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A comparative study on the reducing flow rate design method for a desalination energy recovery pump as turbine

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Abstract

The energy recovery and utilization of high-pressure brine are the key to reduce the cost. Reducing flow rate method (RFM) was put forward firstly to adapt pump as turbine (PAT) to a wide range of operating conditions in this paper. In order to validate advantages of the new method, three different designs of PAT were obtained, respectively, based on three different methods, which were, respectively, selection design method, forward curved method and RFM. Based on numerical method, the external energy characteristics, axial force and internal flow patterns of PATs were analyzed. Results show that matching of the blade inlet angle and flow angle at the blade leading edge has a great influence on the turbine flow patterns which are directly related to the efficiency of PAT. A PAT designed by the new design method RFM improves both the hydraulic efficiency and shaft power under design and partial load conditions. Moreover, the axial force of the PAT designed by RFM is reduced, which can improve the reliability of the PAT operation with high rotating speed.

Keywords Seawater desalination · Energy recovery PAT · PAT design method · Hydraulic performance · Axial force

Abbreviations

BEP	Best efficiency point
CFD	Computational fluid dynamics
PAT	Pump as Turbine
RO	Reverse osmosis membrane
SDM	Selection design method
FCM	Forward curved method
RFM	Reducing flow rate method

1 Introduction

Seawater desalination by reverse osmosis (RO) technology is one of the most important methods to solve the global water shortage [1]. Energy recovery units with pump as turbine (PAT) structure are widely used to recover high-pressure seawater without passing through RO membranes, and a

series of studies have also proved that PAT is suitable for the recovery and utilization of fluid pressure energy [2–5]. The energy consumption in the RO system determines the cost of freshwater production, so the PAT is used to recycle the high-pressure energy of the concentrated brine before membrane. The pressure and capacity of seawater passing through the RO membrane are susceptible to the influence of season, weather and temperature, the pressure and capacity of recoverable high-pressure concentrated brine are also variable, which means that the energy recovery PAT will operate at different flow rate conditions with about 55 bar pressure [6]. Therefore, it is very important to design a PAT with high-efficiency range which can suit for the complex flow and multiple operating conditions. However, the special design method of PAT for the RO system has rarely reported, and there are few researches on the high-efficiency range of PAT under multiple operating conditions.

Selection design method (SDM) and forward curved method (FCM) are two widely used design methods for the PAT design. SDM is a conventional design method of PAT, and the reasonable selection is directly related to the efficiency and operation stability of energy recovery PAT. Liu et al. discussed the relationship between the performance parameters under the pump mode and turbine mode according to the basic energy equation and the assumption that the flow movement is similar [7]. Shi et al. studied the

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performance of pump in the turbine operating condition by experiment and measured the conversion coefficient of flow and head of the PAT with different specific speed [8]. On the basis of collecting and summarizing a large number of experimental data, Yang et al. put forward the conversion equation between pump and turbine performance parameters [9]. It was found that the prediction method of turbine performance parameters at best efficiency point (BEP) given by Yang et al. agrees well with the experimental data compared with the empirical formula proposed by Sharma [10] and Stepanoff [11]. Jain et al. proposed that the turbine efficiency can be increased by 1–4% by rounding the inlet edge of the turbine blade and rounding the outer circles of the impeller shroud [12–14]. Based on the basic parameters of volute, Wang et al. established the relationship expression between the inlet angle of impeller blade and the design flow, and the rationality of the expression was verified by experiment and numerical simulation, which provides a reference for the design of special turbine impeller [15]. David et al. used the numerical simulation method to analyze the internal flow of the PAT under different operating conditions and proposed a model to predict the performance of the turbine under all operating conditions, which could provide a reference for the performance prediction of the PAT [16]. Rossi et al. developed an analytical model, based on a statistical analysis of experimental data related to 32 different PATs, which can predict the off-design operating conditions in turbine mode by the best efficiency point (BEP) of the PAT [17]. This methodology is based on the non-dimensional parameters and the normalization process with respect to the PAT BEP value in turbine mode. A model for predicting the performance of PAT under all operating conditions, which takes the specific speed of pump and turbine as an independent variable, is proposed by Stefania et al. [18]. Stephen et al. used Latin hypercube sampling method and unified regression neural network model to optimize the structural parameters of turbine impellers, which improved the hydraulic performance of the turbine under off-design conditions [19]. In summary, these related research results above mainly focus on the performance relationship between pump and turbine at BEP, or the hydraulic performance of the pump as a turbine is optimized by changing the geometric parameters of the blades. A new PAT design method considering high-efficiency and stable operation of multiple operating conditions is urgently needed to meet the design of RO seawater desalination system.

