Design for New Type Centrifugal Fan and Passageway of the Air Conditioner Indoor Unit

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Abstract. This paper introduces a design flow for the low-speed high-efficient centrifugal impeller applicable to the new type indoor hanging unit, with the matching inlet collector and non-uniform outlet diffusion passageway being also designed. Under the influence of the inlet collector, the attack angle deviation and air intake distortion occur to the centrifugal impeller. While the unsteady pressure fluctuation generated by the non-uniform diffusion passageway can also affect the flow in the passage of the upstream blades. The preliminary design and optimization of the above three parts are described in this paper based on the needs of overall design, the outlet flow rate is also increased by adjusting the static pressure distribution in the inside centrifugal impeller, consequently the flow separation generated by the guide blade in the diffusion passage is improved and the outlet uniformity of the diffuser is enhanced. Finally, the feasibility analysis for the overall design is performed in this paper, to ensure the design requirements can be met.

Keywords: Centrifugal impeller, air conditioner indoor unit, non-uniform, optimization.

1. Introduction
At present, the form of cross flow fan + bottom air supply is adopted for most of the wall mounted air conditioners [1], as shown in Figure 1. The cross flow fan, currently being widely used by indoor hanging units, works with a long barrel-shaped fan impeller. The airflow runs through the impeller and driven twice by the force of the blade, therefore, it can reach a far distance without turbulence [2-5]. Meanwhile, the diameter of the barrel-shaped blade is large enough to ensure the overall air circulation volume under low rotating speed, in order to reduce the noise from high-speed running [6]. However, the airflow is forced to turn back in the impeller, as a result the loss of pressure head is big and the efficiency is low [7]. When the impeller rolls, the airflow enters the blade grid through the open location of the impeller and will be discharged into the volute casing from the blade grid at the other end, forming a work airflow, therefore, the airflow speed and pressure are not uniform at the outlet of the fan, and this is also one of the shortages of the indoor hanging units [8].
Though the cross flow fan has the above mentioned advantages, there are still some problems for the indoor hanging units provided with the cross flow fan, which include: The air supply location is centralized and the air supply is not uniform [9]. Due to the form of bottom air supply, the outlet of the indoor unit is at the bottom and this to some extent affects the user experience of the indoor hanging unit. Therefore, the author cooperates with a home appliances brand and tries to propose a new form for the air conditioner indoor unit, to change the traditional form of top air intake and bottom air supply into middle air intake and air supply from three sides, in order to enhance the cooling/heating effect of the air conditioner in the space. The sectional view of the new type air conditioner indoor unit is shown in Figure 2. There is an axial air intake passage and air supply by the centrifugal fan from four sides can be realized after heat exchange in the heat exchanger.

2. Preliminary design of main components

2.1. Selection of the centrifugal fan design scheme

Figure 3 shows the internal structure diagram of the new indoor hanging unit and Table 1 shows the main physical dimensions preliminarily proposed for it.
Figure 3. Internal structure diagram of the new air conditioner indoor unit

Table 1. Internal main dimensions of the new air conditioner indoor unit

| Parts                      | Dimension (Unit: mm) |
|---------------------------|----------------------|
| Heat exchanger            | 400*800*20           |
| Overall unit              | 850*575*260          |
| Outer diameter of the passageway | 400                 |

According to the given dimensions, firstly it needs to determine whether the combination of fan impeller + diffusion passageway with different design parameters can both meet the conditions of pressure rise and flow rate and match the dimension of the heat exchanger, i.e., the design parameters of the impeller shall be determined firstly to make the combination of impeller + diffusion volute casing meet the pressure rise of more than 30Pa and the flow rate of 320m³/h, and not exceed the dimension limit of 400mm*400mm (one side) for the heat exchanger.

Thus, a series of schemes has been designed for the impeller based on the conditions of pressure rise and flow rate, and the design parameters meeting these conditions are selected. The effect of the blade outlet angle $\beta_b2$ has been considered in advance in the scheme design and the best blade outlet angle $\beta_b2 = 40^\circ$ is selected. The impeller design scheme is shown in the table 2 below.

