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Design and Simulation of Air Compressor Impeller for Fuel Cell Vehicles

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***Abstract* – Centrifugal air compressors have advantages in terms of performance, efficiency, cost, size, and weight, and can be matched with expanders to recover energy, making them the most ideal choice for fuel cell vehicle compressors. This article provides a design method for compressor impellers, including the design process, key parameters, calculation formulas, and coefficient selection principles. This method is used to design an air compressor impeller, and the feasibility of the design method is verified through simulation using fluent. The simulation results show that the total pressure ratio of the designed impeller under design conditions can reach 2.6, the isentropic efficiency is 92.3%, the flow velocity inside the impeller is small, and mechanical energy can be converted into more pressure energy. The design method is feasible.**

***Keywords* – Fuel Cell Vehicles, Centrifugal Air Compressor, Impeller Design, Flow Field Simulation.**

I. Introduction

Nowadays, new energy vehicles have been widely used globally. In terms of environmental protection and resource conservation, fuel cell vehicles have truly achieved zero emissions and no pollutants in their exhaust gases [1]. Air compressors are important components in fuel cell systems, so designing centrifugal air compressors with high pressure ratio and high efficiency is of great significance for improving the efficiency, power density, and cost reduction of fuel cell systems [2].

Reference [3] provides a new method for defining blade profiles, which utilizes the Bernstein polynomial to define the shape of the impeller flow channel, and the curve determined by this polynomial is called the Bezier curve. Reference [4] analyzed the working characteristics of automotive fuel cells, studied the interaction between centrifugal compressors and fuel cell systems, provided design boundaries and requirements for centrifugal compressor design, and improved traditional empirical design methods for centrifugal compressors. Reference [5] used the Kriging model to design the impeller of a fuel cell air compressor under multiple operating conditions. Zhu Shican proposed a scalable surface swept back impeller centrifugal blade modeling method, which achieves the overall shape of the impeller through the extension of the blade surface. After calculation and verification by the quasi orthogonal surface method, this modeling method can be used in the design and machining of centrifugal compressor impellers [6].

This article conducts one-dimensional thermal design and three-dimensional structural design on the impeller of a centrifugal compressor for a fuel cell vehicle, and uses CFD finite element simulation to obtain the total pressure ratio and isentropic efficiency of the impeller, verifying the correctness of the design method and effectively reducing the design cycle.

II. Impeller Design

2.1. *Impeller Design Process*

The impeller is the most important part in a centrifugal air compressor, which can convert the mechanical energy transmitted by the motor into the kinetic energy and pressure energy of the gas. The quality of its structure is related to the efficiency and pressure ratio of the air compressor. The impeller design is divided into one-dimensional thermal design and three-dimensional structural design, as shown in Figure 1 for the design process of centrifugal impellers.

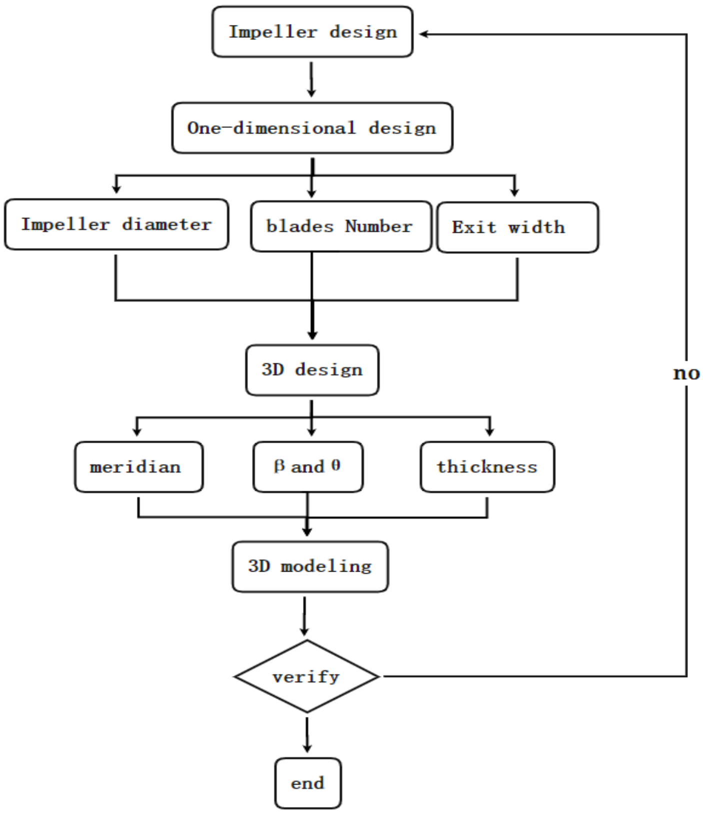

Fig. 1. Design process of air compressor impeller.

