A Study of Product Design and Experimental Analysis for the Centrifugal Cooling Fan
離心式散熱風扇產品設計與實驗分析

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Abstract

The purpose of this study is to investigate a small centrifugal cooling fan applied on notebook computers, which is consisted of a forward-curved impeller and a centrifugal housing. First of all, a basic rotor impeller is designed with the cascade theory. To manufacture the mock-up for experimental verification, the design fan will be expressed in the CAD/CAM format for the CNC machine. All experiment regarding performance characteristic in this analysis is executed in AMCA 210-85 chamber. Thereafter, a parametric study on the geometry of fan is performed to enhance the understanding of the influences caused by those variations. The parameters considered here include the blade number, blade geometry and hub diameter. Experimental results show that blade designed with novel geometry enhances volume flow rate and static pressure outcome. The geometry of blade are demonstrates significant improvements on volume flow rate increment 26.9% and static pressure increment 42.6%. Moreover, an extensive discussion for the influences on above parameters is provided.

摘要

本文是針對筆記型電腦之離心式散熱風扇產品作一研究，此新式風扇乃是由前傾式葉輪與離心風扇之外殼組合而成。首先，依據扇葉理論設計風扇之葉形，再經由 CAD/CAM 系統轉換成加工程式，運用 CNC 加工方式，將散熱風扇葉形實體製造出來。而後利用性能量測設備，以獲取原型風扇之流量及壓力性能曲線。然後再藉由變更各項風扇幾何參數，以評估其對離心式散熱風扇的性能及噪音的影響。此外，爲了確保實驗的可靠性，分別以符合 AMCA210-85 規範之實驗設備進行性能測試。實驗結果顯示，

新式翼型設計葉形對於風扇之性能提升確實有顯著之成效，其中修正葉片幾何形狀，對流量與靜壓皆有明顯的提升，分別提升了 26.9 % 及 42.6 %。其餘的幾何參數影響，在風扇性能提升研究均值得再深入探討。

Keywords: Notebook Computer, Centrifugal Fan, Product Design

關鍵詞：筆記型電腦、離心式風扇、產品設計

I. Introduction

In recent years, the notebook computer is continuously developing in shrinking its size and turning high performance. That would cause the using of power of CPU getting increased. Therefore, the junctions inside the chips would easily be destroyed due to the exceedingly high temperature. Moreover, it will do a great harm to the electronic facilities. This trend urges the thermal management of notebook computer device to become gradually important. The demand for notebook computer is growing nowadays. Over the years, the technology for manufacturing chips has improved greatly. At this moment, engineers have achieved the success in integrating multiple functions into a single computer chip. However, the space within a computer unit is running out when many additional computer components are included in the system. This, on the other hand, reduces the passage for air and induces greater flow resistance. This undesirable flow resistance will severely influence the cooling conditions for CPU or VGA card.

As the number of the components in a computer unit increases, the flow passage within the computer system decreases accordingly and a defective design of the heat sink assembly will severely affect the cooling fan inlet condition. This problem is more critical for notebook computer in which heat management has long been a problem. Centrifugal cooling fans with high static pressure have been used in the heat sink assembly of the notebook computers. To maximize the fan efficiency, it is necessary to thoroughly understand the performance and noise generation associated to the flow patterns under the influence of blade. When the flow resistance upstream of a fan increases, it will greatly influence the fan performance. If the overall flow resistance is taken into consideration at the early stage of heat sink assembly design, the heat sink assembly will be much more capable of meeting the realistic thermal task for CPU cooling. A typical centrifugal cooling fan consists of an inlet cover, a rotor and housing, as shown in Fig. 1.

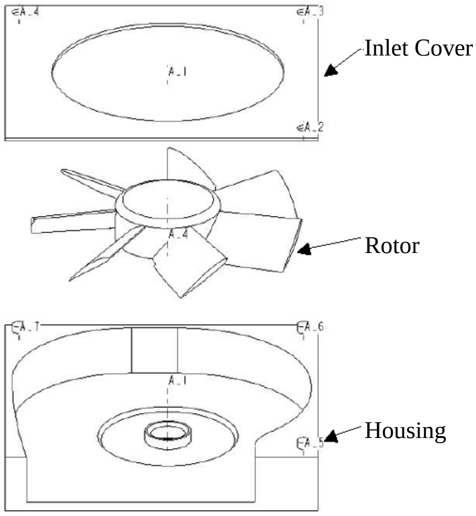


Fig. 1 The structures and components of a typical cooling fan for notebook computer

