DOI: 10.3901/CJME.2013.06.1267, available online at www.springerlink.com; www.cjmenet.com; www.cjmenet.com.cn

Multi-objective Optimization Design and Experimental Investigation of Centrifugal Fan Performance

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Received October 11, 2012; revised July 17, 2013; accepted July 22, 2013

Abstract: Current studies of fan performance optimization mainly focus on two aspects: one is to improve the blade profile, and another is only to consider the influence of single impeller structural parameter on fan performance. However, there are few studies on the comprehensive effect of the key parameters such as blade number, exit stagger angle of blade and the impeller outlet width on the fan performance. The G4-73 backward centrifugal fan widely used in power plants is selected as the research object. Based on orthogonal design and BP neural network, a model for predicting the centrifugal fan performance parameters is established, and the maximum relative errors of the total pressure and efficiency are 0.974% and 0.333%, respectively. Multi-objective optimization of total pressure and efficiency of the fan is conducted with genetic algorithm, and the optimum combination of impeller structural parameters is proposed. The optimized parameters of blade number, exit stagger angle of blade and the impeller outlet width are seperately 14, 43.9°, and 21 cm. The experiments on centrifugal fan performance and noise are conducted before and after the installation of the new impeller. The experimental results show that with the new impeller, the total pressure of fan increases significantly in total range of the flow rate, and the fan efficiency is improved when the relative flow is above 75%, also the high efficiency area is broadened. Additionally, in 65%–100% relative flow, the fan noise is reduced. Under the design operating condition, total pressure and efficiency of the fan are improved by 6.91% and 0.5%, respectively. This research sheds light on the considering of comprehensive effect of impeller structrual parameters on fan performance, and a new impeller can be designed to satisfy the engineering demand such as energy-saving, noise reduction or solving air pressure insufficiency for power plants.

Key words: centrifugal fan, impeller structural parameters, optimization, experimental investigation

1 Introduction

As important auxiliary equipment for power plants, blowers and induced draft fans take large amount of primary energy consumption. Improving the efficiency of fans is of great significance to energy saving and emission reduction. Additionally, the pipe network resistance sometimes increases due to ash deposition and other reasons after the fan is operated for a period of time, leading to the insufficiency of fan flow or total pressure. Fan performance parameters such as efficiency, pressure coefficient, flow coefficient and the noise level mainly depend on the flow field in impeller and volute, so the optimization on impeller or volute structural parameters is necessary for meeting the requirement of performance and energy saving.

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This project is supported by Hebei Provincial Natural Science Foundation of China(Grant No. E2012502016), Fundamental Research Funds for the Central Universities,China(Grant No. 12QN39), and PhD Programs Foundation of Ministry of Education of China(Grant No. 20110036110009)

Previous studies on the volute optimization mainly focus on the volute profile and the internal flow of vortex[1–2]. GUO, et al[3], found that the influence of the different tongue profiles on the performance and operation stability of the centrifugal pump is very remarkable. The high-efficiency range of the centrifugal pump can be widened to some extent while the profile of the tongue is replaced from sharp tongue to middle tongue, and the maximum efficiency point is shifted along the higher flow rate direction. Based on Jameson's time-stepping scheme, WANG, et al[4], studied the swirling flow and the varying details of vortex along different sections in a trapezoidal volute. His results showed that both vortices experience complicated variations along circumferential flow path with respect to volute's different operating conditions. WANG, et al[5], studied the large-scale vortex in volute of a centrifugal fan and designed a vortex-broken device based on the idea of decrease flow loss and vortex noise through breaking large-scale vortex. With the device installed, when relative flow is more than 45%, total pressure increases significantly and high efficiency area is broaden.

