INTRODUCTION

Nowadays, energy and environmental issues have strongly attracted the attention of researchers in these fields. Various clean energy technologies have been proposed and studied. As one highly efficient and clean energy technology, the polymer electrolyte membrane (PEM) fuel cell has been studied in recent years. The significant applications of PEM fuel cells focus on transportation, distributed/stationary, and portable power generation. And the PEM fuel cell becomes the most desirable option for power generation in the 21st century.

The air supply system is one of the crucial subsystems in PEM fuel cell systems. The air supply system characteristics have an important influence on the performance of PEM fuel cell systems. Control methods and the effect on the fuel cell output of the air supply system were researched. In air supply systems, compressors are the key devices that significantly affect the air supply system and the characteristics of fuel cells.
Based on their working principles, two kinds of compressors can be used in the PEM fuel cells, that is, positive displacement compressor and dynamic compressor.

For positive displacement compressors, screw and scroll compressors were mainly selected and investigated in the PEM fuel cell system. Considering the operating environment requirements of PEM fuel cells, the compressed air should be oil-free or with water. Hence, the oil-free and water-injected screw compressor was researched to be used in a 50 kW PEM fuel cell system. The isentropic efficiency of the prototype compressor is 55%, with a discharge pressure of 0.2 MPa and a speed of 9000 rpm. The water-injected scroll compressor was researched for a 30 kW PEM fuel cell with 5000 rpm. Qin et al. researched the operating pressure effect on the performance of a 20 kW PEM fuel cell system based on the experimental performance of an air centrifugal compressor. The results show that the optimum system operating pressure is about 1.2 atm. The pressure of air can be increased up to 2 bar easily by screw and scroll compressors. But the rotation speed of the screw and scroll compressors cannot be extremely high (usually lower than 10,000 rpm), which leads to the large volume and weight. Hence, they are not suitable for mobile devices very well.

As dynamic compressors, centrifugal compressors are probably most suitable for PEM fuel cells because they have such characteristics as high speed (corresponding to the smaller volume and lighter weight), low noise, high efficiency, and reliability. Hence, many researchers have studied centrifugal compressors in different aspects.

Many investigators focus on the operation of centrifugal compressors in fuel cell systems to enhance the efficiency of PEM fuel cells and the stability of centrifugal compressors. Empirical models of centrifugal compressors were introduced, and the available models were evaluated with the experimental data. The dynamic model of a centrifugal compressor capable of system simulation in the virtual tested computational environment was presented, which included different compressor losses. And it can be applied to virtually any centrifugal compressor and fuel cell system. A semi-mechanical and semi-imperial air supply system model based on centrifugal air compressors was established for a 150 kW PEM fuel cell engine. Experimental and simulation results show that the air supply system has good accuracy and celerity for the air compressor control. An automobile fuel cell system using a two-stage turbo-compressor for air supply was researched, and the results show that a two-stage turbo-compressor with a turbo-expander can significantly affect overall system power and efficiency. Hu et al. studied the optimization of speed response of air compressor for hydrogen fuel cell vehicle considering the transient current. And a new speed regulation control method (the transient current injection control method) was proposed.

Surge and stall are the crucial characteristics of the centrifugal compressor. Azizi et al. studied the stall/surge of a multi-stage compressor in a hybrid solid oxide fuel cell-gas turbine system. The results show that the compressor provides a sustained air flow rate during the mild stall/surge event. Godard et al. tested the surge characterization of a centrifugal compressor for fuel cell, and the compressor surges abruptly with no pre-stall activity at a high rotation speed. An agglomerate model is introduced, and a fuel cell system model including a dynamic compressor is implemented. The results indicate that the oxygen concentration is most strongly affected by the surge evolution of the compressor. Lagrouche et al. proposed a load governor based on constrained extremum seeking PEM fuel cell oxygen starvation and compressor surge protection, and the scheme was verified by experiment. A fuzzy logic control solution was proposed for the compressor group supplying an embedded 5 kW PEM fuel cell system.

Several studies focus on the design, performance optimization, and new bearings of centrifugal compressors. A nonlinear autoregressive moving average with exogenous inputs model for an air compressor of PEMFC system was established, and it has a small fitting error. Wan et al. researched the improved empirical parameters design method for centrifugal compressors in PEM fuel cell vehicle applications. And the water-lubricated bearing for high-speed centrifugal compressors was studied by Ren et al.. The performance of centrifugal compressors for a fuel cell was improved using aerodynamic optimization and data mining methods.