A fan with high static pressure is capable of effectively removing the heat energy generated from the system of high flow resistance. Therefore, the development of high-performance fan can effectively overcome the flow resistance due to system components. Since the technology of manufacturing fans has greatly improved, available fan models sold in the market are mostly designed following the design concepts proposed by Wallis [1] and Eck [2]. The blades for the fan with desirable specifications are produced in accordance with NACA blade shape, so that the fan parameters associated to high aerodynamic performance can be identified. After that, CAM is employed to convert the desired geometry into CNC-code to fabricate the prototypes for physical testing, as had been done by Lin et al. [3, 4, 5].

The purpose of this study is to investigate a small centrifugal cooling fan applied on notebook computers, which is consisted of a forward-curved impeller and a centrifugal housing. First of all, a basic rotor impeller is designed with the cascade theory. To manufacture the mock-up for experimental verification, the design fan will be expressed in the CAD/CAM format for the CNC machine. All experiment regarding performance characteristic in this analysis is executed in AMCA 210-85 chamber. Thereafter, a parametric study on the geometry of fan is performed to enhance the understanding of the influences caused by those variations. The parameters considered here include the blade number, blade geometry and hub diameter. Not only its influence on the performance and noise characteristics of the centrifugal fan is identified, but also the corresponding variation on the thermal resistance of a heat sink assembly is evaluated.

II. Experimental Apparatus and Procedure

Even the most sophisticated design method requires empirical results for validation. Actually, the best way to establish the performance and noise characteristics of any centrifugal cooling fan is by experimental testing. Experimental testing should be conducted on a setup that meets code requirements. The following subsections illustrate the setups used in this study.

A. Performance Measurement Setup

Many renowned engineering societies and industry organizations throughout the world have published fan test codes. Among them, AMCA Standard 210-99 [6] is one the most widely used and accepted in the fan industry. This test code was developed by the Air Movement and Control Association (AMCA) and was documented in the publication entitled “laboratory methods of testing fans for aerodynamic performance rating”. All performance characteristics of fans in this study are carried out in an AMCA test chamber to yield reliable measurements.

The AMCA test chamber and instrument setup are schematically presented in Fig. 2. A fan discharge is mounted on the AMCA test chamber to simulate free-inlet and free-outlet conditions. Flow settling means are installed in the chamber to guarantee proper flow conditions for measurements. For a measuring plane located upstream of the settling means, the settling screen absorbs the kinetic energy of the upstream jet and allows the flow to undergo a normal expansion. For a measuring plane downstream of the settling screen, this screen ensures a substantially uniform flow ahead of the measuring plane. Adding an auxiliary fan at the end of the test chamber controls the operating point of the fan.

In this experiment, the rotational speed is measured using a photo non-contact tachometer with a resolution of 1 rpm and accuracy up to 0.05%. The discharge static pressure is measured using a pressure transducer at location No. 9 and No. 10 in Fig. 2. The airflow rate can be calculated based on the differential pressure measured across the multiple nozzles (i.e., location No. 9 and No. 10) as shown in Fig. 3. Therefore, the measurement uncertainties of discharge air-flow-rate and static pressure are mainly affected by the accuracy and the calibration of the pressure transducer. Since the static pressure is very small for a typical cooling fan, a pressure transducer, designed for exceedingly low differential pressure measurement applications, is chosen in this experiment. With full ranges below 56 mm-H2O, the accuracy of this instrument is estimated to be within 0.25% of its full scale.

Moreover, a precise pressure calibration system (Fig. 3) is used to calibrate the ultra-low-range pressure transducers with a resolution of 0.05 mm-H2O and an accuracy of 0.5% of reading. The pressure calibration system consists of an accurate pump and an inclined manometer. A three-way valve is used to connect the pump with the manometer and the pressure transducer for ensuring an equivalent pressure level on both of them. Thus, the pressure exerted on the pressure transducer can be offered precisely by the pump and be read easily from the manometer. Hence, the relationship between the pressure loading and the output (i.e., current) of the pressure transducer can be established by changing the exerted pressure through the pump. In summary, the uncertainty of the performance measurement is estimated to be approximately 2.0%.