The impeller is the most important working component in a fan, and it is a fast and convenient way of retrofitting the original fan with a new type of impeller to optimize the impeller structure without changing the volute and current collector. Based on the entropy production calculation, WANG, et al[6], found that the entropy production in impeller region is the highest. So more optimization studies focus on the impeller area[7–11]. ROGALSKY[12], optimized a two-dimensional airfoil based on the inverse optimization algorithm and the program with the C++ software, and he obtained the proper lobe type, gate angle and aspect ratio. KAZUYUKI[13], developed an aerodynamic shape optimization method suitable for centrifugal impellers and diffusers. The shapes were parameterized using non-uniform rational B-spline curves with special attention being paid to the blade's edge profiles. The simulated annealing and neural network took turns in controlling the optimization processes, not only for maximizing the efficiency of global exploration, but also for minimizing the risks of automation failures or of reaching an incorrect optimum. CHON, et al[14], conducted optimization design and calculation on the fan airfoil under given condition with the established expert system. In his study, the significant factors for the centrifugal fan such as input variables, output variables and constraints were conceptualized. HUANG, et al[15], examined the inverse design problem of determining the optimal three-dimensional shape of a centrifugal fan based on a desired airflow rate. The geometry of the redesigned fan was generated using several design variables, which enables the shape of the fan to be constructed completely. And he verified that the performance of optimal fan is greatly improved. LEE, et al[16], presented a method for redesigning a centrifugal impeller and its inlet duct. He utilized a numerical optimization with experiential steering techniques to redesign the fan blades, inlet duct, and shroud of the impeller. BAMBERGER, et al[17], optimized a fan with swept blades with respect to sound emission and efficiency. Geometrical parameters describing variations of the blade section and the hub contour were defined. And A moderate increase in efficiency at the design point was observed. HUANG, et al[18], combined the advantages of splitter blades and air blowing at boundary layer, and conducted optimization on the parameters of slitting blade. The results showed that the total pressure of the fan increases while the noise decreases. LI, et al[19], investigated the influence of enlarged impeller in unchanged volute on G4-73 type centrifugal fan performance. Comparisons were conducted between the fan with original impeller and two larger impellers with the increments in impeller outlet diameter of 5% and 10% respectively in the numerical and experimental investigations. Her results indicated that the flow rate, total pressure rise, shaft power and sound pressure level have increased, while the efficiency have decreased when the fan operates with larger impeller. SHI, et al[20], studied the performances of the deep-well centrifugal pump with four different impeller outlet widths by the numerical, theoretical and experimental methods. His results verified that the oversize impeller outlet width leads to poor pump performances and increasing shaft power. Changing the performance of deep-well centrifugal pump by alter impeller outlet width is practicable and convenient, which is worth popularizing in the engineering application.

In summary, optimization of fan performance mainly focuses on two aspects: one is to improve the blade profile, and the other one is that only considering the influence of single impeller structural parameter on fan performance. However, there is little research on the comprehensive effect of the key parameters such as blade number, exit stagger angle of blade and the impeller outlet width on the fan performance.

The rest of this paper is organized as follows. First, the orthogonal experiments scheme is designed with three factors, including blade number, exit stagger angle and the impeller outlet width. The training samples can be obtained by means of parametric numerical computing platform. And the fan performance parameters prediction model with different impeller structural parameters is established based on the BP neural network model. Then the impeller structural parameters are optimized for the maximization of total pressure and efficiency with genetic algorithm. Finally, to verify the correctness of optimization, a new impeller with the optimized parameters is made. And the performance test and noise test are separately conducted on the fan before and after the installation of the new impeller.

2 Performance Parameters Prediction Model Based on the BP Neural Network

2.1 Geometric model

The study on G4-73 type centrifugal fan, which is widely used in power plants, is of great practical significance and engineering value. As shown in Fig. 1, the centrifugal fan is composed of current collector, impeller, volute and an anti-vortex ring inside. The impeller diameter is 80 cm, exit stagger angle of blade β_{2y} is 45°, and the blade number Z is 12. The airfoil blades are circumferentially distributed in the impeller. The impeller outlet width b_2 is 20 cm. The axial width of volute is 52 cm, and the volute tongue gap is 8 cm.

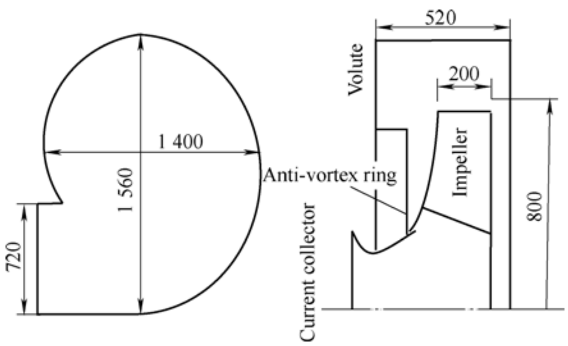


Fig. 1. Structural diagram and main dimensions of fan

2.2 Orthogonal design

The effect of impeller structural parameters, including the blade number, exit stagger angle and the impeller outlet width on fan performance such as total pressure and efficiency should be fully considered. There are three factors considered, and each factor has many levels. So the number of tests will be too large if comprehensive tests for each level of each factor are conducted. For example, if the tests have five factors and four levels, then the number of the comprehensive experiments is 1 024. It's a waste of time and money. However, it will cause poor study ability of the machine with random or less tests. So it's important to select the samples wisely. Orthogonal design is a method which can make scientific arrangement and analysis of multiple factor with the use of orthogonal table.

When a factor's quality cannot be affected by another factor's quality, there is no interaction between the two factors. And it's called interaction between factor A and B when one factor's quality level is affected by another's, denoted by A×B. In this paper, when one of the impeller structural parameters changes, the influence of the other parameters on fan performance almost can be ignored. The relevant numerical calculations have been carried out as shown in Fig. 2. The results show that the interaction between the three impeller structural parameters has little influence on the fan performance. Therefore, the interaction between the factors is neglected.

Orthogonal table L49 (7^3) was used without considering the interaction between the factors. Where L represents orthogonal table, 49 represents the number of rows, namely the number of the test required to do. And 3 represents the number of columns, namely the max number of the factors. 7 represents the number of levels. The factors are arbitrarily arranged to each row as shown in Table 1.