The seawater desalination energy recovery PAT in the RO system is a key device with special structure. High-pressure booster pump unit and PAT unit are combined at two sides of the machine with same shaft, and PAT working as the motor supplies the power to the pump without extra power. In this paper, a new design method which considered the efficiency and stability of the multiple operating conditions

was studied to design the desalination energy recovery PAT. Firstly, three models which are separately designed by SDM, FCM and RFM were established. Secondly, based on numerical calculation method, SST $k-\omega$ turbulence model is used for numerical calculation and verified through experimental test. Finally, the combined performance of the hydraulic efficiency, axial force and flow patterns of the three different PATs is compared and analyzed based on the factors affecting the hydraulic efficiency and operation reliability of the PAT. RFM is verified to be the optimized method based on SDM, which provides a new theoretical reference for the hydraulic design of a PAT, especially applied in the RO seawater desalination system.

2 Hydraulic design methods of energy recovery PAT

2.1 SDM for PAT design

The SDM is commonly used for hydraulic design of PAT. Figure 1 shows the process of SDM. Firstly, the specific speed of the PAT is calculated by Eq. (1) according to its design parameters. Secondly, according to Eqs. (2)–(4), which is the relationship between the performance parameters and specific speed of a centrifugal pump under pump condition and PAT condition [9], the design parameters of BEP and a specific speed under pump operating conditions are calculated. Thirdly, based on specific speed of pump, a pump model is selected from pump model base with excellent hydraulic performance. Then, the hydraulic performance and the BEP of PAT are obtained by numerical calculation. Finally, according to the actual design parameters of the turbine under the designed condition, and based on the similar conversion criterion [20], the turbine model is scaled in the same proportion to obtain the required turbine.

Specific speed N_s is defined as follows:

Fig. 1 Process of SDM

$$N_s = \frac{3.65n\sqrt{Q}}{H^{3/4}} \quad (1)$$

where n is rotational speed, Q is flow rate, H is head. The specific speed, flow rate and head could be converted from the pump operating condition to the turbine operating condition by following equations [21]:

$$N_{sp} = 1.12N_{st} + 6.21 \quad (2)$$

$$\frac{Q_t}{Q_p} = \frac{1.2}{\eta_p^{0.55}} \quad (3)$$

$$\frac{H_t}{H_p} = \frac{1.2}{\eta_p^{1.1}} \quad (4)$$

where N_{st} , N_{sp} are the specific speed of turbine and pump. Q_t , Q_p are the design flow rate of turbine and pump, m^3/h , respectively. H_t , H_p are the design head of turbine and pump, m , respectively. η_p is the design efficiency of pump.

2.2 FCM for PAT design

Figure 2 shows the design process of FCM [15]. The blades structure of impeller is forward curved. Firstly, the volute model of PAT and the inlet and outlet diameter of impeller were designed by SDM method. Secondly, according to the geometrical parameters of the volute, blade inlet angle and outlet angle of an impeller are designed by using Eqs. (5)–(9) [15].

$$\psi_1 = 1 - \frac{\delta_1 Z}{2\pi R_{1c}} \quad (5)$$

$$k_1 = \frac{\varphi_0}{720\pi \left(a_0 - \sqrt{a_0^2 - \rho_0^2} \right)} \quad (6)$$

Fig. 2 Process of FCM

$$\rho_0 = \sqrt{F_0/\pi} \quad (7)$$

$$\tan(180^\circ - \beta_1) = \frac{60Q_t}{\pi b_1 \psi_1 (120k_1 Q_t - \pi n D_1^2)} \quad (8)$$

$$\beta_2 = \arctan \frac{Q_t}{A_2 u_2} \quad (9)$$

where ψ_1 is blade inlet extrusion coefficient, δ_1 is circumferential thickness of blade inlet, mm . R_{1c} is radius of the bus center at the blade inlet, mm . k_1 is volute structure coefficient. φ_0 is the wrap of volute, $^\circ$. a_0 is distance from center of the eighth section of volute to center of the impeller, mm . ρ_0 is equivalent circular radius of the eighth section of volute, mm . F_0 is area of the eighth section of the volute, mm^2 . β_1 is blade inlet setting angle, $^\circ$. b_1 is blade inlet width, mm . D_1 is blade inlet diameter, mm . β_2 is blade outlet setting angle, $^\circ$. A_2 is actual cross-sectional area of the axial flowing at the outlet of turbine impeller, mm^2 . u_2 is circumferential velocity at the impeller outlet, m/s .

