4}$$

$$c_1^{'} = \frac{c_1}{\tau_1} = \frac{25.61}{0.9540} = 26.84 \tag{5}$$

According to the fan manual [8], multi-stage fan impellers have higher working efficiency when the blade outlet angle is 60°. In the process of this transformation, consider the actual requirements for the production of the sheet metal parts of the impeller, using a flat disk design, and the fan lacks a curve, the blade outlet is changed to 60° by experience. It can be concluded that the blade radius $R_k = 735.8$mm, the center radius of the blade $R_0 = 656.5$mm, the modified impeller shown in figure 2.

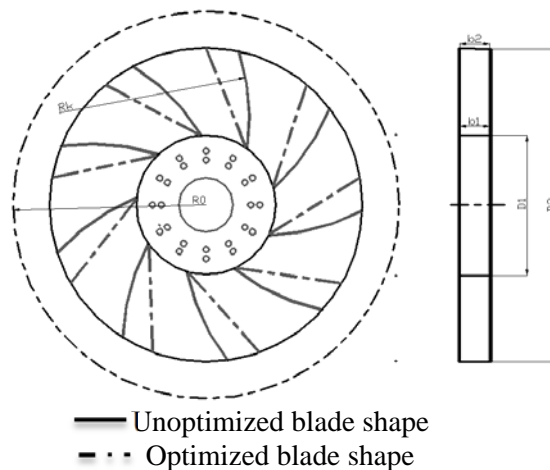Unoptimized blade shape

Optimized blade shape

Figure 2. Before and after the transformation of the rotary impeller

For the transformation of the returner, the number of blades was changed to 7. According to the principle of constant momentum and flow rate, the radial velocity of the impeller inlet is 12.24m/s and 4.15m/s, respectively. The inlet angle of the impeller blade is designed to be 17° according to the standard of high speed.

The angle β_4 of the outlet of the returner blade and the angle β_4 of the airflow outlet of the returner will differ by a backward angle. so that the outlet angle of the returner is 83°. By calculation, the radius of the arc of the blade of the refluxer blade is R_k=107.65mm, and the radius of the center circle of the blade is R_0=126.1mm. The returner after the transformation is shown in figure 3.

—— Unoptimized blade shape
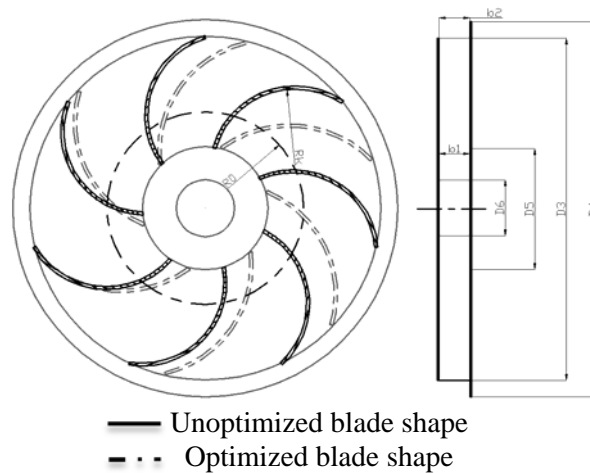– ·· Optimized blade shape

Figure 3. Before and after the transformation of the returner

4. Optimize fan performance and results analysis

The test data after the transformation of the fan is shown in table 2. After analysis, the pressure of the fan at the working flow of 1900m3/h was increased from 16153pa to 17271pa, the pressure was increased by 6.9%, and the efficiency was increased from 61.41% to 64.64%. In the working flow range of 300m3/h to 1900m3/h, the pressure curve is flat and the working range becomes larger. Comparison of test performance before and after the fan transformation is shown in figure 4. The overall advantages of the fan's overall efficiency are relatively advanced compared to the design conditions. Observing the consideration of the size of the fan, the design ideas of multi-stage fan bends and diffusers can be omitted, resulting in the possibility that the gas may be severely separated during the circulation in the air duct. Loss, secondary flow loss and impact loss, etc. The optimized fan blades have greatly improved the impact loss of the fan airflow, and the test results also verified the corresponding relationship between the airflow inlet angle of the blade and the fan operating flow.