The 49 experiments were conducted with the use of parametric numerical computing platform developed by our research team. Then the 49 samples were obtained to establish the performance parameters prediction model.

2.3 Parametric numerical computing platform

Parametric numerical computing platform mainly consists of parametric modeling module, calculation module and data analysis module. The function of parametric modeling module is to obtain the impeller parameters and processing methods for geometric modeling, and provide the mesh size input for the geometry. The impeller outlet width, blade number and exit stagger angle are taken as variables, and the other structural parameters are invariable. Figs. 3(a), 3(b) illustrate the parametric modeling module input interface. The impeller main structural parameters can be input and the modeling mode can be selected in Fig. 3(a). After setting the mesh options in Fig. 3(b), modeling program starts, then automatic geometric modeling and meshing are conducted, and a mesh file used for numerical calculation can be exported.

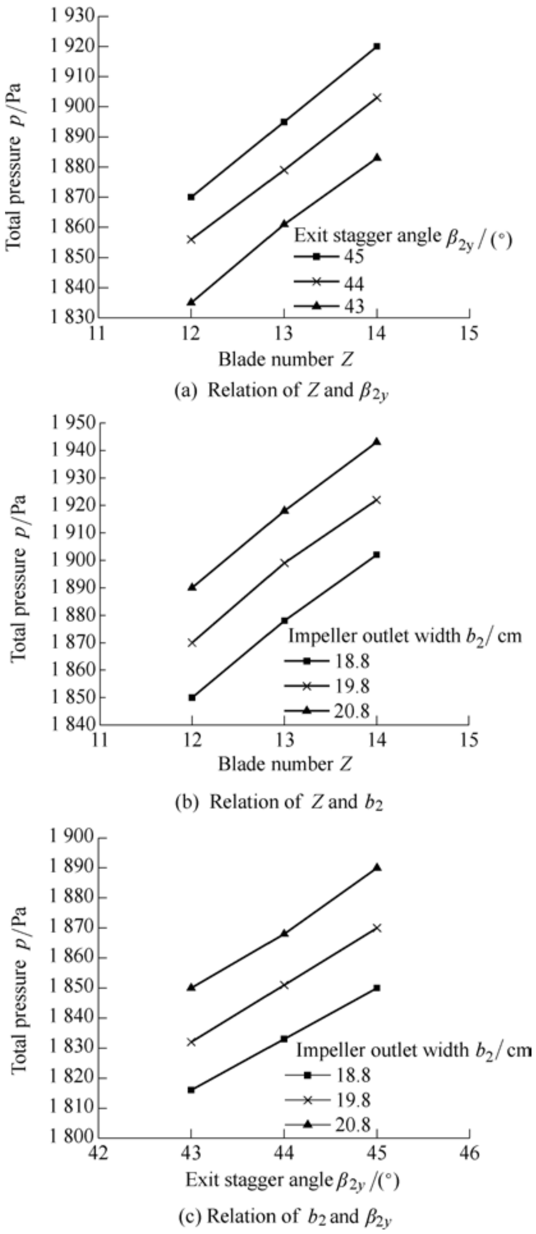


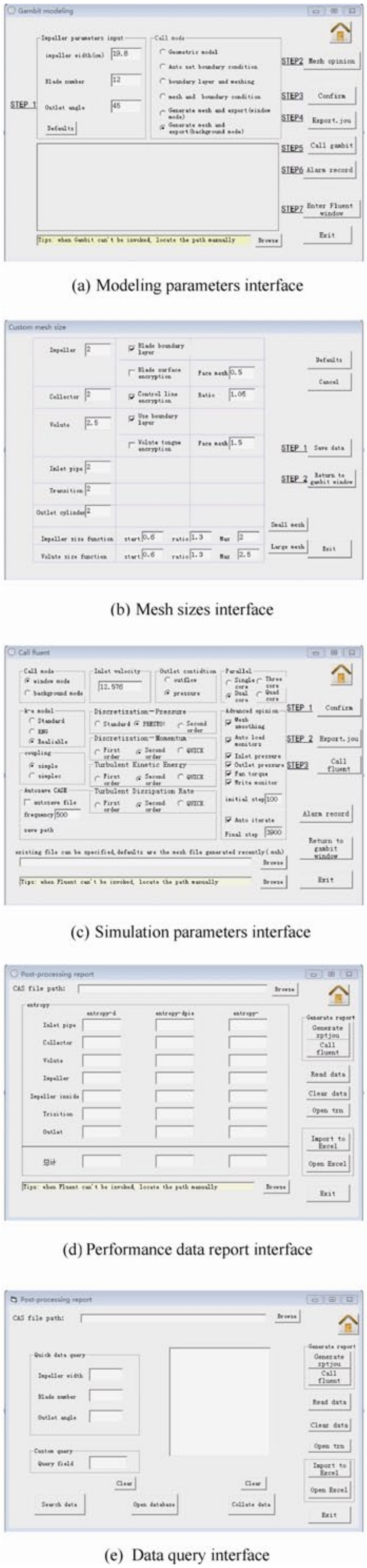
Fig. 2. Influence of the interaction of three parameters on fan performance

Fig. 3(c) illustrates the numerical calculation module interface. Numerical calculation module can automatically realize the grid quality check, solving model loading, discrete format selection and setting of boundary conditions. According to the selections, automatic calculation can be done, and the files used for post-processing can be exported (CAS file and DAT file). The working process doesn't need additional intervention, and the module has the ability of error monitoring and reporting, which is quite convenient when used for simulated computations on many different operating conditions.