2.3 RFM for PAT design

The operating condition of seawater desalination energy recovery PAT is changeable, and PAT efficiency curve generally decreases steeply from design condition to part-load condition but decreases slowly from design condition to large flow condition. Besides, the high-pressure booster pump and PAT are combined at two sides of the device, so the axial force balance between booster pump and PAT needs to be considered to improve the operating reliability. Therefore, a new method called RFM for designing energy recovery PAT is proposed, which is by reducing the flow rate. RFM is based on SDM, but it is obviously different from SDM. The PAT designed by SDM has a good performance only under design condition, but the performance decreases sharply at off-design conditions, especially under partial load conditions. The PAT designed by RFM is not only small in size, but also has a good performance when the PAT operates under design condition and off-design conditions. Therefore, the PAT designed by FCM can operate efficiently and stably at multiple operating conditions.

Figure 3 shows a main process of RFM. Firstly, when designing an energy recovery PAT, the rated flow rate of the PAT should be appropriately reduced using Eq. (10) with keeping other design parameters such as head and speed unchanged. Secondly, based on the new designing parameters, the energy recovery PAT could be designed by using SDM. In this design case, the optimal scaling coefficient λ is 0.764.

Fig. 3 Process of RFM

Fig. 4 RO seawater desalination system

$$Q_t^* = \lambda \cdot Q_t \quad (0 < \lambda < 1) \quad (10)$$

where the Q_t is design flow rate of turbine, m^3/h . Q_t^* is actual design flow rate of turbine, m^3/h .

3 PAT design models and meshing

3.1 Design models of PAT

Figure 4 shows a reverse osmosis seawater desalination system. The integral machine device consists of high-pressure centrifugal pump unit and PAT unit for energy recovering device. The high-pressure pump driven directly by the PAT with same shaft is used to pressurize the seawater to pass through the permeable membrane. A part of the high-pressure seawater without penetrating membrane will flow into the PAT. The energy recovery PAT converts the high-pressure energy of concentrated seawater to drive the high-pressure pump on the other side. Thus, the energy of high-pressure seawater without passing reverse osmosis membrane can be recovered totally. This new energy recovery PAT was considered as object in this paper.

The design parameters of the turbine are flow rate $Q_t = 50.9 \text{ m}^3/\text{h}$, inlet pressure $P_{in} = 5 \text{ MPa}$, rotational speed $n = 20,000 \text{ r/min}$. In order to study the hydraulic performance and axial force distribution of PAT designed by different methods, three different PAT models were, respectively, designed by SDM, FCM and RFM. The main parameters of impeller and volute are shown in Table 1. The model A is the impeller model established by SDM, the model B is the impeller model established by the FCM, and the model C is the model designed by RFM. Figure 5 shows the corresponding three-dimensional models of PAT impeller designed by different methods.

3.2 Computation domain and meshing

Figure 6 shows the computation domain of the designed PAT, which mainly includes suction chamber, volute, impeller, front and back chamber and draft tube. A hexahedral structured mesh was created in ANSYS-ICEM to discretize the entire computational domain. The meshes near the impeller blade and volute tongue wall were partially refined

Table 1 Basic parameters of different PAT models

Components	Parameters	Model A	Model B	Model C
Turbine impeller	Inlet diameter D_2 (mm)	82	82	75
	Inlet width b_1 (mm)	6.5	6.5	5.8
	Inlet blade angle β_{b2} ($^\circ$)	26	90	26
	Outlet blade angle β_{b1} ($^\circ$)	24	24.9	24
	Blade wrap angle ($^\circ$)	130	35	130
	Number of blades Z	6	11	6
	Outlet diameter D_3 (mm)	35	35	32
Volute	Base circle diameter of volute D_3 (mm)	86	86	78.5
	Volute width b_3 (mm)	18	18	13
	Volute inlet diameter D_1 (mm)	25	25	23

Fig. 5 Impeller hydraulic models. **a** Model A designed by SDM, **b** Model B designed by FCM, **c** Model B designed by RFM

Fig. 6 Computation domain of PAT

Fig. 7 Turbine mesh generation. **a** Impeller grid, **b** Y^+ values of blades

to accurately simulate the characteristics near the wall. The Y-type topology was used to generate the mesh of impeller calculation domain inlet. As shown in Fig. 7, the mesh diagram of turbine impeller and Y Plus distribution of blade were shown, in which the thickness of the first mesh on the blade wall was 0.02 mm and the maximum Y^+ value was 34.