One dimensional design first calculates the inlet and outlet diameter and hub diameter of the impeller based on the design speed, then calculates the number of blades based on the outlet angle and inlet and outlet diameter of the arc in the impeller, and finally calculates the outlet width of the impeller based on the design flow rate, etc.

The three-dimensional design first uses the Bezier curve to design the meridian channel of the impeller, determines the approximate shape of the impeller channel, then designs the three-dimensional distribution of the installation angle and envelope angle of the impeller, and finally uses variable thickness blades to design the thickness distribution curve of the impeller blades.

After completing the one-dimensional and three-dimensional design of the impeller, conduct impeller modeling, extract the single channel watershed of the impeller, and perform fluid simulation verification.

2.2. *One Imensional Design Calculation of Impeller*

2.2.1. *Design Indicators*

On vehicles, the gas supply system pressure required for fuel cell systems is generally 1 to 4 atm [4].

The design parameters of the impeller are shown in Table 1.

Table 1. Impeller Parameter Design Indicators.

Parameter	Numerical Value
Speed/(r·min^{-1})	100000
Rated Flow Rate/(kg·s^{-1})	0.09
Pressure Ratio	2.5
Total Inlet Pressure/pa	101325
Inlet Temperature/K	293

The working medium is ideal air. The rated flow at the design working point is 0.09 kg/s, the rated pressure is 250 kpa, the total temperature at the inlet condition is 293 K, the total pressure is 1 standard atmospheric pressure, and the specific heat capacity of the air is 1.005 kJ/(kg · ℃).

2.2.2. *Impeller Inlet and Outlet Diameter*

$$D_2 = \frac{60u_2}{\pi n} = \frac{60\sqrt{\frac{h_{pol}}{\psi_{pol}}}}{\pi n} = \frac{60\sqrt{\frac{\frac{k}{k-1}\eta_{pol}RT_{in}(\varepsilon^{\frac{k}{k-1}\eta_{pol}}-1)}{\psi_{pol}}}}{\pi n} \tag{1}$$

In the equation, u_2 is the circumferential speed of the impeller, n is the design speed of the impeller, and h_{pol} is a variable energy head, ψ_{pol} is the coefficient of polytropic energy head, k is the heat capacity ratio of air, usually 1.4, η_{pol} is the polytropic efficiency, R is the air gas constant, usually 288, T_{in} is the inlet temperature of the impeller, ε Design the pressure ratio for the impeller.

The impeller type adopts a semi open impeller, which has a higher circumferential speed and a higher pressure ratio. The designed impeller is a backward impeller, and the variable efficiency of the backward impeller is generally 0.76~0.84. In this article, 0.8 is taken. From this, it can be concluded that the variable energy head of the impeller is 91429.5 J/kg.

Take the installation angle of the impeller outlet β_{b2} and entrance installation angle β_{b1} is 40 ° and 55 °, respectively. Based on experience, different β the variable energy head coefficient of b2 value is taken as follows:

When β_{b2}=15 °~30 °, ψ_{pol} =0.35~0.52

When β_{b2}=30 °~60 °, ψ_{pol} =0.40~0.65

When β_{b2}=90 °, ψ_{pol} =0.65~0.75

According to β_{b2} preselected variable energy head coefficient, selecting ψ_{pol} = 0.6, the circumferential velocity of the outer diameter of the impeller can be obtained as 390 m/s.

Overall, it can be concluded that the outer diameter of the outlet impeller is 74 mm. If the diameter ratio is chosen as 0.48, then the diameter of the inlet impeller is 36 mm. The diameter of the wheel hub is 16 mm.