Fig. 3(d) shows the data analysis module interface. After the specified CAS and DAT files are selected, the pressure value, the pressure difference and the entropy production of the inlet and outlet of each part are displayed in the interface automatically. Meanwhile, the total pressure and efficiency of fan are also calculated. The module also provides historical data query function as shown in Fig. 2(e).

Table 1. Orthogonal design scheme

Serial No.	Blade No. Z	Exit stagger angle β_{2y} / (°)	Impeller outlet width b_2 / cm
1	10	47	21.3
2	10	41	18.3
3	10	42	18.8
4	10	43	19.3
5	10	44	19.8
6	10	45	20.3
7	10	46	20.8
8	11	45	18.8
9	11	46	19.3
10	11	47	19.8
11	11	41	20.3
12	11	42	20.8
13	11	43	21.3
14	11	44	18.3
15	12	43	19.8
16	12	44	20.3
17	12	45	20.8
18	12	46	21.3
19	12	47	18.3
20	12	41	18.8
21	12	42	19.3
22	13	41	20.8
23	13	42	21.3
24	13	43	18.3
25	13	44	18.8
26	13	45	19.3
27	13	46	19.8
28	13	47	20.3
29	14	46	18.3
30	14	47	18.8
31	14	41	19.3
32	14	42	19.8
33	14	43	20.3
34	14	44	20.8
35	14	45	21.3
36	15	44	19.3
37	15	45	19.8
38	15	46	20.3
39	15	47	20.8
40	15	41	21.3
41	15	42	18.3
42	15	43	18.8
43	16	42	20.3
44	16	43	20.8
45	16	44	21.3
46	16	45	18.3
47	16	46	18.8
48	16	47	19.3
49	16	41	19.8



(a) Modeling parameters interface

(b) Mesh sizes interface

(c) Simulation parameters interface

(d) Performance data report interface

(e) Data query interface

Fig. 3. Interfaces of parametric numerical computing platform

2.4 Data normalization

The transfer function of the neural network has a limit on the input, and the input is limited in the range of [0, 1] or [−1, 1]. The units and magnitude of the original data are different. It can easily lead to neuronal supersaturation and make certain features with lower magnitude submerged. Therefore, it is necessary for the original data to be normalized. The function of mapminmax in MATLAB was selected, and all the data are in the range of [−1, 1]. The function can be expressed as follows:

$$x_n = \frac{2(x - x_{\min})}{x_{\max} - x_{\min}} - 1, \tag{1}$$

where $x_{\max}$ and $x_{\min}$ are respectively the maximum and minimum values of the variables in all operating conditions, x_n is the normalized data, and x is the original data.

2.5 Establishment and verification of prediction model

Based on the platform of MATLAB 7.10, a forward BP neural network was created with the use of newff. A single hidden layer of the network structure was adopted. The transfer function from the input layer to the hidden layer is the S-shaped tangent function tansig. And the transfer function from hidden layer to the output layer is a linear function purelin.

L-M algorithm was adopted as the network training function, namely trainlm training function with the learning rate set to 0.01 and training mean square error less than 0.001. The trail calculation of the hidden nodes was conducted at the same time, and a network structure with 11 nodes was used.

Comparisons of the sample values and predicted values of total pressure and efficiency obtained from the fan performance prediction model are shown as Fig. 4. The figure shows that the training samples and predicted values are almost equal, and the maximum relative errors of total pressure and efficiency are respectively 0.974% and 0.333%. It can be seen that BP neural network shows strong generalization ability and fully reflects the inherent law of reaction parameters, which can be used for the prediction and optimization of the fan performance.

3 Multi-objective Optimization of Fan Performance Based on Genetic Algorithm

3.1 Genetic algorithm

Genetic algorithm is the most widely used and successful algorithm of intelligent optimization methods. The basic idea is to construct a fitness function to evaluate, genetic calculate and choose a population of multiple solutions based on the objective function of the problem. Through multiple generations of breeding, the individual with the best fitness value is obtained as the optimal solution of the problem.

Genetic algorithm consists of five basic elements: parameter encoding, the setting of the initial population, the design of the fitness function, the design of genetic manipulation, and the setting of control parameter. The real coding method was selected in this study, which is suitable for a large range of number and searching space with high precision and efficiency.