Table 2. Test data after the transformation of the fan

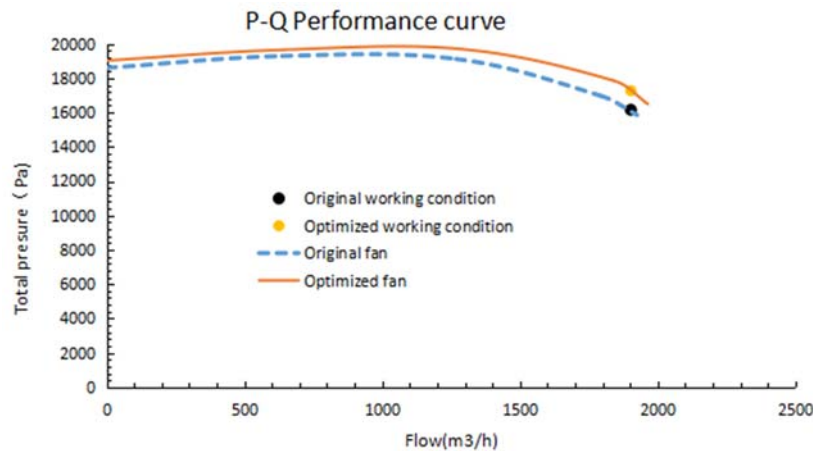| Test point	Flow (m³/h)	Total pressure (Pa)	Motor power (KW)	Efficiency
1	0.21	19025.67	4.81	0.03%
2	643.17	19651.15	7.93	50.71%
3	1287.60	19689.65	12.02	67.12%
4	1839.29	17856.31	15.47	66.66%
5	1964.39	16798.05	16.52	61.58%

Figure 4. Comparison of fan performance test before and after the transformation

5. Conclusion

The original fan blade inlet angle was changed from 33.5° to 30°, and the impact loss of the airflow was reduced by reducing the airflow angle of attack at the working flow rate.

Under the working condition flow rate, the rotary impeller and the returner blade are optimized. The overall wind pressure of the fan is increased by about 2.3%, and under the working condition flow rate, the total pressure is increased by about 6.7%.

For multi-stage centrifugal fans that eliminate the design of diffusers and curves, the fluid loss coefficient in the flow path of the fan is large. Changing the linear shape of the blade has less effect on the loss in the flow channel, and by changing the angle of the blade inlet and outlet, it reduces The impact loss optimization method works better.

Acknowledgments

References

[1]   Hu J C, Cui X Y, Ji Z G, "Numerical simulation and analysis of internal flow field in miniature centrifugal fans,"University of Shanghai for Science and Technology, 2010. 32 (3), 220-223.

[2]   Huang, C.H., Hung, M.H, "An optimal design algorithm for centrifugal fan,"theoretical and experimental studies. J. Mech. Sci. Technology, 2013.27, 761–773 .

[3]   He X B,"Research on Inner Flow Fields and Structural Optimization of Multistage Centrifugal Fan," South China University of Technology, 2012.

[4]   Xi G, Wang Z H, Wang S J, "Approximate model method and its application in turbomachinery aerodynamic optimization design,"Journal of Xi'an Jiaotong University, 2007, 41 (2): 125-135.

[5]   Li J Y, Niu Z, Liang Y, "Centrifugal Impeller Optimization Design Method for Controlling Average Flow Velocity," Proceedings of the Conference of Engineering Thermophysics Fluid Mechanics Conference, 2008.

[6]   Zhao Y B, Wang C M, "Numerical simulation of flow field of fan based on mass inlet boundary," Compressor Blower & Fan Technology, 2010 (3): 20-28.

[7]   Cheng X D, "Blade pump, ventilator, compressor (principle, design, operation, strength)," Beijing: China Machine press, 2011: 48-86.

[8]   Kui C, Wang H Q, Gai J F, "Fan manual, " Beijing: China Machine press, 2010: 696-720.