2.2.3. *Number of Blades*

For the number of blades, the following equation can be used to estimate:

$$Z \approx 8.5 \times \frac{\sin (90-\beta_{b2})}{1-\frac{D_1}{D_2}} \approx 13 \tag{2}$$

In the equation β_{b2} is the outlet installation angle, D_1 is the inlet diameter, and D_2 is the outlet diameter.

2.2.4. *Impeller Outlet Width*

The outlet width of the impeller can be obtained from the outlet flow formula of the impeller [7]:

$$b_2 = \frac{Q_{in}}{\pi D_2 \varphi_{2r} u_2 \tau_2 k_{v2}} \tag{3}$$

In the formula, Q_{in} represents the inlet volume flow rate, and D_2 represents the outer diameter of the impeller, φ_{2r} is the flow coefficient, u_2 is the circumferential velocity of the impeller, τ_2 is the outlet blockage coefficient, and k_{v2} is the ratio of specific volume at the inlet and outlet of the impeller.

The inlet temperature of the impeller is 19.85 ° C, and the density of air at this temperature ρ Approximately 1.205 kg/m3, from which the inlet volume flow rate can be obtained as,

$$Q_{in} = Q_0 \times \rho = 0.10845 m^3/s \tag{4}$$

Although an increase in the flow coefficient will increase the average velocity of gas in the blade passage to contain gas backflow, it is not conducive to improving the efficiency of the impeller, and the appropriate value is 0.18~0.32 [8]. Figure 2 shows the relationship between flow coefficient of vaneless diffuser impeller and impeller outlet angle [9]:

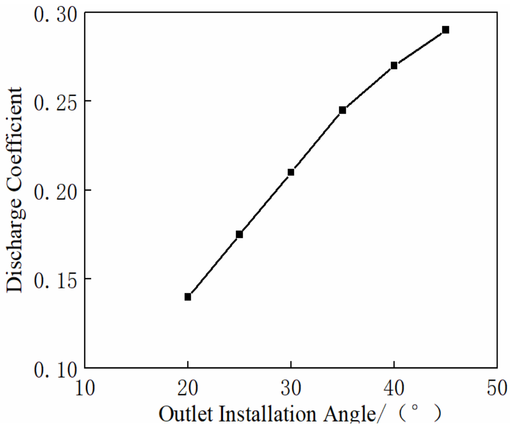

Fig. 2. Relation between discharge coefficient of vaneless diffuser impeller and outlet angle of impeller.

According to the selected outlet installation angle, the flow coefficient is 0.27. k_{v2} is the ratio of specific volume at the inlet and outlet of the impeller, and its calculation formula is:

$$k_{v2} = \frac{v_1}{v_2} = \frac{T_2}{T_{in}}^{\frac{1}{m-1}} \tag{5}$$

In the formula, v_1 is the inlet specific volume, v_2 is the outlet specific volume, T_2 is the outlet temperature, and m is the polytropic index.

Based on the traditional compressor design experience, the absolute Mach number at the compressor inlet is M_{c1} = 0.215 [10], then the inlet air stagnation temperature can be calculated from the Mach number M_{c1} of the inlet air and the air heat capacity ratio as follows:

$$T_{1st} = T_{in}\left(1 + \frac{k-1}{2}M_{c1}^2\right) = 295.708\text{K} \quad (6)$$

By pressure ratio ε, the inlet stagnation temperature T_{1st} and the heat capacity ratio k can calculate the impeller outlet temperature as follows:

$$T_2 = T_{1st}\varepsilon^{\frac{k-1}{k}} = 331.931K \quad (7)$$

According to the following formula,

$$\frac{m}{m-1} = \frac{k}{k-1} \times \eta_{pol} \quad (8)$$

The polytropic index m obtained is 1.56. From this, it can be concluded that the ratio of specific volume at the inlet and outlet of the impeller is 1.24.

The formula for the blockage coefficient at the impeller outlet is:

$$\tau_2 = 1 - \frac{\frac{\delta_m}{\sin(90-\beta_{b2})}}{\frac{\pi D_2}{Z}} \approx 0.854 \quad (9)$$

In the equation δ_m is the maximum thickness of the blade, taken as 1.25 mm.

Based on the above, the impeller outlet width can be obtained is 4mm.

2.2.5 *Preliminary Structural Arameters of Impeller*

Table 2. Impeller Structural Parameters.