4 Numerical method and settings

4.1 Numerical setup

The flow inside the PAT is essentially turbulent flow. The shear stress transport (SST) turbulence model was used in this work. The SST $k-\omega$ turbulence model improves the definition of eddy viscosity to consider the effect of turbulent principal shear stress transport, which improves the prediction of inverse pressure gradient flow [22–24]. Therefore, the SST $k-\omega$ turbulence model was used for this numerical calculation, where the turbulent kinetic energy k and

turbulent frequency ω were obtained by solving the following equations [25]:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(pk\bar{u}_i) = \frac{\partial}{\partial x_j}\left(\Gamma_K \frac{\partial k}{\partial x_j}\right) + G_K - Y_K \quad (11)$$

$$\frac{\partial}{\partial t}(\rho \omega) + \frac{\partial}{\partial x_j}(p\omega\bar{u}_j) = \frac{\partial}{\partial x_j}\left(\Gamma_\omega \frac{\partial \omega}{\partial x_j}\right) + G_\omega - Y_\omega + D_\omega \quad (12)$$

The equation for calculating the turbulent dynamic viscosity coefficient μ_t is

$$\mu_t = \frac{\rho k}{\omega} \cdot \frac{1}{\max\left[\frac{1}{\alpha^*}, \frac{SF_2}{\alpha_1 \omega}\right]} \quad (13)$$

The PAT impeller was set as the rotating domain, and the rotating speed was the same as that of the impeller. Suction chamber, volute, front and back chamber and draft tube were set as stationary domain. The reference pressure was set to 1 atm. The main boundary conditions were set as follows: The inlet was fixed with the corresponding mass flow rate, the outlet was free flow with the static pressure value zero. The wall roughness was set to 50 μm , and the automatic wall function was used in the near wall region. The direct pressure–velocity coupling method was used to solve the governing equations, and the high-resolution scheme was used for the convection terms. Governing equations were discretized with two-order upstream scheme. The convergence accuracy is set to 10^{-5} .

4.2 Mesh independence test

The grid independence test was carried out by setting five groups of computational domain grids with different numbers, and the results are shown in Fig. 8. The results show

Fig. 8 Grid independence analysis

that the calculated head is steady when the number of grids exceeds 4.0×10^6 . In order to ensure the accuracy of numerical calculation and save calculation time, mesh 4 was used to study. The total number of grid cells used in this research model is 4185773.

4.3 Numerical results and validation

It is a very important step to test the performance of PAT by experiment before comparing and analyzing the performances of PATs designed by different methods based on numerical calculation results. However, the PAT performances were unable to test at 20000 rpm because of the limited test conditions in the laboratory, and the speed of the energy recovery turbine is too high. Therefore, the centrifugal pump used in the process of designing turbine by SDM method was processed, and the performance of the centrifugal pump at different speeds in turbine mode was tested experimentally and calculated numerically. The selecting PAT model is a scaled model for desalination energy recovery turbine, and its design rotational speed is 1450 rpm. According to similarity criterion [26] and similar conversion Eq. (14), the geometric dimension of PAT model is 2.9 times that of energy recovery turbine. In order to verify the reliability of the numerical calculation, the performances of the PAT model at different rotational speeds were tested and calculated.

$$\frac{Q_{md}}{Q_d} = \left(\frac{n_{md}}{n_d}\right) \left(\frac{D_{2m}}{D_2}\right)^3 \quad (14)$$

where Q_{md} , n_{md} and D_{2m} are design flow rate, design rotational speed and impeller outlet diameter of PAT model, respectively. Q_d , n_d and D_2 are design flow rate, design rotational speed and impeller outlet diameter of PAT, respectively. D_{2m}/D_2 is similar conversion size scale of PAT.

Figure 9a shows the schematic diagram of the turbine model unit performance test stand. During the test, the pressure and flow signals were collected and processed by signal box and displayed on the computer. In this paper, the flow rate and the output pressure of boost pump were regulated by the frequency converter, and the parameters at the input of the turbine were changed. The speed of the turbine was controlled by adding a load to the dynamometer, thus the performance characteristic curves of the turbine at different speeds were obtained. Figure 9b shows the field test diagram of turbine model machine, in which the measuring range of pressure sensor was 1.6 MPa and the measuring accuracy was 0.01 Pa. The pressure-bearing range of electromagnetic flowmeter was 0–1.6 MPa and the measuring accuracy was $10^{-4} \text{ m}^3/\text{h}$.