Many of these studies on centrifugal compressors used in PEM fuel cells are concerned with the operating performance and control methods of centrifugal compressors in PEM fuel cell systems. This study presents the structural parameters of a small high-speed centrifugal compressor for PEM fuel cells, and the main factors affecting the performance of compressors are discussed. The performance of a compressor is evaluated by the CFD method. And it is verified by the methods of experiment. The results of this paper can be the references for the R&D of small high-speed centrifugal compressors.

### 2 | GEOMETRIC MODEL AND PARAMETERS

#### 2.1 | Design input parameters

The main parameters of the compressor are the mass flow rate and discharge pressure because the primary function
of compressors is supplying compressed air (oxygen) to the stacks of PEM fuel cell systems.

The inlet temperature $T_{\text{in}}$ and pressure $p_{\text{in}}$ are another two input parameters. The inlet temperature $T_{\text{in}}$ is the atmospheric temperature, that is, 293.15 K, which is suitable for most locations in the world. The inlet pressure $p_{\text{in}}$ usually is chosen to be 1.01325 bar (standard atmospheric pressure).

The suction filter needs to be installed on compressors to get clean air. And the flow loss $\Delta p$ in suction filters should be counted. The value of $\Delta p$ is about 300~3000 Pa which depends on the performance of suction filters. Hence, the inlet pressure of compressors can be calculated as follows Equation (1). And the inlet temperature $T_{\text{in,D}}$ can be obtained on the assumption that the pressure decreases with the isentropic process.

$$p_{\text{in,D}} = p_{\text{in}} - \Delta p$$ (1)

The mass flow rate of compressors influences the oxygen concentration in the stack of fuel cells. And the mass flow rate of compressors is decided by the output power of PEM fuel cell systems. For different power scale PEM fuel cell systems, the theoretical value of the mass flow rate of compressors can be calculated by Equation (2).\(^30\)

$$Q_{m,T} = 3.58 \times 10^{-4} \times \lambda \times \frac{P_e}{V_c} \text{ (kg/s)}$$ (2)

where $\lambda$ is the air stoichiometry, the value is usually about 2.0; $P_e$ is the power of the PEM fuel cell system (kW); $V_c$ is the average cell voltage, and the value of it is 0.65 V.

Considering the leakage, the mass flow rate of compressors should be a little bigger than the theoretical value. Hence, the mass flow can be obtained by Equation (3).

$$Q_{m,D} = 1.03 \times Q_{m,T}$$ (3)

The mass flow rate of this study is 70 g/s, which can be used in the PEM fuel cell system with a maximum power of 60 kW.

The discharge pressure of compressors is usually equal to the operating pressure of fuel cell stacks. And there is an optimal value of operating pressure considering the efficiency and cost of PEM fuel cell systems. The operating pressure is usually between 1.0 and 4.0 bar (absolute pressure).\(^14\) And the common operating pressure is 1.5~2.5 bar for PEM fuel cells used in automobiles. To keep some design margin, the discharge pressure $p_{d,D}$ is generally determined as follows.

$$p_{d,D} = p_{\text{in,D}} + (1.02 \sim 1.05)(p_d - p_{\text{in,D}})$$ (4)

2.2  |  Compressor parameters

The centrifugal compressor for PEM fuel cells is usually composed of the impeller, diffuser, volute, rotor, motor, bearing, and controller. The main working parts of the centrifugal compressor are the impeller, diffuser, and volute. Figure 1 shows the main structure of the impeller. The main parameters of the prototype in this paper are shown in Table 1.

2.3  |  Impeller strength analyses

The structural parameters of the impeller should be designed carefully because the rotation speed of the impeller is usually more than 100k rpm, and the peripheral velocity of the impeller is more than 400 m/s, which means the centrifugal force of the impeller is very high. To ensure the reliability of impellers, the unshrouded impeller is usually selected even though unreasonable structure parameters and material selection would still lead to the cutdown of the lifetime and damage of impellers.

Figure 2 shows the results of strength analyses for different blade thicknesses of the design impeller. The values of the blade thickness are shown in Table 2. From Figure 2,
it can be seen that the maximum stress of the impeller is 399 MPa for model A. And it is located on the later part of the blade root. When the thickness of the blades increases (from model A to model B shown in Table 2), the maximum stress is reduced to 191 MPa. Simultaneously, the chamfer radius between the blades and hub is another key parameter for the stress distribution of impellers. And the suggested value of the chamfer radius at the blade roots is over 1 mm.