B. Noise Measurement Setup

In this experiment, A weighted and 1/3-octave band sound pressure levels are measured using a RION NL-14 portable sound level meter (SLM) and analyzed using an AND AD-3524 FFT frequency analyzer. The sound pressure creates an analog electric signal in the SLM microphone. From SLM, this signal is then fed into the FFT analyzer to generate noise characteristics. For verification, the consistency and calibration of the above devices are checked by a piston phone (94 dB at 1000Hz) both before and after a set of measurements. CNS-8753 code [7] is employed to measure the noise levels at the two fan outlet positions as shown in Fig. 4. To further guarantee meaningful measurements, the measurements are taken in a semi-anechoic chamber (Fig. 4) because it offers an appropriate test environment. The transmission decay is larger than 35 dBA. The noise level of the test sample must be 10 dBA lower than the noise level of the environment.

1. Test fan
2. Auxiliary fan
3. Variable supply
4. Flow straightener
5. Multiple nozzle
6. Static pressure tap is front nozzle
7. Static pressure tap is back nozzle
8. Static pressure tap of the inlet
9. Pressure transducer No. 1
10. Pressure transducer No. 2
11. Adjustment device of the optical fiber tachometer
12. Thermocouple
13. Optical fiber tachometer
14. data acquisition card
15. Utility sign switch card
16. Personal computer
17. Lacer printer
18. Thermometer and Hygrometer
19. Barometer
20. Digital frequency transformer
21. Power supply
22. Hand hole