Parameter	Numerical Value
Impeller Circumferential Velocity/($m \cdot s^{-1}$)	390
Impeller Outlet Diameter/mm	74
Impeller Inlet Diameter/mm	36
Hub Diameter/mm	16
Exit Width /mm	4
Number of Blades	13
Outlet Installation Angle /°	40
Inlet Installation Angle /°	55

2.3. *Three Dimensional Design Calculation of Impeller*

In this paper, the impeller design software Blade Gen is used to carry out the three-dimensional design and modeling of the impeller. The initial three-dimensional model of the impeller can be obtained by inputting the one-dimensional structural parameters of the impeller calculated in the previous chapter into Blade Gen. This software has four main views, namely, the meridian view of the impeller, the auxiliary view, the view of blade installation angle and wrap angle, and the view of blade thickness distribution.

2.3.1. *Impeller Meridian Design*

The control method of the shape of the impeller meridian surface is Bézier curve, which is divided into nodes and straight lines. For example, the wheel housing alignment is controlled by five nodes and two straight lines. The meridian surface of the impeller reflects the inlet diameter, outlet diameter, hub diameter, and axial height of the impeller. The outline of the hub and shield on the meridian plane determines the approximate shape of the flow channel. Figure 3 shows the meridian view of the impeller in this article。

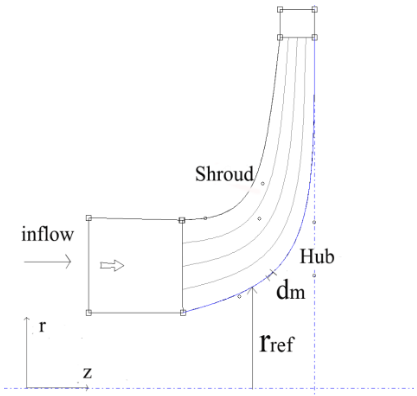

Fig. 3. Meridian view of impeller.

In the figure, r_{ref} is the radius of the wheel hub, and d_m is the differential of the meridian length of the impeller, with a size of,

$$d_m = \sqrt{{d_r}^2 + d_z^2} \tag{10}$$

In the equation, dr is the differential of the radial length of the impeller meridian, and dz is the differential of the axial length of the impeller meridian.

2.3.2. *Impeller Installation Angle and Wrap Angle Design*

The installation angle and wrapping angle of the impeller have the following relationship [11].

$$\beta = \tan^{-1} \frac{r_{ref} d_\theta}{d_m} \tag{11}$$

Therefore, when designing the blade installation angle and wrap angle, only one of them needs to be controlled, and the other can be indirectly controlled. In this paper, the Bezier curve control method is used to control the blade installation angle β, Divide the meridian view of the blades into five layers, mainly controlling the blade angles of the wheel cover and wheel hub layer. Figures 4 and 5 respectively show the installation and wrapping angles of the wheel cover and wheel hub layer blades.

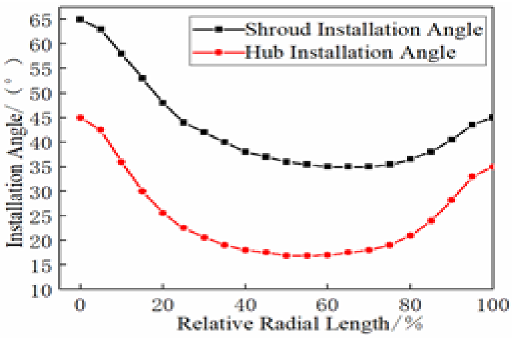

Fig. 4. Blade installation angle.

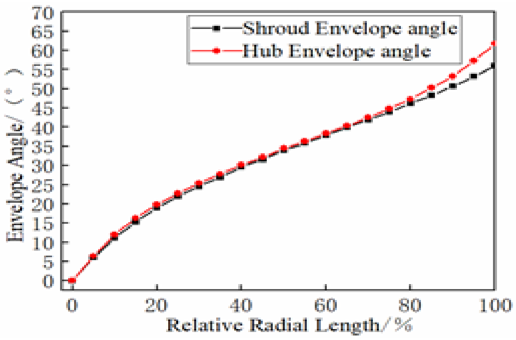

Fig. 5. Blade wrap angle.