TABLE 1 Main parameters of compressor

| Parameters                        | Value |
|-----------------------------------|-------|
| Hub diameter $d$/mm               | 15.0  |
| Suction diameter of impeller $D_1$/mm | 39.0  |
| Outlet diameter of impeller $D_2$/mm | 67.0  |
| Inlet diameter of diffuser $D_3$/mm | 70.0  |
| Outlet diameter of diffuser $D_4$/mm | 101.0 |
| Width of impeller $b_1$/mm        | 3.5   |
| Width of diffuser $b_2$/mm        | 3.5   |
| Blade angle $\beta_{2A}$          | 53.0  |
| Number of blades $z$              | 10    |

3 | CFD SIMULATION AND FLOW ANALYSIS

3.1 | CFD model

Based on the design, the 3D model is established and simulated by the CFD method to predict the internal flow and performance of the centrifugal compressor. The inlet boundary is set at the total pressure boundary,
and the outlet boundary is the static pressure boundary. The frozen rotor approach is used between the impeller passage and diffuser. The two-equation $k-\omega$ based shear stress transfer (SST) model is used as the turbulence model, and the upwind method is used in the solver. The models are calculated under different outlet mass flow rates and rotation speeds. Figure 3 shows the mesh of the CFD model. The element number of the impeller is 568607, and the total element number of the whole model is 1229819.

Table 3 shows the details of boundary conditions. And the working conditions of the compressor during the simulation and experiment are present in Table 4.

To verify the independence of the grid, some cases with the different numbers of the grid (752168, 1014857, 1229819, and 1456580) were calculated. The results show that the effect of the grid on the compressor efficiency can be ignored when the number of grids is greater than 1229819.

3.2 | CFD results for compressor

The CFD simulation of the compressor is carried out on the whole machine model (including the impeller, diffuser, and volute). Figure 4 shows the internal flow in the centrifugal compressor. The outlet pipe is added to the simulation model, and the length of the outlet pipe is 5 times its diameter. Figure 4A shows the static pressure in all the compressor domains, and Figure 4B shows the air velocity in all the parts of the compressor when the rotation speed is 115,000 rpm.

3.3 | Flow in impeller and tip clearance

The internal flow of impellers mainly influences the performance of centrifugal compressors. The working condition is a factor having an essential effect on the internal flow of impellers. Figure 5 shows the internal flow in the impeller when the rotation speed is 100k rpm (at 50% span of blades). The mass flow rates are 30 g/s and 70 g/s, respectively, which correspond to two typical working conditions (small and medium flow rates) of compressors. The results of velocity streamlines in impeller passages show that the area of flow separations increases with the decrease in the mass flow rates. The size of the vortex in impellers is shown as red circles in Figure 5. The vortex leads to flow loss in impellers. Hence, there is a great change in the efficiency of centrifugal compressors when the operating conditions are changed.

The value of tip clearances has a significant effect on the performance of centrifugal compressors when the structure of the impeller is unshrouded. There is a leakage flow between the suction surface and the pressure surface of blades because of the pressure gradient and the relative motion between the blade tip and the shroud wall. The flow is simulated when the tip clearances are 0.0, 0.25, 0.5, and 0.75 mm. Figures 6 and 7 show the details of leakage at the tip clearance and their influences on the performance of centrifugal compressors.

Figure 6 shows the velocity contours at impeller outlets when the rotation speed is 100k rpm and the mass flow rate is 70 g/s. The leakage flow is obvious in Figure 6 if the impeller is an unshrouded structure. With the increase in tip clearance, the leakage from the pressure...
surfaces to the suction surfaces of the blades becomes serious, and the uniformity of the velocity at the outlet deteriorates. The leakage flows result in additional flow loss in impellers, usually called the drowned flow loss. And then, the efficiency of compressors goes down because of these flow losses. Hence, the tip clearance value should be minimized by considering the characteristics of the rotor and driving systems. Figure 7 shows the total discharge pressure of the impeller at different tip clearances. The total pressure of the compressor decreases with the increase in the tip clearance when the mass flow rate is fixed. The total pressure can reflect the ability of impellers to some extent, which means that the leakage flow leads to the decrease in the impeller’s ability to do the work. For the engineering design of centrifugal compressors for PEM fuel cell systems, the suggested tip clearance is around 0.5 mm, which can still guarantee the performance of impellers.

### EXPERIMENTAL METHOD

Based on the design of the centrifugal compressor, the prototype was manufactured. Figure 8 shows the main parts (impeller/volute) of the compressor. The material of the impeller and volute is aluminum alloy.