Table 2. Comparison of Design Schemes for Centrifugal Impeller

| Rotating speed (rpm) | Effective power (w) | Total pressure rise (Pa) |
|----------------------|---------------------|--------------------------|
| 30                   | 2.7                 | 30                       |
| 500                  | 310                 | 320                      | 350                     |
| 750                  | 120                 | 125                      | 140                     |
| 950                  | 210                 | 220                      | 230                     |
|                      | 8.1                 | 8.2                      | 8.5                     |
|                      | 8.1                 | 8.6                      | 9.1                     |
|                      | 70                  | 80                       | 80                       |
|                      | 160                 | 170                      | 180                     |
|                      | 8.1                 | 8.5                      | 9.1                     |
|                      | 65                  | 70                       | 75                       |

As shown in Figure 4, when the impeller is fixed at the center, the position limiting the dimension of the diffusion passageway exists at the horizontal direction, i.e., whether the distance between the impeller outer diameter and the right side edge can meet the design dimension requirement of the volute
casing. Therefore, all the openings of the volute casing in the above table refer to the width of the volute casing at the horizontal direction as shown in the figure.

![Figure 4. Impeller and Volute Casing Diffusion Passageway](image)

As shown in Table 2, with the increase of the rotating speed and design pressure rise, the outer diameter of the impeller and the opening of volute casing increase, the outer diameter increases to 350mm from 160mm within the range of 500rpm - 960rpm and 30Pa - 40Pa, and the overall dimension increases to 490mm from 225mm after the volute casing passageway is added. Given the pressure loss after adding the inlet collector and outlet passageway, the 750 rpm and 40Pa are selected as the best design parameters for the impeller. With these parameters, the dimension of the combination impeller + diffusion passageway can meet the overall dimension requirement and will not cause a too small impeller not matching the guide blade. At the same time, in order to improve the capacity for work, the impeller blade outlet angle is increased to 43° and the pressure rise coefficient is increased to 0.415 under the premise of ensuring the outer diameter not to be changed.

2.2. Preliminary Design of the Fan Impeller
According to the selection of the design scheme, the preliminary pneumatic design for the impeller is performed and see Annex 1 for the design process. The main parameters of the impeller are obtained, as shown in Table 3.1.

| Blade height | Impeller diameter | Total pressure rise | Number of blades |
|--------------|-------------------|---------------------|------------------|
| Inlet        | 70 mm             | 175 mm              | 40 Pa            | 18               |
| Outlet       | 57 mm             | 230 mm              |                  |                  |

2.3. Preliminary Design of the Collector
The inlet collector is placed at the rear of the heat exchanger and in front of the fan inlet, and the inlet air flowing through the heat exchanger needs to be guided to the fan inlet on the premise of minimizing the loss. The dimension of the collector shall be determined based on the preliminary design result of the fan and the contraction drainage effect of the collector can make the airflow generate a big attack angle deviation during entering the fan, therefore, the design and optimization processes of the collector and fan are carried out simultaneously.

The axial dimensions of the taper arc collector which has a higher overall efficiency are shown in Figure 5. The distance between the inlet and the fan hub is 120mm, with the effect of this dimension, the angle of throat at the taper section of the collector inlet is 70°, being much larger than the design experience value (30° - 40°).
As the angle of throat of the collector is larger, the inlet airflow easily separates in the throat, so it needs to control the radius of the throat, inhibit flow separation and smoothly guide the airflow to the blade front edge, avoiding the larger attack angle deviation generated at the blade tip. The molded line of the throat ring shall be connected smoothly with the conical surface in front and the impeller shroud at the rear.

As shown in Figure 6, the impeller has a molded line of meridian plane, the mean radius of the impeller blades $R_{1m} = 85\text{mm}$, then the largest radius of the blade tip is:

$$ R_{1\text{max}} = 1.03\times D_{1m} = 88\text{mm} $$

In order to make the airflow be guided to the radial direction by the axial direction after turning the throat, a distance of 5 - 10mm shall be reserved at the minimum radius between the blade tip and the throat, as shown in Figure 7.
To reduce the flow separation in the throat, 6mm is selected as the distance between the throat radius and the maximum radius of the blade tip after many times of value calculation and comparison. The molded line of the throat is geometrically and smoothly connected with its front and back parts, and the inlet collector shown in Figure 8 is obtained.

