Numerical research on the effects of impeller pump-out vanes on axial force in a solid-liquid screw centrifugal pump

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A commercial CFD code has been used to predict the performance of a screw centrifugal pump with pump-out vanes, especially when changing regularity of impeller axial force based on the solid-liquid two-phase flow. The Unsteady Reynolds Averaged Navier-Stokes (URANS) approach has been applied to solve the unsteady, incompressible, three-dimensional turbulent. The SIMPLEC algorithm, standard wall functions and mix two-phase flow model were applied. The RNG $k \varepsilon$-model was used to account the turbulence effects. By changing the number of impeller pump-out vanes and width, six different screw centrifugal pump numerical simulation projects were given, and each scheme in the different solid volume fraction were calculated respectively. The change rules of axial force, velocity and pressure distribution of flow field were obtained on the different condition and different volume fraction. The results showed that the axial forces values based solid-fluid two-phase greater than based single-phase clear water, but both changing regularity of the axial force were consistent; as same condition, the same solid-phase volume concentration, with the increase of pump-out vanes number or width, the impeller axial force increased as well. Meanwhile the number of the pump-out vanes and the width of pump-out vanes in balancing the impeller axial force, there are the most optimal value.

1. Introduction

A screw centrifugal pump has superior performance that works against the blockage of solids because it has one three-dimensional spiral blade added to a conical hub cone and has a wide flow passage\textsuperscript{[1]} (Figure.1). Screw centrifugal pumps are now widely used in transporting high viscosity liquid containing large solids, fibrous fabric and other occasions \textsuperscript{[2]}. Impeller pump-out vane seals are non-contact hydrodynamic seals \textsuperscript{[3]}. Pump-out vanes can reduce the pressure of the seal, reduce the leak at the shaft seal, on the other hand it can reduce the solid particles into the shaft seal to protect the seal, and also can balance part of the axial force \textsuperscript{[4]}. Therefore it is particularly suitable for sealing liquid media containing solid particles, which has been widely applied in the slurry seal structure \textsuperscript{[5]}. 
As compared to single-phase flow, the complexity of solid-liquid two-phase flow increases significantly because the law governing the movement of particles in the pump channel is complicated [6]. The literature [7] studied the influences of variable-pitch blade on the screw centrifugal pump axial force under the solid-liquid two-phase flow medium, due to the calculation of the axial force in this study did not consider the part of the axial force on the back cover plate, therefore the variation of the axial force it got can not accurately reflect the axial force of the screw centrifugal pump under the solid-liquid two-phase flow medium. This research based on CFD (Computational Fluid Dynamics) calculated the axial force of the screw centrifugal pump with impeller pump-out vane at different solid volume fraction and different parameters of the impeller pump-out vanes.

2. Calculation Schemes and Research Method

2.1. Calculation Schemes
The main performance parameters of the screw centrifugal pump at the design point in this study are shown in Table 1. Figure 2 is the structure sketch of the screw centrifugal pump. Maintain the after chamber gap s=5mm unchanged, first keep the same number of impeller pump-out vanes, three different impeller schemes are designed by changing the width of pump-out vane; Second, keep the width of impeller pump-out vanes unchanged, the other three different impeller schemes are designed by changing the number of pump-out vanes, and the performance test of scheme two is carried out at the opening test rig. The specific schemes are shown in Table 2.

| Scheme | Numbers of pump-out vane/z | Width of pump-out vane/b |
|--------|---------------------------|-------------------------|
| Scheme 1 | 6                          | 2                      |
| Scheme 2 | 6                          | 3                      |
| Scheme 3 | 6                          | 4                      |
| Scheme 4 | 4                          | 3                      |
| Scheme 5 | 6                          | 3                      |
| Scheme 6 | 8                          | 3                      |