Figure 10 shows the external characteristic curves of the numerical calculation and experimental test of the PAT

Fig. 9 Performance test of model turbine. **a** Schematic diagram of turbine performance test bench, **b** Turbine field test

model under different rotational speed. The equations for turbine head H , shaft power P and efficiency η are as follows:

$$H = \frac{P_{\text{in}} - P_{\text{out}}}{\rho g} \quad (15)$$

$$P = M \cdot \frac{2\pi n}{60} \quad (16)$$

$$\eta = \frac{P}{\rho g Q H} \times 100\% \quad (17)$$

where P_{in} is the total pressure at the turbine inlet, P_{out} is the total pressure at the turbine outlet. M is the torque; n is the rotating speed.

The results show that the experimental and numerical results have a good agreement on the trend of the external characteristics with the flow rate, and the operating conditions of the highest efficiency point are also basically the same. It is noted that there are certain deviations between the

numerical and experimental results, which may be caused by the difference of geometry and the roughness because the PAT model used for the simulation is impossible to be totally same as that for experimental investigation. However, generally speaking, the error between the numerical calculation and the experimental results of the turbine model is very small in the global range. Under the optimal efficiency condition at design rotational speed (1450 rpm), the relative errors of turbine head and efficiency are 10.1% and 0.7%, respectively. The reason for the large head relative error is that the head of turbine is only 14.9 m. Therefore, the selected mesh arrangement and numerical methods are qualified to guarantee the accuracy and reliability of the present work.

Based on the reliability verification of the numerical calculation method, the performance of the turbine model at rotating speed ($n = 1450$ rpm) and at rotating speed ($n = 20,000$ rpm) was calculated, and the corresponding velocity triangles were analyzed and compared. Figure 11 shows the velocity triangles of turbine inlet at different rotational speed. u_1 , c_{u1} and c_{m1} are rotational velocity, circumferential velocity and axial velocity, respectively, which can be obtained from the CFD-post-process. Therefore, the velocity triangle can be obtained. As shown in Fig. 11a, at $n = 1450$ rpm, the u_{10} , c_{u10} and c_{m10} of the turbine model under the best efficiency condition are 31.88 m/s, 15.98 m/s and 4.78 m/s, respectively, which gives a β_{10} of 16.73°. And at $n = 20,000$ rpm, the u_{11} , c_{u11} and c_{m11} of the turbine under the best efficiency condition are 78.5 m/s, 27.4 m/s and 15.6 m/s, respectively, which gives β_{11} equaling to 16.97°. The results show that at different rotating speed, the velocity triangle is similar and the flow angle is also basically the same. Therefore, the numerical results of the turbine at the design rotational speed are considered to be reliable.

5 Results and Discussion

5.1 Hydraulic performance of PATs designed by different methods

Figure 12 shows the hydraulic performance curves of PAT models designed by different methods, and Table 2 mainly shows the performance parameters under $0.8Q_d$, $1.0Q_d$ and $1.2Q_d$ conditions. As shown in Fig. 12a, the head increases with the increase in flow rate, the head of three kinds of PATs increases gradually, and the slope of the $Q-H$ curve of the PAT model C is the largest. Under $0.7Q_{\text{opt}}-0.9Q_{\text{opt}}$ condition, where Q_{opt} is design condition, the head of model B is higher than that of the other two PATs, while after $0.9Q_{\text{opt}}$ operating condition, the head of the model C is raised rapidly, higher than that of the other two PATs. Under the design condition, the maximum head of the model C is

Fig. 10 Numerical simulation and experimental test results of turbine model

Fig. 11 Velocity triangle of turbine inlet at different rotational speed **a** $n = 1450\text{ rpm}$ **b** $n = 20000\text{ rpm}$

510.7 m. It is 17.9% and 10.4%, respectively, higher than that of model A and Model B.

Figure 12b is $Q-\eta$ curves. Model B shows a trend of efficiency increasing gradually with the increase in flow rate, while the efficiencies of model A and model C increase firstly in a large range and then decrease gradually. The efficiencies at the design condition of model A, model B and model C are 81.4%, 70.1% and 79.1%, respectively. The efficiency of model C is only 1.5% lower than that of model A. However, the maximum efficiency of the model C is 80.9% under $0.8Q_{\text{opt}}$ condition, which is 8.1% and 24.3% higher than that of model A and B, respectively, under partial load conditions. The results show that the model C can improve the efficiency under partial load conditions.