![Figure 8. Inlet Collector](image)

Its main parameters are shown in Table 4.

| Outer edge | Throat diameter | Angle of throat | Arc diameter of the throat | Distance between the contraction section and the blade tip |
|------------|-----------------|-----------------|---------------------------|---------------------------------------------------------|
| 400*400 mm | 167 mm          | 70°             | 10 mm                     | 6 mm                                                   |

2.4. Preliminary Design of the Outlet Passageway

The outlet passageway is divided into 2 sections: The volute casing section and the outlet section, as shown in Figure 9.

The volute casing is designed based on the is volumetric dilatation and equal circulation dilatation, its molded line is an Archimedes evolvent and the opening of different cross sections $\varphi$ are:

$$A_{\varphi} = \frac{Q}{2\pi B_{2u}} \cdot \varphi$$  \hspace{1cm} (1)
As the specific speed of the impeller is low, the equal sides primitive method is adopted for drawing. The distance between the volute throat gap and the impeller outlet is selected as 20mm. too large distance can reduce the efficiency, while too small one can cause high noise. The outlet passageway as shown in Figure 10 is obtained after many times of calculations of the two volute throat gaps and outlet location. Where, the number and location of the outlet guide blade needs to be designed according to the further improving result of the impeller.

3. Problems in the Preliminary Design and Optimization

3.1. Adjustment of the air intake angle of the impeller
The attack angle deviation at the blade tip is large when the airflow reaches the front edge of the blade due to the effect of the inlet collector. Therefore, the impeller and the inlet collector are calculated
unitedly, and the blade is divided into root, middle and tip cross sections, to obtain the air intake attack angle at each cross section when the airflow reaches the front edge of the blade through the throat.

Figure 11 and 12 show the vector diagram and flow line diagram of the relative speed at 95% blade height. It can be seen from the figures the deviation of the air intake attack angle is large and there is a big separation in the blade passageway.

![Figure 11. Vector Diagram of Relative Speed at 95% Blade Height](image1)

![Figure 12. Flow Line Diagram at 95% Blade Height](image2)

According to the result of united calculation, the air intake angle is adjusted gradually, and Figure 13 and 14 show the vector diagram and flow line diagram of the relative speed at 95%, 50% and 5% blade height. It can be seen from the figures the deviation of the air intake attack angle is small, the separation is eliminated and the design point efficiency is increased to 83% from 76%.

![Figure 13. Vector Diagram of Relative Speed at Different Blade Heights](image3)

(a) 95% blade height  
(b) 50% blade height  
(c) 5% blade height
3.2. Reduction of static pressure rise in the impeller and increase of speed at the impeller outlet

Table 5 shows the proportion of each pressure rise in the parts. It can be seen from Table 5, the static pressure inside the impeller is too large and it causes the outlet speed is small, separation and backflow being easily generated in the outlet passageway.

|                     | Overall | Inside of collector | Inside of impeller |
|---------------------|---------|---------------------|--------------------|
| Static pressure rise| 27 Pa   | -9 Pa               | 36 Pa              |
| Total pressure rise | 36 Pa   | -2 Pa               | 38 Pa              |

Figure 15 shows the speed vector diagram at the passageway plane corresponding to 95%, 50% and 5% blade height.
It can be seen from the figure that the flow rate at the passageway plane corresponding to the impeller root is high, though there is a flow separation caused by the poor distribution of the guide blade, and it can be improved by adjusting the guide blade. However, the overall flow rate at the passageway plane corresponding to the blade tip is low with large-area backflow and separation.

It can be concluded that the speed at the impeller outlet is uneven in the axial direction, the speed near the blade tip is lower and that at the blade root is higher. There is a separation and backflow zone of large area as shown in Figure 15(a), and the reason for this is preliminarily judged as that there has been a separation zone when the blade tip is at the impeller outlet before it enters the passageway.

Furthermore, the figure 16 shows the speed vector diagram at the meridian plane of the impeller blade. It can be seen in the Figure 16, there is a large-area low speed zone for the blade at the trailing edge of the blade tip. This low speed zone further speeds down after entering the diffusion passageway and gradually form the low-speed backflow zone of a large area in the flow field as shown in Figure 15(a). Therefore, the main direction for improving the flow field of the outlet passageway is to inhibit the low-speed backflow zone at the trailing edge of the blade tip.
There are mainly two possible reasons for the existing of a large-area separation of the flow field at the impeller outlet:

1. The impeller blade has an attack angle deviation, causing serious separation at the impeller outlet. The power angle characteristic of the blade has been analysed in the previous work, and the low-speed separation zone at the trailing edge of the blade tip is not caused by attack angle deviation.

