Comparative Study on Turbulence Models of High-Speed Centrifugal Pump with Low Specific Speed

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Abstract: It is very important to use high-speed centrifugal pump with low specific instead of multi-stage high-pressure pump to reduce the volume of the whole marine reverse osmosis seawater desalination system. Based on the design model of high-speed centrifugal pump with low specific speed, we did a comparative study of 11 kinds of turbulence models with numerical simulation. Five methods (the convergent curves, calculation time step, characteristics curves, impeller fluid velocity and pressures, volute section surface streamline) are included to find the most suitable turbulence model. The results show that k-ε is the most suitable turbulence model to compare with design conditions and external characteristics, SST and RNG k-ε turbulence models appears more surface streamline details in the volute section.

1. Introduction

Reverse osmosis desalination equipment has the advantages of low energy consumption, simple, stable and reliable operation. It has developed rapidly in the desalination market. After 1980s, reverse osmosis seawater desalination equipment appeared in marine reverse osmosis desalination system. The equipment consists of preparatory treatment equipment, high pressure pump, water treatment equipment, disinfection system, valve instrument and control box. Multi-stage high-pressure pump is one of the three key equipment in reverse osmosis seawater desalination. It is very important to use high speed centrifugal pump with low specific speed instead of multi-stage high pressure pump to reduce the volume of the whole reverse seawater osmosis device. The desalination process using a high-speed centrifugal pump with low specific speed is shown in Fig 1.

The accuracy of numerical simulation of centrifugal pump depends on the selection of turbulence model. Yi Shi et al[1] the numerical simulations are implemented by solving the three-dimensional steady Reynolds-Averaged Navier-Stokes (RANS) equations and the effects of various turbulence models(k-ε, SST, SSG) shows that the predicted performance with Shear Stress Transport (SST) turbulence model at the optimal conditions agrees well with the experiment data. Gamal R.H. et al[2] use two turbulence models (k-ε and RNG k-ε) are utilized and the results are compared with experimental ones for the case of using impeller with 7 blades. It was found for the studied cases that the case of using 7 blades is the best case. Hamed Alemi et al[3] use three turbulence models (standard k-ε, low-Re k-ω and SST) to examine the pressure distributions with centrifugal pump, the results show that the trend of CFD results are in overall agreement with the measurements.

From the above references, we can see that many studies focus on centrifugal pumps with low specific speed, but the effects of different turbulence models on high-speed centrifugal pump with low specific speed are rarely studied, and the results of different turbulence models may be quite different.
11 known mature turbulence models are used in this paper. The study can help to reduce the experimental work of high-speed pumps.

2. Three-Dimensional Model and Mesh
The pump is divided into three areas: impeller, volute and outlet. Each region is independent as shown in Fig 2 and Fig 3.

In order to make the numerical simulation more accurate and to observe more details of internal flow, the mesh of centrifugal pump blade inlet, tongue and volute with large curvature was refined by using hexahedral structured mesh and tetrahedral structured mesh, as shown in fig 4. The number of element and nodes are 3142782 and 10936949, respectively.

3. Numerical analysis
3.1. The convergent curve
Figure 5. Convergence curves of numerical simulation can be divided into three categories, as shown in Fig 5:

- Normal convergence, smooth convergence of curve from high to low (SST(Q=30³/h), EVTE(Q=30³/h), k-ω(Q=30³/h), k-ω(Q=30³/h)).

- Abnormal convergence, oscillation convergence before approaching residual value (LRR(Q=30³/h), BSL(Q=30³/h), ω (Q=30³/h), RNG k-ω(Q=30³/h), ω (Q=25³/h)).

- It is not convergent, and there is a numerical effect between the turbulence model and the solver, which makes the calculation result unable to converge. And the results show that it is not in accordance with the objective law (EARSM k-ω(Q=30³/h), SSG(Q=30³/h), QI(Q=30³/h)).
3.2. Calculation time step

According to Fig. 6, the time step of RNG turbulence model reaches 333 steps when the flow rate is 35 \( m^3/h \), which means that the convergence time of RNG \( k-\varepsilon \) is much longer than that of other turbulence models. In order to observe the time step of different turbulence models more obviously, the RNG \( k-\varepsilon \) turbulence model is removed and the time step of each turbulence model is shown in Fig 7. The time step gradually decreases, and the least time step is \( k-\varepsilon \), followed by EVTE, \( k-\varepsilon \), SST.

3.3. Characteristics curve

Different turbulence models have similar trends in efficiency, head and axial power. In Fig 8, The flow rate of centrifugal pump increases gradually from 25 \( m^3/h \) to 35 \( m^3/h \), and decreases after it is higher than 35 \( m^3/h \). It can be seen that the maximum efficiency of each turbulence model is 5 \( m^3/h \) higher than the designed flow rate. In Fig 9, The head of centrifugal pump decreases gradually with the increase of flow rate, and the head D-value of turbulence model \( k-\omega \) and RNG \( k-\varepsilon \) is 48.032m at
the maximum flow rate. In Fig 10, Power curve is positively correlated with flow rate. With the increase of flow rate, the load of centrifugal pump shaft increases.

3.4. Impeller fluid velocity and pressure

As shown in Fig 11, the velocity of the blade working face is greater than that of the suction surface. The velocity increases gradually from the blade inlet, but there is a low speed zone on the suction surface of the impeller. From Fig 12, we can see that the pressure increases gradually from 0.5 MPa at the impeller inlet to 6.0 MPa at the impeller outlet, which conforms to the design pressurization law.

3.5. Volute section surface streamline

As shown in Fig 11, The velocity of the blade working face is greater than that of the suction surface. The velocity increases gradually from the blade inlet, but there is a low speed zone on the suction surface of the impeller. From Fig 12, we can see that the pressure increases gradually from 0.5 MPa at the impeller inlet to 6.0 MPa at the impeller outlet, which conforms to the design pressurization law.
4. Conclusion

From what has been clearly discussed above. There is the following conclusion:

(1) The flow rate of high-speed centrifugal pump with low specific speed should be properly increased to 35 m$^3$/h for practical use that is in order to improve efficiency.

(2) It should be controlled that the pump flow rate designed should not exceed 40 m$^3$/h in practical application prevent for more D-value.

(3) If only considered calculation time steps and convergence, $k$-$\varepsilon$ is the most suitable turbulence model to compare with design conditions and external characteristic (Efficiency, Head, Power).

(4) The velocity and pressure of turbulence models are the same in impeller. But SST and RNG $k$-$\varepsilon$ turbulence models can have more surface streamline details in the volute section more and more complex vortices and turbulences can be observed in the pump.

References

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