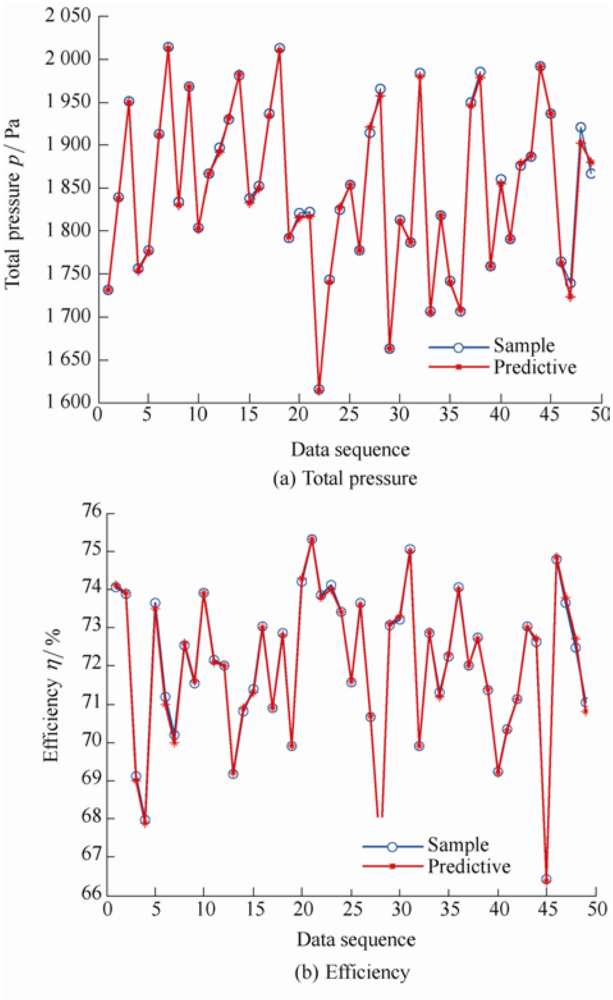


Fig. 4. Comparisons of predictive values and sample values of fan performance parameters

Genetic manipulation includes three basic genetic operators: selection, crossover and mutation. Roulette wheel selection method was adopted, which is a selection strategy based on the proportion of the fitness.

3.2 Global optimization fitness function

Optimization design based on parametric numerical calculation includes optimization variables, constraints and objective function. In this paper, the goal is to optimize the efficiency and the total pressure of the fan, thus the fitness function conducts linear summation, the two target multiplied by the weighting respectively, and the multi-objective problem can be turned into single objective problem. With the change of weight, optimization results also have each emphasis. According to the actual need, appropriate weight was proposed, and the optimal solution was obtained. Global optimization fitness function is as follows:

$$\min F = -m \cdot \eta - n \cdot p + k\,, \tag{2}$$

where η is the efficiency, p is the total pressure, m, n are respectively the weight of efficiency and total pressure, and $m+n=1$. k is a constant, that ensures the positive value of fitness function.

3.3 Optimization of parameters and constraints

The geometric parameters of volute and collector are kept invariant with the original fan for saving cost and the space limitation of plant. And the blade profile is also the same with the original fan for convenience of manufacture. The blade number, exit stagger angle and the impeller outlet width are selected as the optimization variables. Considering the restrictions on fan flow range and the number of numerical simulation, the optimization variables must be restricted to a certain range, otherwise it could lead to the optimization results inconsistent with the actual situation. Therefore, each variable was limited to the range defined in the orthogonal table before.

3.4 Optimization results

Real-coded genetic algorithm was selected with the population size set to 100 and evolution algebra set to 100 times. The selection strategy of the individual entering genetic algorithm was roulette method with crossover probability 0.8 and mutation probability 0.01.

Due to the increasement of pipeline resistance, some fans suffer insufficiency of efficiency and total pressure after running for some time. The optimization goal is to improve the total pressure and the efficiency at the same time with the resistance curves remaining the same. Therefore, the weight of total pressure and efficiency are respectively 0.7 and 0.3.

Fig. 5 shows the optimization process of total pressure and efficiency. It can be seen from the figure that probably after 50 times of evolutionary computation, the optimal values of total pressure and efficiency are obtained. The impeller structural parameters are as follows. The blade number is 14, exit stagger angle of blade is 43.9°, the impeller outle width is 21 cm, and the optimization results of total pressure and efficiency are respectively 2 019.71 Pa and 73.02%. The picture of the new impeller with optimized structural parameters is shown in Fig. 6.

3.5 Numerical results

Numerical simulations of fan performance before and after the optimization were conducted with parametric numerical computing platform. The fan performance curves are shown in Fig. 7. It indicates that after the optimization, the fan total pressure improves significantly. At the location of maximum efficiency, the total pressure and the efficiency are improved by 146 Pa and 0.78%, respectively.