Figure 12c is $Q-P$ curves where P is shaft power of PAT. With the increase in flow rate, the shaft powers of three kinds of PATs increase gradually, and the growth trends are similar. However, under $0.7Q_{\text{opt}}-1.0Q_{\text{opt}}$ operating condition,

Fig. 12 External characteristics of PATs designed by different methods. **a** Flow rate—head curves, **b** Flow rate—efficiency curves, **c** Flow rate—shaft power curves

Table 2 Performance parameters under different conditions

Model	Parameters	$0.8Q_d$	$1.0Q_d$	$1.2Q_d$
Model A	H (m)	336.57	419.54	537.57
	P (kW)	27.18	47.35	72.2
	η (%)	72.8	81.4	80.7
Model B	H (m)	438.77	470.5	562.32
	P (kW)	27.56	45.73	72.8
	η (%)	56.6	70.1	77.8
Model C	H (m)	373.35	510.66	684.78
	P (kW)	33.5	56.03	85.37
	η (%)	80.9	79.1	74.9

the shaft power of the model A and model B is basically the same, and the shaft power of model C is higher than the other two PATs. Under the design condition, the shaft power of the model C is 56 KW, which is 15% and 18.3% higher than that of model A and model B, respectively. Therefore, PAT designed by FCM can provide maximum output of shaft power.

To sum up, under the design condition, the efficiency of model C is only 0.4% lower than model A, but its head is 17.8% higher than model A, so the output power is 15% higher than model A. Meanwhile, if the PAT operates under partial load conditions, model C can not only have a high head, but also operate efficiently and provide the maximum output power.

5.2 Axial forces comparison of PATs

The axial force of the PAT mainly comes from the axial force caused by the asymmetry of the shroud and hub of the impeller and the dynamic reaction force of the impeller [26], both of which are vectors. Equation (15) is the calculation formula of axial force.

$$F_{\text{axial}} = F_{\text{shr}} + F_{\text{hub}} + F_{\text{rea}} \quad (18)$$

where the F_{axial} is total axial force of the PAT, F_{shr} is the axial force of the shroud, F_{hub} is the axial force of hub, F_{rea} is the dynamic reaction force of the impeller.

Figure 13 shows the axial force changing curve of PAT designed by different methods. The results show that the axial force of the PAT increases with the increase in flow rate. For model B, its axial force is much higher than that of model A and model C under the whole operation condition, which indicates that the axial force of the PAT designed by FCM is too high, which is not conducive to safe and stable operation of the combined machine with pump and PAT. For model A and model C, the axial force of model C is lower than that of model A under the condition of $0.7Q_{\text{opt}}-1.4Q_{\text{opt}}$. Under the condition of $1.0Q_{\text{opt}}$, the axial force of model C is

Fig. 13 Axial force of PAT designed by different methods

1450 N, which is 9% and 53.8% lower than that of model A and model B, respectively. Under the condition of $0.8Q_{opt}$, the axial force of model C is far lower than the axial force of model A and model B, respectively. In conclusion, the PAT designed by RFM can effectively reduce the axial force of PAT under design condition and off-design condition, which is conducive to the reliable operation of PAT.

5.3 Velocity distribution in the volute of PAT

To study the circumferential velocity distribution inside the volute, a section (I–VIII) diagram of the volute is selected, and Fig. 14 shows the location and shape of each section of the volute. The volute is divided into eight sections, and the interval between adjacent sections is 45 degrees. The impeller of PAT rotates clockwise from the inlet. Figure 15 shows the comparison of the actual average velocity distribution of volute section and the theoretical values with the dotted line. It can be seen that, on the whole, the internal velocity of the volute accelerates with the contraction of the volute section, and the velocity change within the small section

Fig. 14 Volute cross section (I–VIII)

Fig. 15 Average velocity distribution of volute section. **a** $0.8Q_{opt}$ condition, **b** $1.0Q_{opt}$ condition

area of the volute, section I–IV, is the most obvious. For model A and model B, the average velocity of the volute section changes with the contraction of the volute section to the largest extent under $0.8Q_{opt}$ condition, but the variation amplitude of model B is better than that of model A. With the increase in flow rate to $1.0Q_{opt}$, the variation amplitude of the average velocity is improved. The variation amplitude is more similar with each model, and the difference from the theoretical variation trend is smaller than that under $0.8Q_{opt}$ condition. For model C, the average velocity of the volute section is lower than the theoretical average velocity.