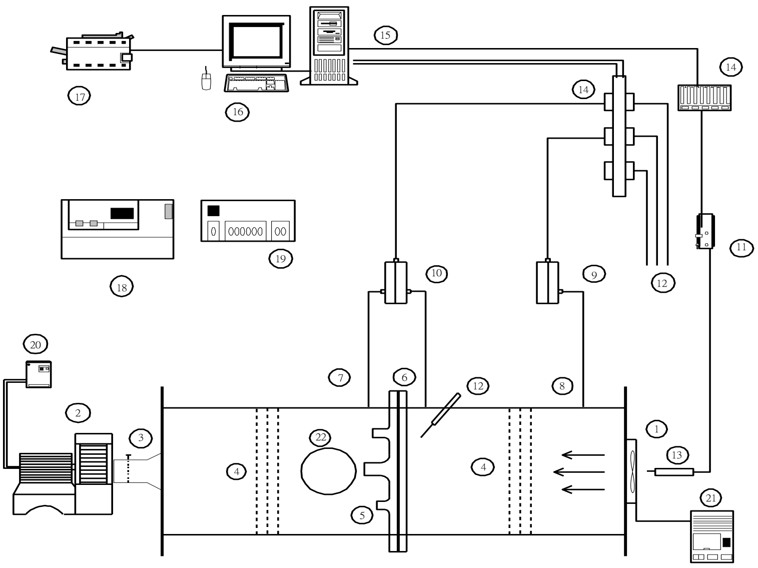


Fig. 2 Sketch of AMCA test chamber and instrument setup

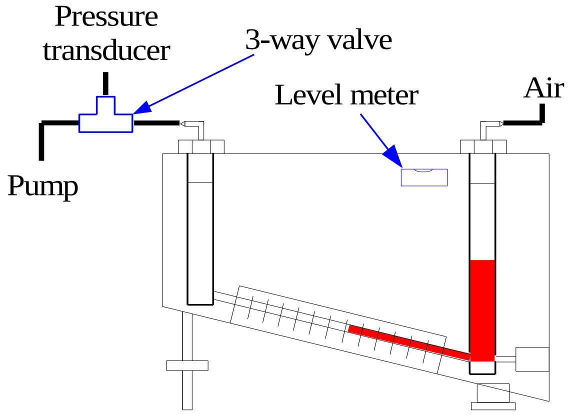


Fig. 3 The pressure calibration system

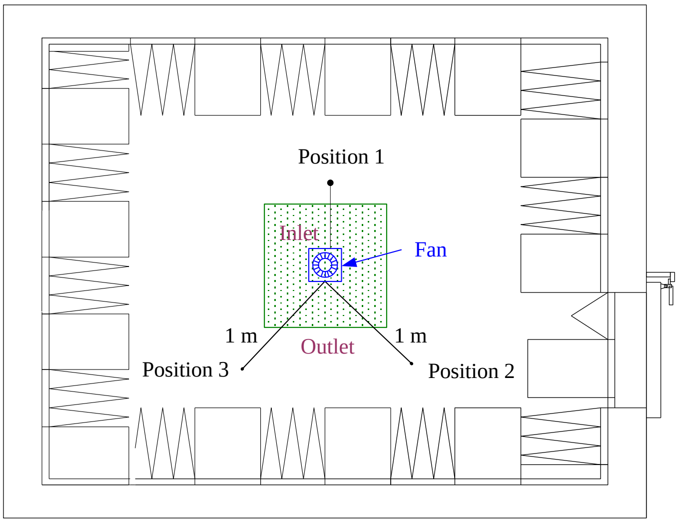


Fig. 4 Schematics for the positions of noise measurement

C. Experimental Procedure

The experimental procedure of this study is shown in Fig. 5. At first, the impeller of the original centrifugal cooling fan was analyzed. In the beginning, a centrifugal cooling fan available in the market was randomly chosen for comparison with the redesigned centrifugal cooling fan. The chosen centrifugal cooling fan is denoted as the base centrifugal cooling fan in this study. Also, the NAC A-4410 airfoil profile (1949) [8] was selected as the basic blade shape to meet the requirement used in the base centrifugal cooling fan for high static pressure. Then, the cascade theory and parameter design method were utilized to generate three-dimensional blade configuration of the impeller including the rotor that fit in the original housing. After that, prototypes of the impellers as shown in Fig. 6 were