2. The area dilatation ratio in the blade and passageway is too large, and it can be concluded through the Bernoulli equation and conservation of mass that it will cause the problem of too low speed and too large static pressure rise. Therefore, it needs to analyse the cross-sectional area change in the blade passage.

Inlet area of the blade passage at the impeller inlet (complete cycle):

$$A_1 = \pi D_1 b_1 \sin \beta_1 = 10925 \text{ mm}^2$$  \hspace{1cm} (2)

Outlet area of the blade passage at the impeller outlet (complete cycle):

$$A_2 = \pi D_2 b_2 \sin \beta_2 = 28074 \text{ mm}^2$$  \hspace{1cm} (3)

The inlet and outlet area ratio of the blade passage is 2.6, the equivalent dilatation angle of the blade passage is 13.4°, much greater than the design experience value 2° - 5°, so the outlet speed is low and the static pressure rise is large.

While the main reason for the too small inlet area of the blade passage at the impeller inlet is that the blade inlet angle $\beta_1$ is too small and the mean from the blade tip to the root is only 16°. But as being limited by the contraction effect of the collector, $\beta_1$ only the smaller angle can be selected to make the blade front edge be consistent with the direction of the incoming flow and the attack angle deviation be smaller. If the parameters such as $b_2$ and $\beta_2$ are changed, the entire design will need to be adjusted, therefore, the impeller outlet height $b_2$ is adjusted to reduce the area dilatation ratio and achieve the purpose of reducing the static pressure rise and increasing the outlet outlet flow rate.

The outlet blade height $b_2$ is reduced to 51 mm from 57 mm within the value range of design parameter. Other parameters are not changed and the inlet and the outlet area ratio of the blade passage is reduced to 2.1. The molded line of the impeller shroud is adjusted, the speed vector diagram at the meridian passage of the impeller blade as shown in Figure 17 is obtained and it is compared with that before modification.

![Figure 17. Comparison between the Speed Vector Diagrams at the Meridian Plane of the Impeller Blade before and after Modification](image-url)
It can be found that the low-speed zone at the blade tip basically disappears, the mean outlet flow rate increases to 7.8 m/s from 6.5 m/s.

3.3. Optimal Design of the Outlet Guide Blade
The design of the outlet guide blade mainly includes the following three aspects:

① Number and mounting location of the guide blades
The number and mounting location of the guide blades shall be determined firstly. To facilitate the processing, isometric mounting is adopted for the guide blades of the two outlets. The number of guide blades shall be as less as possible to reduce losses. The preliminary design is two guide blades on each side, but in order to reduce the flow separation of a single blade, the number of the guide blades is gradually increased to achieve the effect of increasing the consistency and reducing separation. Finally, the number of guide blades is determined as 5 on each side.

② Chord length of the guide blade
The chord length of the guide blade is controlled to be equidistant to the chord length of the impeller blade. In order to reduce losses, the thickness is controlled to be 1.5mm-2mm and an arc chamfering is made for the front section.

③ Air intake angle of the guide blade
After such parameters as number, location and length of the guide blades have been determined, the location of the front end of the guide blade in the flow field can be basically determined. Observation points are set in the flow field to determine the air intake angle of the front end of the guide blade and make the guide blade nearly has no inlet attack angle. The guide blade shall be perpendicular to the outlet plane at the outlet.

After many times of adjustments and modifications, the guide blade configuration as shown in Figure 18 is obtained.

![Figure 18. Guide Blade Configuration](image)

Numerical simulation is performed for the inlet collector, fan impeller and outlet passageway. Figure 19 shows the speed vector diagram at different passageway planes corresponding to blade heights.
Figure 19. Speed Vector Diagram at Different Passageway Planes Corresponding To Blade Heights

It can be seen from the figure that there is almost no flow separation and backflow in the flow field above the zone of 50% blade height and the outlet flow field is even. But backflow and flow separation gradually appear in the zone below 50% blade height. At the 95% blade height, though the flow field characteristics are improved compared to that before optimization, there is an obvious flow separation and low-speed backflow zone. This is partially caused by the inherent characteristic of the centrifugal impeller, but the main reason is the low speed at the trailing edge of the blade tip induces separation in the passageway with a low-speed backflow zone being formed.