Under $0.8Q_{opt}$ condition, in the corresponding volute section, the average velocity of model C is the maximum, and the variation range of the average velocity inside the volute is the minimum, while that inside volute of model A is the maximum. Under $1.0Q_{opt}$ condition, in the corresponding volute section, the average velocity of model C is still the largest, but the range of variation in the average velocity

inside the volute of model A is the smallest and the flow is the most stable. The above discussion indicates that the PAT designed by RFM with the small volute in size has the more uniform flow pattern in the volute, especially under partial load conditions. With the increase in the flow rate, the fluctuation of the average velocity distribution occurs with a small magnitude. The results are also consistent to the efficiency distribution shown in Fig. 12.

5.4 Blade load distribution of PAT blade

To study the load distribution of PAT blades, the loading at the middle section of the blade as shown in Fig. 16 is discussed. Figure 16 shows the relative position between the blade and the volute in which the load is extracted and shows the change direction of stream-wise from 0 to 1. Figure 17 shows the loading distribution of PAT blades under different operating conditions. The loading of the same blade profile increases with the increase in flow rate, and the loading of the blade decreases gradually from the inlet to the outlet in generally. For the PAT blade (Model A), under both $0.8Q_{opt}$ and $1.0Q_{opt}$ conditions, the pressure of pressure surface is lower than that of the suction surface at the blade inlet, which is caused by the mismatching of the blade setting angle and the flow angle. However, with the increase in flow, the chord length of the pressure surface loading at blade inlet lower than that of the suction surface decreased under $1.0Q_{opt}$ condition, indicating that the matching between the blade setting angle and the flow angle is improved compared with that under the $0.8Q_{opt}$ condition, and the loss at the inlet of impeller is reduced. For Model B, the loading on the pressure surface at the blade inlet is also lower than that on the suction surface under $0.8Q_{opt}$ and $1.0Q_{opt}$ conditions. Moreover, the blade loading distribution in the two operating conditions is basically similar, and the chord is basically equal in which the blade loading on pressure surface is lower than that of suction surface. For Model C, the load on the

pressure surface at the blade inlet is also lower than that on the suction surface under $0.8Q_{opt}$, but the chord length is very short. In addition, as the flow rate increases, the blade loading distribution is the best under $1.0Q_{opt}$, and the load on the pressure surface at the blade inlet is greater than that on the suction surface.

For the three kinds of PAT blades, under $0.8Q_{opt}$ condition, there is a phenomenon that the blade inlet angle and the flow angle are not completely consistent. For the Model C, the chord of the blade inlet with this phenomenon is the shortest and the impact loss of the inlet is the minimum. The reason for that is a little hydraulic shock on the suction side at the leading edge, which could be eliminated by the optimization of the shape of inlet. Under $1.0Q_{opt}$ condition, the inlet of model A blade and model B blade still have the phenomenon that the loading on the suction side is greater than that on the pressure side, but this phenomenon does not occur at the inlet of the PAT blade model C, and the inlet flow is streamline pattern.

5.5 Circumferential distribution of flow angle

Figure 18 shows the circumferential distribution of flow angle in the middle section of PAT impeller inlet, and the position of the eighth section in Fig. 14 is defined as 0 degrees. As can be seen from Fig. 18, the flow angle at the impeller inlet is periodic distribution in general, and its period depends on the blade number. In addition, the distribution of flow angle is affected by the action of the spiral case tongue shown by blue dash in Fig. 17 near the impeller inlet. For the PAT model A, the flow angle at the inlet of the impeller passage is smaller than the inlet setting angle under $0.8Q_{opt}$ and $1.0Q_{opt}$ conditions. In addition, the flow angle fluctuates obviously near the suction side of the blade, which indicates that the fluid shocks on the suction side of the blade. With the increase in flow rate, the change range of flow angle decreases. For

Fig. 16 Relative position of blade extracted by PAT. **a** Back curved PAT, **b** Forward curved PAT

Fig. 17 Blade loading distribution of PAT. **a** Blade load distribution of model A, **b** Blade loading distribution of model B, **c** Blade loading distribution of model C

Fig. 18 Circumferential distribution of flow angle. **a** $0.8Q_{opt}$ condition, **b** $1.0Q_{opt}$ condition