2.3.3. *Blade Thickness*

The thicker the blade, the stronger the structural strength of the impeller, but it also correspondingly reduces the flow channel area of the impeller. The impeller in this article adopts a blade thickness that can vary according to the location of the watershed. Taking the thickness of the blade's central arc as an example, excessive blade thickness at the inlet can hinder the flow of air. In order to reduce losses, the inlet should be as small as possible, taking 0.87 mm. After the inlet, the thickness of the blade slowly increases, reaching a maximum value of 1.25 mm at 65% of the axial position of the impeller, and then gradually decreases. The thickness at the outlet of the impeller is 1.1 mm, As shown in Figure 6, the thickness distribution of the blade middle arc is shown.

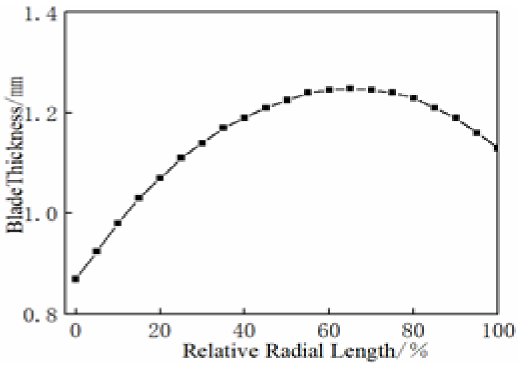

Fig. 6. Thickness distribution diagram of blade middle arc.

After conducting one-dimensional and three-dimensional design, the impeller data can be imported into the modeling software to obtain the three-dimensional modeling of the impeller, as shown in Figure 7.

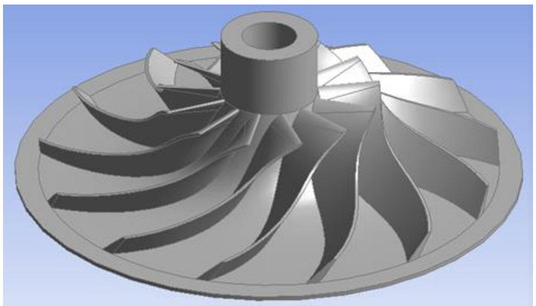

Fig. 7. Modeling of impeller.

III. Computational Simulation

3.1. *Momentum Conservation Equation*

The momentum conservation equation is a motion equation, which is one of the control equations in fluid simulation. It can be explained that the rate of change of fluid momentum over time is equal to the combined force of external volume force, pressure, and viscous force acting on the element. The simplified equation considering fluid viscous force is,

$$\begin{cases} \rho\frac{D_u}{D_t} = -\frac{\partial P_x}{\partial x} + \frac{\partial \tau_{xx}}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} + \rho f_x \\ \rho\frac{D_v}{D_t} = -\frac{\partial P_y}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yy}}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} + \rho f_y \\ \rho\frac{D_w}{D_t} = -\frac{\partial P_z}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + \frac{\partial \tau_{zz}}{\partial z} + \rho f_z \end{cases} \tag{12}$$

In the equation, P is the pressure on the element, τ Is the viscous force, ρ Is the fluid density, f is the volume force of the element per unit mass, and u, v, and w are the components of the velocity vector on x, y, and z.

3.2. *Boundary Conditions and Solvers*

This article uses a single channel impeller model for simulation. When conducting fluid simulation of rotating machinery, fluent provides a calculation method called MRF (Multiple Reference Frame) method. This method directly sets the rotational speed of the dynamic grid part through a rotating coordinate system, and converts the transient model into a steady-state model for solution. The turbulence model adopts the Spalart Allmaras equation. The inlet boundary condition is a pressure inlet, with a total inlet pressure of one atmosphere and an inlet temperature of 293 K. The outlet boundary condition is a mass flow outlet, with a size of 0.0074 kg/s. The wall that rotates with the rotating area is defined as a moving wall, with a movement method of rotation and a speed of 100000 r/min. The solver chooses a pressure velocity coupling method, using a second-order upwind scheme for pressure, density, kinetic energy, and turbulent kinetic energy.