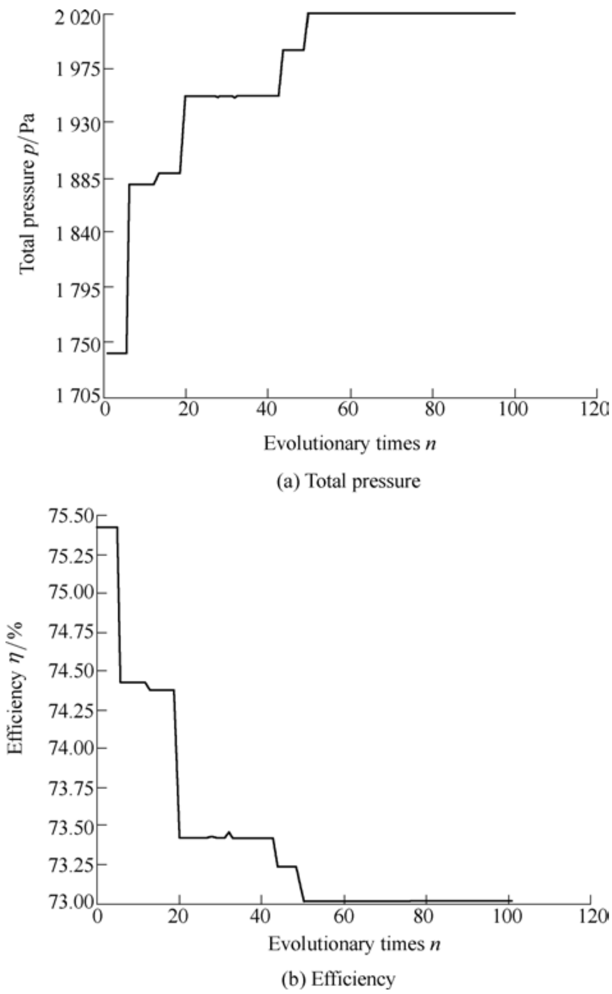


Fig. 5. Searching process for total pressure and efficiency of fan

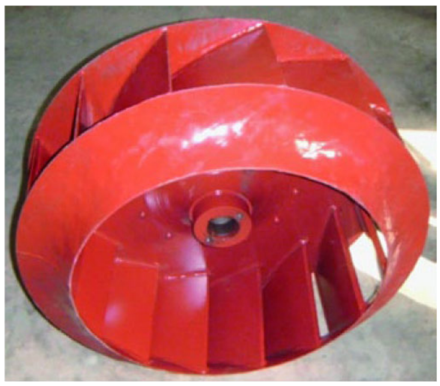


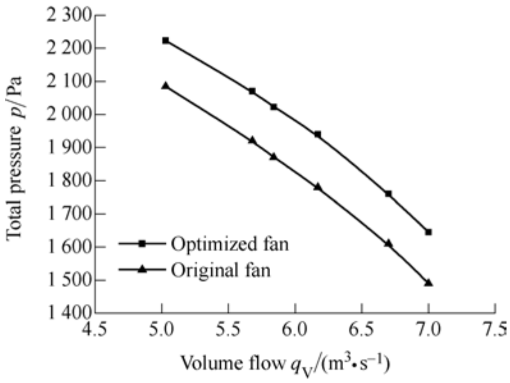
Fig. 6. Picture of optimized impeller

It should be noted that rotating stall phenomenon will occur when flow rate is low, and the correct solution cannot be obtained by steady calculation. Therefore, the corresponding numerical simulations were conducted near the design condition.

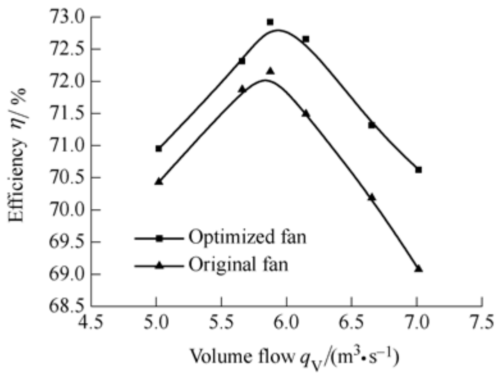
4 Experimental Study on Optimization of Fan Performance

4.1 Performance tests

In order to verify the actual effect of the optimization, the performance tests were respectively conducted on the fan with the original impeller and the optimized impeller. The operations followed GB/T1236–2000 "Industrial fans-performance testing using standardized airways"[21]. And the experimental device mainly includes an air inlet pipeline, centrifugal fan, axial guide vane, connection pipeline, pressure measuring devices and the throttle cone.



(a) Total pressure vs. volume flow curves



(b) Efficiency vs. volume flow curves

Fig. 7. Fan performance curves obtained from numerical simulation

A throttle cone was adopted to regulate the fan flow. The flow was measured by tilting micro-manometer at the inlet pipe. The static pressures of the inlet and outlet pipes were separately measured with U type tube. The dynamic pressure of outlet pipes could be obtained by the pitot tube and the tilting micro-manometer. A CYB-803S type torque sensor was used to measure the shaft power with the measuring range of 200 N • m. With non-contact transmission mode, the sensor has high measuring precision and small data deviation. The motor type is Y180L-4 with the rated power of 22 kW. A frequency converter was used to regulate the fan speed. The experimental platform and arrangement of equipments are shown in Fig. 8.