Model B, the angle of blade inlet is 90° and the flow angle of impeller inlet is far less than that of blade inlet under $0.8Q_{opt}$ and $1.0Q_{opt}$ conditions, and the circumferential distribution of liquid flow angle is disordered. It shows that the flow collides with the suction side of the blade, and the inlet flow pattern is poor. For Model C, the flow angle of the impeller inlet fluctuates around the blade angle under $0.8Q_{opt}$ condition, which indicates that the blade inlet angle and flow angle match well and the inlet loss is small. However, under $1.0Q_{opt}$ condition, the flow angle is basically larger than the setting angle, which indicates that the fluid impinges on the pressure surface of the blade.

5.6 Velocity-streamline distribution in impeller

Figure 19 shows the velocity and streamline distribution of the middle section of the PAT impeller. For Model A, under $0.8Q_{opt}$ condition, a low-speed zone appears at the pressure surface side of the upstream and middle of the blade, which is called the flow stagnation zone. The reason for that is the mismatching of blade angle and the flow angle. The suction surface of impeller has impact and the pressure surface has flow separation, especially in flow passages marked No.1 and No.6 near the volute tongue. With the increase in flow rate, the flow patterns inside the impeller are significantly improved under $1.0Q_{opt}$ condition compared with that under $0.8Q_{opt}$ condition. However, there is still a low-speed zone on the upper and middle pressure surface of the blade, and there is a high-speed zone in the middle of model A near the volute tongue. For Model B, the flow patterns in the whole impeller show the different scales of vortices under $0.8Q_{opt}$ and $1.0Q_{opt}$ conditions, especially in the blade inlet pressure surface, there is an obvious low-speed zone, with vortex generated and serious loss in the flow passage. For Model C, under $0.8Q_{opt}$ operating condition, the flow inside the impeller is uniform and streamlined, and there is no obvious vortex, indicating that the blade angle and the flow angle are relatively matched. Under $1.0Q_{opt}$ condition, the flow pattern inside the impeller is also steady. However, there is a small low-velocity region which may result in the flow separation. This is the reason the efficiency of Model C is a little bit lower than under $0.8Q_{opt}$ condition.

For the three kinds of PAT impellers with three different design method, the flow pattern in the PAT impeller designed by RFM is the best and the loss is the least under $0.8Q_{opt}$ condition. Under $1.0Q_{opt}$ operating condition, the flow patterns inside the impellers (Model A and C) were similar, although the position marked with blue dotted line at the inlet of impeller shows a small region with low speed. However, the area of high-speed zone in the middle of no. 1 impeller of the model A is larger than that of the design impeller with RFM.

6 Conclusions

In this paper, the three models of PAT designed by SDM, FCM and RCM separately are studied by combined flow field and external curves. The axial force distribution and internal flow pattern of the three PATs are compared and analyzed. The PAT designed by RFM has good hydraulic performance and output power under partial load conditions and design condition. In addition, the axial force is low in the whole operation conditions, which is helpful to the reliable operation of the PAT. This provides a reference for the

Fig. 19 Velocity-streamline distribution in the middle section of impeller. **a** Velocity-streamline distribution under $0.8Q_{opt}$, **b** Velocity-streamline distribution under $1.0Q_{opt}$

design of PAT. The following conclusions can be drawn as followed:

(1). The efficiency of the PAT designed by RFM is only 1.5% lower than that of the PAT designed by SDM at design condition. However, the maximum efficiency of the model C is 80.9% under $0.8Q_{opt}$ condition, which is 8.1% and 24.3% higher than that of model A and B, respectively, under

partial load conditions. The results show that the model C can improve the efficiency under partial load conditions.

(2). The axial force of the PAT designed by RFM is the lowest under $0.7Q_{opt} - 1.1Q_{opt}$ conditions, which is 9.9% and 14.6% lower than PAT designed by SDM, respectively, under design condition and $0.8Q_{opt}$ condition and is maintained to the reliable operation of the PAT.

(3). The average velocity variation in the volute of PAT designed by SDM is the most uniform, and the flow pattern in the impeller passage is the best when the PAT operates under $1.0Q_{opt}$ condition. But there is flow impact near the inlet of the impeller and the suction surface of the blade. The velocity variation inside the PAT volute designed by RFM is the most uniform, the inlet flow is the most stable, and the flow state in the impeller runner is the best under $0.8Q_{opt}$ condition.

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