The prototype was tested on the compressor test system. The schematic of the test system is shown in Figure 9. The test system is composed of the air filter, compressor, air cooler, power, control valve, controller, pipe, test sensors, and the data acquisition system. The inlet/outlet temperature, pressure, and flow rate are tested by sensors and obtained by the data acquisition system. During the

**TABLE 3** Boundary conditions in CFD model

| Boundary                        | Conditions   |
|---------------------------------|--------------|
| Inlet of impeller               | Total pressure|
| Outlet of volute                | Static pressure|
| Interface of impeller and diffuser | Frozen rotor   |
| Walls of impeller               | Rotating wall|
| Other walls                     | Stationary wall|

**TABLE 4** Working conditions of compressor

| Items                        | Conditions       |
|------------------------------|------------------|
| Rotation speed/rpm           | 0–115,000        |
| Inlet pressure/*10^5 Pa       | 1.0              |
| Inlet temperature/°C         | 25               |
| Outlet pressure/*10^5 Pa      | 1.0–2.4          |

**FIGURE 4** Flow in all parts of centrifugal compressor. (A) Pressure contour, (B) Velocity contour

**FIGURE 5** Flow in impeller at different mass flow rates. (A) 30 g/s, (B) 70 g/s
test process, the coolant is used to reduce the temperature of the compressor and controller. The rotation speed of the compressor is changed by the controller, and the discharge pressure is set by changing the opening of the control valve.

In the experiment process, the rotation speed of the compressor is set to a certain speed by the controller firstly. Secondly, the flow rate of the compressor and discharge pressure are tested by the data acquisition system. Then, the control valve opening is changed, and the flow rate and discharge pressure are changed accordingly. Finally, the performance of the compressor under different discharge pressure and rotation speeds is obtained.

5 | RESULTS AND DISCUSSIONS

The discharge pressure (static absolute pressure) is obtained from the simulation and experiment research under different mass flow rates and rotation speeds. Figure 10 shows a comparison of these results. The simulated discharge pressure is close to the test data in the
studied working conditions, and the maximum error is less than 6%. The maximum error occurs in the working conditions of small and large mass flow rates, which is near the surge and choke conditions of centrifugal compressors. When centrifugal compressors are operated in the surge and choke conditions, there are strong unsteady flows in the impeller passage. And the steady-state simulation by the CFD method cannot predict the performance of the compressor operating in unsteady flow conditions very well, which is the main reason that the maximum error occurs in the working conditions of the small and large mass flow rates. Hence, the CFD model of the whole machine can predict the designed centrifugal compressor very well, especially in the operating conditions keeping away from the surge and choked conditions.

Figure 11 shows the full test conditions characteristic curve of mass flow rate vs. discharge pressure. The surge line (red dotted line in Figure 11) is obtained from the experimental results. Centrifugal compressors would be damaged if they are operated in surge condition for a long time. Hence, the centrifugal compressor cannot be operated in the left region of the surge line. To guarantee the reliability of centrifugal compressors, the surge line data with an allowance should be input to the controller of PEM fuel cell systems before centrifugal compressors are operated. Then, the controller can make sure that the controlled compressor is not working in the surge region by collecting the data of the pressure, rotation speed, and mass flow rate and comparison them with the input data of the test surge line.

The actual working conditions of centrifugal compressors are decided by both the performance of compressors and the flow resistance characteristic of the PEM fuel cell stacks. The blue area in Figure 11 is the suggested operating region for centrifugal compressors because the high-level performance of compressors can be gotten in this area.

6 | CONCLUSIONS

A small centrifugal compressor is simulated with the CFD method and tested in the actual working conditions in this
The design method of centrifugal compressors for PEM fuel cells is present in this paper. The design input parameters of centrifugal compressors can be obtained based on the output power of PEM fuel cells and their operating pressure. The stress analysis of high-speed impellers is simulated, and the weak region is the root of the blades. The thickness of the blades and the chamfer radius of blade roots are the main factors that should be decided carefully.

The simulation results show that the internal flow in compressors is influencing the performance of centrifugal compressors strongly. As the value of tip clearance rises, the uniformity of the velocity at the outlet deteriorates. The suggested tip clearance is around 0.5 mm.
The prototype of the centrifugal compressor is manufactured and tested. The simulated discharge pressure is close to the test data in the studied working conditions, and the maximum error is less than 6%. The full conditions characteristic curve of mass flow rate vs. discharge pressure is obtained by experiment. The isentropic efficiency of the prototype is up to 70% under the main working conditions, and it is 75% on the design point. When the PEM fuel cell is operated under the part-load working conditions, the surge of centrifugal compressors should be considered. And the control method of the compressor of PEM fuel cells is proposed by considering the surge line.

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DATA AVAILABILITY STATEMENT
Data are included in the manuscript.

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