fabricated using a CNC machine and were then installed in the original housing of the centrifugal cooling fan to replace the original impellers. Regarding the mockup manufacture, with the input of impeller's geometry, the CNC milling route was programmed using commercial CAM software. The performance and noise of the centrifugal cooling fan were tested in an AMCA 210-99 chamber and a semi-anechoic chamber following the CNS-8753 codes. Furthermore, the effects of the redesigned impellers on the performance enhancement and noise reduction of the centrifugal cooling fan were examined by comparing the experimental results between the redesigned and original impellers. Also, the necessity and feasibility for redesigning the impeller of the base centrifugal cooling fan will be evaluated in this stage. Thereafter, a parametric study on the geometry of fan is performed to enhance the understanding of the influences caused by those variations. The parameters considered here include the blade number, blade geometry and hub diameter.

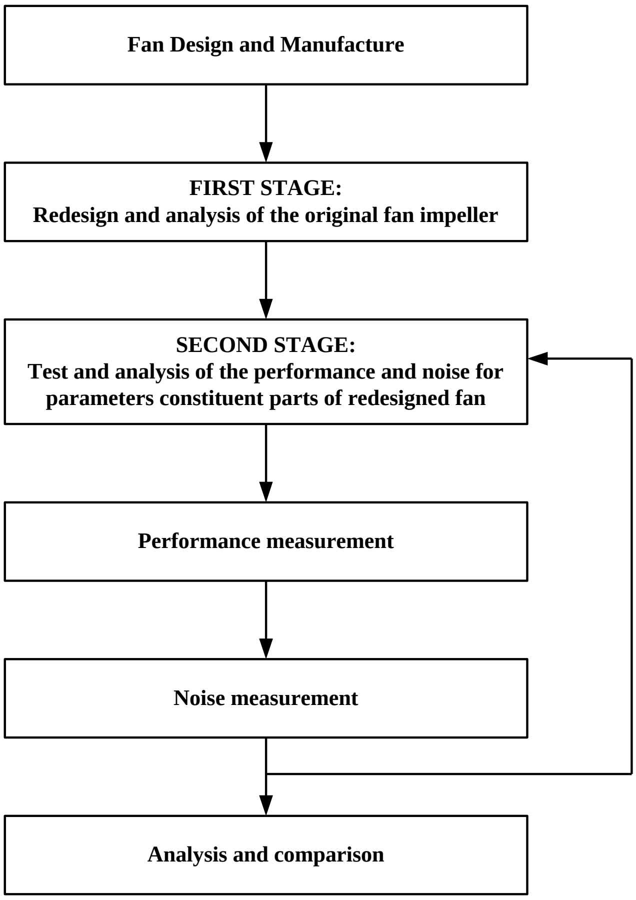
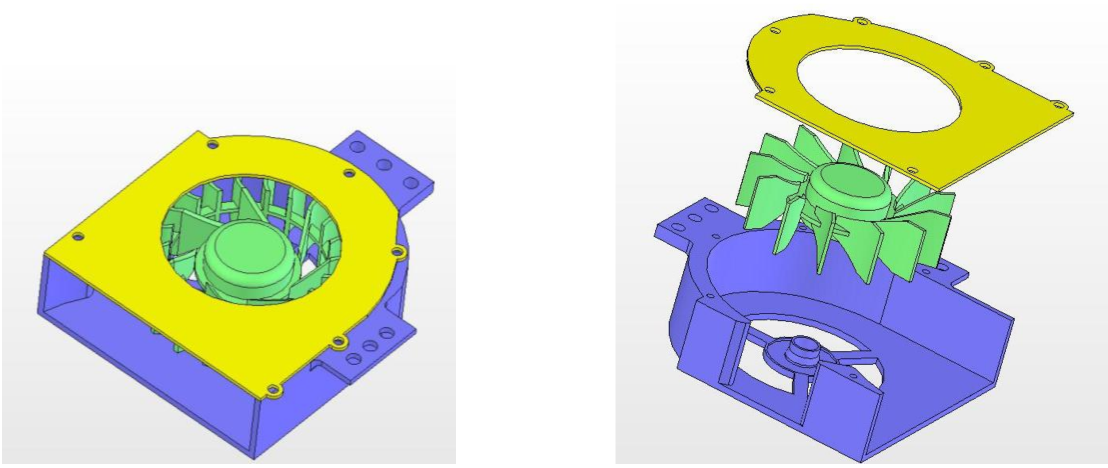