In the performance tests of centrifugal fan, the errors of total pressure and efficiency are mainly resulted from the systematic errors caused by the accuracy of measuring instruments. The accidental error has been minimized with multiple tests. The accuracy of instruments used in performance tests and measuring errors of performance parameters are shown in Table 2.

The measured data were processed by Origin software, and B-spline interpolation was used to fit curve. The performance curves of fan between total pressure and relative flow before and after the installation of new impeller are shown in Fig. 9. The figure shows that in the full flow range, total pressure of the fan with new impeller is significantly higher than that of the original fan. When the relative flow is 100%, the total pressure of fan increases by 126.7 Pa, which is 6.91% of total pressure of the original fan.

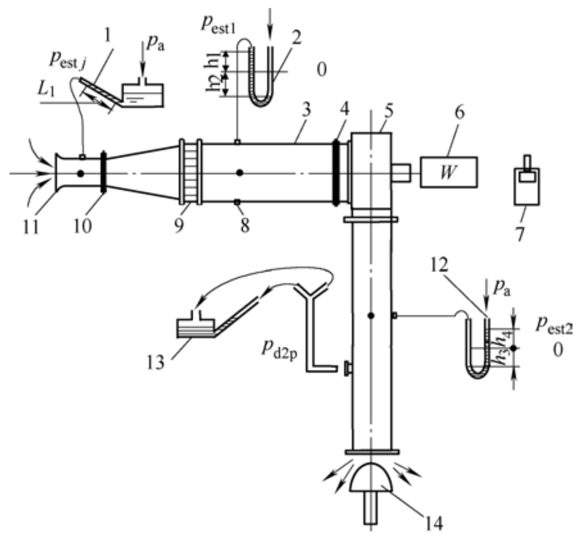


Fig. 8. Experimental arrangement of centrifugal fan

1. Tilting micro-manometer;
2. U-tube manometer (inlet static pressure);
3. Inlet pipeline; 4. Axial guide vane;
5. Centrifugal fan; 6. Torque sensor;
7. Handheld tachometer; 8. Static pressure measuring hole;
9. Stabilizing grid; 10. Throttling network;
11. Current collector;
12. U-tube manometer(outlet static pressure);
13. Tilting micro-manometer; 14. Throttle cone;

Table 2. Accuracy of instruments and measuring errors

Instrument	Range	Accuracy /%	Fan performance parameter	Relative error /%
YYY-2000 tilting micro manometer	0–2 kPa	1.0	Flow rate	1.73
U-tube manometer	0–2 kPa	1.0	Total pressure	1.38
CYB-803S-type torque meter: Torque	0–200 N • m	0.1	Power	0.50
CYB-803S-type torque meter: Speed	0–3 000 r/min	0.5	Efficiency	2.25

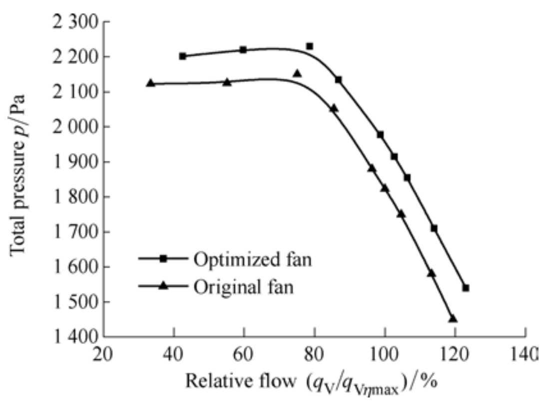


Fig. 9. Total pressure vs. relative flow curves

Fig. 10 shows the performance curves of fan between efficiency and relative flow before and after the installation of new impeller. The figure shows that when the relative flow is less than 75%, the efficiency of the fan with new impeller is lower than that of the original fan. When relative flow is more than 75%, the efficiency of the new fan is higher than that of the original fan. And as the flow increases, the efficiency also improves. At the maximum efficiency point, the efficiency is 0.5% higher than that of the original fan, and the high efficiency area is broadened. Due to the increase of impeller outlet width and blade number, the flow rate at the maximum efficiency point of the new fan increases, which is 103% of the original fan, and the efficiency at this point also increases by 1%.

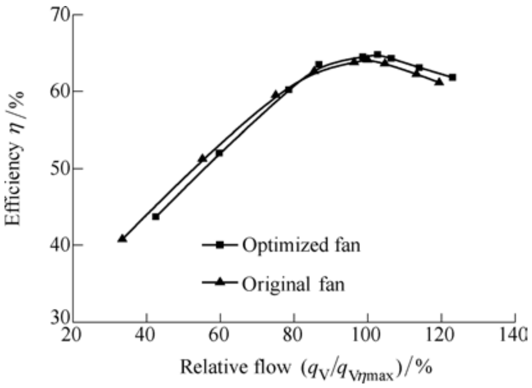


Fig. 10. Efficiency vs. relative flow curves

With the reduction of exit stagger angle, and the increase of impeller outlet width and blade number, the performance of fan are improved. When the impeller rotation speed remains the same, the decrease of exit stagger angle can lead to decrease of the impeller wake region and expansion of the shear layer between the jet-wake structures. Also, the intensity of the entropy production area gets weakened, and the flow field is enhanced. Due to the decrease of exit stagger angle, the positive incidence angle decreases, leading to the improvement of the blade inlet flow field. The increase of blade number reduces the diffusion of the impeller passage, and weakens the impact of axial vortex, which also makes flow field better. The impeller outlet width mainly influences the fan flow. When the outlet width is broadened, the hydraulic loss can be reduced, and friction loss between the impeller and volute also decreases. Therefore, through the optimization of the three impeller structural parameters above, the fan total pressure and efficiency can be improved.