Figure 20 and 21 respectively show the speed and total pressure distribution at the lower and side outlets, it can be seen from the figures that the speed and total pressure distribution at both outlets are uneven.
Figure 20. Cloud Chart of Speed Distribution at Two Outlets

(a) Speed distribution at the lower outlet

(b) Speed distribution at the side outlet

Figure 21. Cloud Chart of Total Pressure Distribution at Two Outlets

(a) Total pressure distribution at the lower outlet

(b) Total pressure distribution at the side outlet
4. Overall Feasibility Analysis and Conclusions

This paper provides a design flow for the centrifugal fan applicable to the new type indoor hanging unit, with its performance being optimized. Based on this, the matching inlet collector and non-uniform outlet diffusion passageway are designed to obtain better heat exchange efficiency and outlet uniformity. The main conclusions and analysis of the design feasibility are as follows.

1. The inlet collector can greatly affect the performance of the centrifugal impeller. The main reason is that it affects the inlet attack angle of the impeller, causes an attack angle deviation at the inlet of the blade passage and then a big flow separation is formed in the whole blade passage. As a result, the overall performance matching is needed in the design process of the centrifugal impeller, i.e., the geometric effect of the inlet shall be considered.

2. The centrifugal impeller can have an uneven flow rate along the radial outlet due to the limit of its operating principle. The uneven outlet flow rate is further enlarged in the outlet diffusion passageway and can affect the outlet uniformity. Specifically, the low flow rate at the blade tip of the centrifugal impeller can cause a large amount of flow separation at the corresponding plane in the diffusion passageway. Reducing the static pressure distribution in the blade passage and increasing the flow rate at the blade tip outlet can effectively improve the flow separation in the diffusion passageway and enhance the outlet uniformity of the diffusion passageway.

3. The combining method of diffusion with blades + diffusion without blades is adopted for the non-uniform diffusion passageway at the outlet, i.e., volute casing diffusion + guide blade passage. The key factor to improve the outlet uniformity of the diffusion passage is the inlet angle of the guide blade, that is to reduce the flow separation generated due to the existence of the diffusion guide blade. Meanwhile, increasing the outlet flow rate of the centrifugal impeller within the range of design requirements can also improve the flow state in the diffusion passageway.

4. The analysis of design feasibility is as follows:

The overall performance of the centrifugal fan is shown in Table 6.

Table 6. Overall Performance Parameters

|        | Flow rate (m³/h) | Air speed (m/s) | Total pressure rise (Pa) | Fan efficiency | Fan power (w) | Fan pressure rise (Pa) |
|--------|------------------|-----------------|--------------------------|----------------|---------------|------------------------|
| Side edge | 120              | 3               | 22                       | 86.4%          | 4             | 38                     |
| Bottom edge | 200              | 4               | 26                       |                |               |                        |

① Flow rate

The flow rate is required to be 300m³/h, and the design flow rate is 320m³/h with the inlet clearance leakage being taken into consideration to meet the flow rate requirement.

② Air speed

The requirement for air speed is that the breeze can be felt respectively in 5 meters and 3 meters away from the lower and side outlets. After verification, the speed of breeze is defined as 0.5m/s, and the free jet empirical formula is used:

\[ \frac{U_m}{U_0} = 0.95 \left( \frac{L_m}{D_0} \right)^{-0.49} \]  (4)

Where, \( U_m \) is the flow rate at the location \( L_m \) away from the free jet origin, \( U_0 \) is the flow rate at the free jet origin, \( D_0 \) is the hydraulic diameter at the free jet origin and all of them can be set as 0.1m and the following values can be obtained:

\[ U_{0 \text{, side}} = 2.78 \text{ m/s} \]  (5)
Then the air speed at the side and lower outlet can meet the requirement under the required flow rate.

Power

The power of the configured motor is 10w, the calculated power for the fan is 4 w and there is some allowance for the power after the efficiency and all aspects of the loss being considered.

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