Fig. 5 The flowchart of experimental program



(a) centrifugal cooling fan

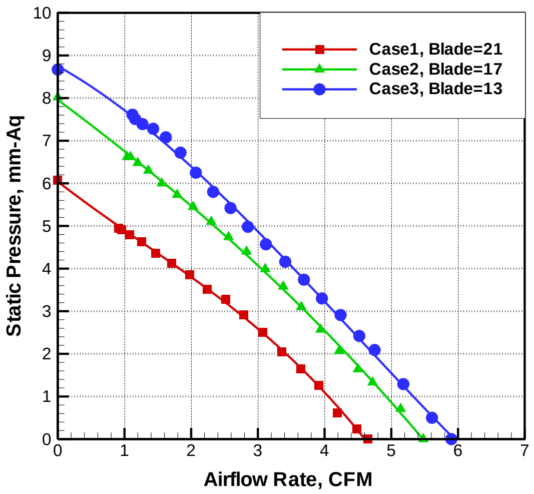
(b) the structures and components of a centrifugal cooling fan

Fig. 6 The prototypes of a forward-curved impeller and a centrifugal housing of the redesign centrifugal cooling fan

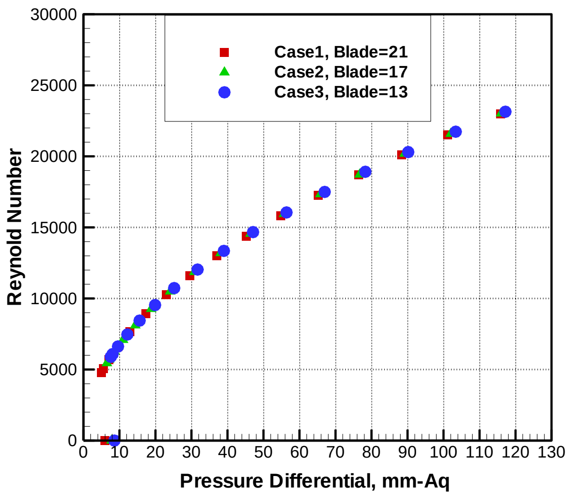
III. Results and Discussions

A. Redesign and Analysis of the Fan Impeller

To investigate if it was necessary and more favorable to enhance the performance and reduce the noise of the centrifugal cooling fan by redesigning the impeller of the base centrifugal cooling fan, we first fixed the housing configuration and only redesigned the impeller of the base centrifugal cooling fan. The experimental results indicated that the centrifugal cooling fan with the redesigned impeller, in comparison with the base centrifugal cooling fan, has increased its maximum flow rate by 17.8% and reduced its noise by 0.5~2.0 dBA, as shown in Fig. 7, Table 1 and Table 2. That is because the more streamlined NACA airfoil profile, which was selected as the base blade shape of the redesigned impeller, has better aerodynamic characteristics. This allows the delay of flow separation from the blade and therefore the reduction of friction between the flow and the blade. Also, the maximum static pressure of the centrifugal cooling fan with the redesigned impeller has increased by 31.9% comparing with the base centrifugal cooling fan, as shown in Fig. 7 and Table 1. Considering the above reasoning and manufacturing cost, a single layout blade configuration was selected for the redesigned impeller in this study. Consequently, the static pressure of the redesigned impeller increased because the single layout blade of the redesigned impeller had a smaller contact area between the blade surface and fluid than the multiple layout blades can to drive the flow and lift the pressure. By means of the above analysis, results related to the impeller of the base centrifugal cooling fan revealed that there is still room for improvement in terms of performance enhancement and noise reduction of the impeller in the base centrifugal cooling fan.



(a) the relation of the airflow rate and the static pressure



(b) the relation of the pressure differential and the Reynold number

Fig. 7 The performance curves for different case of centrifugal cooling fan

Table 1. The performance comparison for different case of centrifugal cooling fan

Items	Case1 (Base)	Case2	Case3
Max. Airflow Rate Q (CFM)	4.65	5.48	5.90
Percentage (%)	—	17.80 %	26.88 %
Max. Static Pressure P_s (mm-Aq)	6.08	8.02	8.67
Percentage (%)	—	31.90 %	42.59 %

Table 2. The noise comparison for different case of centrifugal cooling fan

Items	Noise (dBA)	Position1	Position2	Position3
Case1 (Base)	Overall Value	42.65	37.62	37.32
	Max. Value	25.95	23.78	26.44
Case2	Overall Value	41.33	36.52	35.89
	Max. Value	24.56	23.26	24.78
Case3	Overall Value	46.51	42.21	41.86
	Max. Value	29.46	26.77	28.14

B. Vary the Blade Numbers of Rotor

In this section, the rotor with a 44° stagger angle, which had demonstrated a more superior performance and noise characteristics in the previous experimental results, was used as the base rotor to combine with different number of blades. A series of experiments to investigate the influence of varying rotor blade numbers on the performance and noise of centrifugal cooling fan were performed. Figure 7 indicates that the maximum flow rate of the centrifugal cooling fan, the decreasing rotor blade number from 21 to 13. The experimental results indicated that the centrifugal cooling fan with the vary the blade numbers of rotor, in comparison with the base centrifugal cooling fan, has increased its maximum flow rate by 26.88% and the maximum static pressure has increased by 42.59%, as shown in Fig. 7 and Table 1. One of the reasons was the absolute tangential velocity of the fluid when leaving the impeller was guided and turned into an axial direction flow when entering the rotor. Apparently, the axial velocity component of the flow increased and caused the volume flow rate in the axial direction to increase accordingly. Other than the reason stated above, the shape of the rotor blade was based upon the NACA airfoil profile. This profile was capable of not only elevating the lift but also reducing the drag and energy loss. Furthermore, the deviation angle is small when the fluid passed through the curved channels formed by two adjoining blades whose cross-section followed the NACA airfoil profile. This also led to the increase of the airflow rate. Besides, the maximum static pressure of the centrifugal cooling fan increased with the rotor blade number. This phenomenon indicated that the rotor indeed achieved the goal of elevating the pressure. The reason is the fluid flowing out of the impeller can pass through the rotor more smoothly due to the decrease of the blade’s number. Moreover, the contact area between the blade and the airflow increased and led to the change of pressure between the upper and lower blade surfaces. Thus, the vortex region in the rotor flow channel can be decreased and lead to the raise of static pressure.

IV. Conclusions

This study has improved the performance and noise characteristics of a centrifugal cooling fan. First, the impeller was redesigned to fit the original housing. Then, prototypes were fabricated using a CNC machine and tested for further modification. By performing a parametric study on the geometry, a comprehensive assessment of their influences on the performance and noise of centrifugal cooling fan was obtained. The parameters considered in this work include the blade number, blade geometry and hub diameter. Experimental results showed that the redesigned impeller, in comparison with the original design, resulted in a 17.8% increase in maximum volume flow rate and a 21.9% increase in maximum static pressure, and the vary the blade numbers of rotor, in comparison with the base centrifugal cooling fan, has increased its maximum flow rate by 26.88% and the maximum static pressure has increased by 42.59%. Regarding noise improvement, the proper selection of impeller shape reduced noise output by 0.5~2.0 dBA.

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