The analysis above shows that the new designed impeller can solve the following problem. After a period of operating time, due to the increase of pipeline resistance, the total pressure and flow tend to be insufficient. Without moving the volute, collector and changing the pipeline resistance curve, the installation of the new impeller can lead to considerable increase in total pressure, and slight improvement in efficiency.

4.2 Noise tests

An AWA6270A noise spectrum analyzer was used to measure noise, which is applicable to all types of transient noise measurement and spectrum analysis of noise. Noise measurement followed GB2888-1982 “fan and blower noise measurement method”[22]. A-weighted sound level noise of the fan in different flow was measured according to the measurement standard. The measuring instrument and the measuring point arrangement are separately shown in Fig. 11 and Fig. 12.

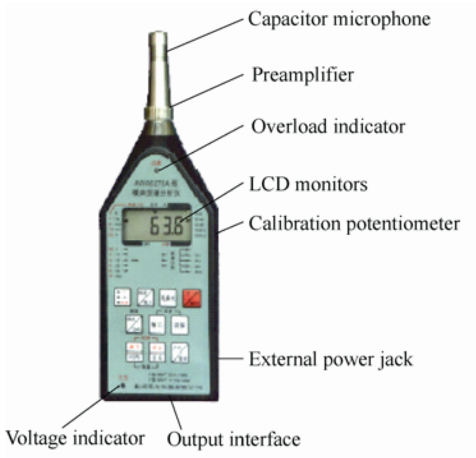


Fig. 11. AWA6270A noise spectrum analyzer

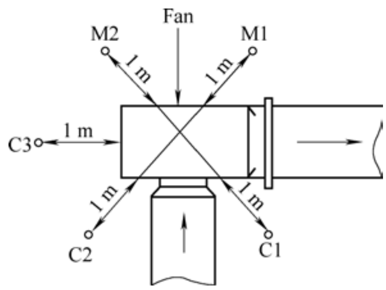


Fig. 12. Noise measuring points arrangement diagram

Fig. 13 shows the relationship between A-weighted sound level noise and relative flow before and after optimization. It indicates that with the increase of flow rate, the noise decreases first, and then increases, and the fan has the lowest noise when running at 85% relative flow. While under low flow, the periodic pressure and velocity fluctuation increase greatly when the rotary blades pass by the volute tongue, leading to the deterioration of flow field in the fan, or even serious destructive vibration. Compared with the original fan, when the relative flow rate is between 65% and 100%, fan noise with the new impeller is lower. When the relative flow is 85%, the noise reduces about 0.5 dB, and the noise under design condition is substantially constant. While in other flow range, noise of the fan with optimal impeller increases, but the maximum of increase doesn’t exceed 0.8 dB. Since the centrifugal fan often runs in the range of 60%–100% relative flow in practice, after the optimization of impeller structural parameters, with the total pressure and efficiency improving, the noise is effectively controlled at the same time.

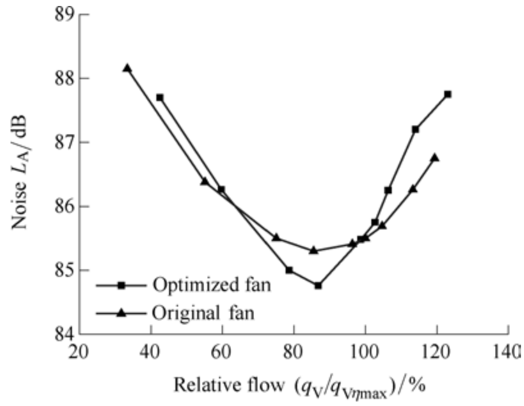


Fig. 13. A-weighted sound level noise vs. relative flow curves

5 Conclusions

(1) Based on orthogonal design and BP neural network, a centrifugal fan performance parameter prediction model is established, which has a high accuracy of prediction. The optimum combination of impeller structural parameters is obtained using genetic algorithm.

(2) The total pressure of fan is increased significantly in the total range of the flow rate, and the high efficiency area is also broadened after installing the new impeller. Moreover in 65%–100% relative flow, the fan noise is reduced. Also, the mechanism of the new impeller with better performance is presented.

(3) The presented method is also meaningful for performance optimization of the similar fan. And the new impeller obtained has great engineering value for energy saving and solving the insufficiency of wind pressure in power plants.

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