3.3. *Analysis of Calculation Results*

The isentropic efficiency of the air compressor impeller is as follows.

$$\eta_{pol} = \frac{(p_2/p_1)^{\frac{k-1}{k}} - 1}{\frac{T_2}{T_{1st}} - 1} \tag{13}$$

Where p_2 is the total outlet pressure, p_1 is the total inlet pressure, k is the heat capacity ratio of air, T_2 is the outlet temperature, and T_{1st} is the inlet stagnation temperature.

Based on the simulation results, draw the performance curves of the impeller at the design speed as shown in Figures 8 and 9.

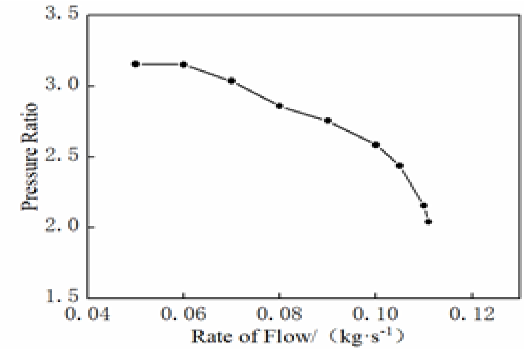

Fig. 8. Pressure ratio flow curve.

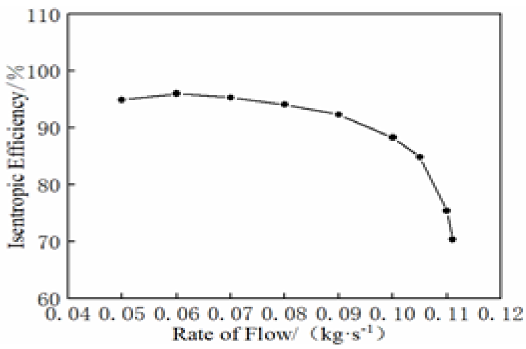

Fig. 9. Efficiency flow curve.

The isentropic efficiency of the designed impeller under design conditions (flow rate of 0.09 kg/s, speed of 100000 r/min) is 92.3%, which is greater than 80% and meets the design requirements. Figure 10 shows the total pressure distribution on the meridian plane. It can be seen from the figure that the total pressure at the impeller outlet under design conditions is 259100 pa, so the calculated pressure ratio is about 2.6, slightly higher than the design pressure ratio of 2.5. Since there will be some loss when the diffuser and volute flow, the design requirements can be met with margin. Figure 11 shows the distribution of Mach number on the meridian surface of the impeller. It can be seen from the figure that the maximum Mach number is 0.6655, and the flow velocity inside the impeller is subsonic, resulting in relatively small losses.

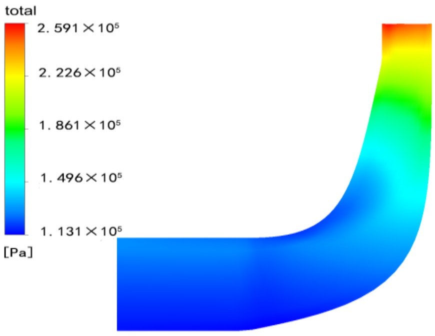

Fig. 10. Meridian total pressure distribution map.

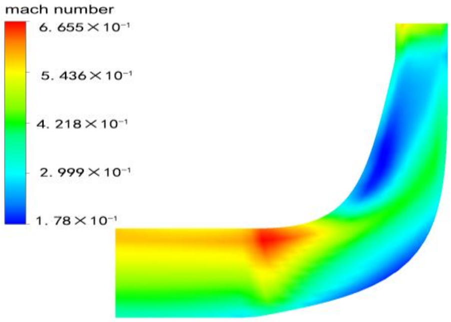

Fig. 11. Mach number distribution on the meridian plane.

IV. Conclusion

A design method for the impeller of a fuel cell centrifugal air compressor was proposed, including the design process, key parameters, calculation formulas, and coefficient selection principles. The impeller of a fuel cell compressor was designed and modeled using this method. By using the MRF method, the transient model was transformed into a steady-state model for finite element simulation, and the aerodynamic performance of the designed impeller was verified. The results showed that the total pressure ratio of the designed impeller under design conditions was 2.6, and the isentropic efficiency was 92.3%, meeting the requirements of fuel cell vehicle air compressors, indicating the feasibility of this design method.

Acknowledgments

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