2.2. Research Method
Use water and sand water as working medium respectively, considering the interphase velocity slip, particle dispersion, phases coupling, in particular the effect of particles on fluid. Small particles for the solid-liquid two-phase flow turbulence model has better adaptability \cite{8}, and the collision between small particles and the interaction between the particles and the fluid seem relatively weak, it will make the numerical results more accurate and reasonable. What is more, in practical engineering application, the wear in fluid machinery caused by small particles cannot be under-estimated, the seal problems caused by the complex motion of small particles also need to focus on discuss and study, so the small particles are selected as research object, which median diameter is 0.076mm \((d_{50}=0.076\text{mm})\) and density \(\rho=2700\text{kg/m}^3\), and the solid volume fraction \(C_v\) is 5\%, 10\%, 15\% respectively. The screw centrifugal pump at each volume fraction is carried out respectively on 0.4\(Q_{\text{bep}}\), 0.6\(Q_{\text{bep}}\), 0.8\(Q_{\text{bep}}\), 1.0\(Q_{\text{bep}}\), 1.2\(Q_{\text{bep}}\) and 1.4\(Q_{\text{bep}}\) multiple conditions numerical calculation and analysis.

3. Model description and computational method

3.1 Basic Assumptions
The flow field in the impeller of the screw centrifugal pump is the complex solid-liquid two-phase flow, in order to ensure the accuracy of the numerical calculation of the two-phase flow in the flow channel, and now make the following assumptions:
- The fluid phase is incompressible fluid, the solid phase (dispersed phase) is continuous fluid, and the physical properties of each phase are constant;
- Solid phase particles are spherical and uniform size;
- Without considering the phase transition.

3.2. Calculation methods and boundary conditions
Using regional division grid technology, the computational domain is divided into five parts: the inlet tube area, the impeller region, the volute casing region, the pump-out vanes region and the after chamber clearance region (Figure 3). The total number of the grid is ultimately about 1million by the grid independence examination. Using MRF (Multiple reference frame) model, the continuity equation and the full three-dimensional incompressible Reynolds Navier-Stokes equations under the relative coordinate system are established; the equations are closed and simplified by RNG k-\(\epsilon\) model and selecting the Mixture model as the multiphase model.

![Figure 3. Grid of the computational domain](image)

![Figure 4. Experimental and numerical global performance characteristics](image)
Inlet condition: Using velocity inlet, the axial velocity was determined according to the mass conservation and imported no deformation assumption, supposing the solid concentration at the inlet is well-distributed; Wall condition: Turbulence was simulated by means of standard wall functions to calculate the flow variables near solid boundaries, supposed as adiabatic walls with a non-slip condition; Outlet condition: Using outflow condition at the outlet.

3.3. Analysis of the axial force

Based on the calculation theory of axial force of the centrifugal pump in the literature at home and abroad, combined with the structural characteristics of the screw centrifugal pump in this study, the axial force under solid-liquid two-phase flow can be attributed to the following main aspects:

- The axial force $F_1$ is generated by the asymmetric pressure distribution and different action area on the outer surface of the front and back cover plate.
- The axial force $F_2$ is generated by the pressure difference between suction surface and pressure surface of the impeller.
- The axial force $F_3$ is generated by the impeller hub. Therefore, the magnitude and direction of the axial force of the screw centrifugal pump under solid-liquid two-phase flow in this study is determined by the vector sum of $F_1$, $F_2$, $F_3$.

$$F = F_1 + F_2 + F_3$$

4. Experimental validation

The pump head and efficiency experimental values and calculated values with water medium are compared in Figure 4. The data shows that the calculated results have a good agreement with the experimental values, and the head errors do not exceed 5% at the design point and the efficiency errors are less than 4%, and the numerical efficiency is lower than the test value, which fully proves that the method used in this study has high reliability. When the solid-liquid two-phase flow in the impeller is simulated by the computational model and the three-dimensional model, the calculation results should have a certain degree of accuracy, and the method is capable of this study.

5. Effects of pump-out vane parameters on the axial force under solid-liquid two-phase flow

5.1 Effects of the number of pump-out vanes on the axial force

Figure 5 shows the variation of calculated axial force at different flow rate when the number of pump-out vanes is four, six and eight respectively under water corresponding to solid volume fraction $C_v=10\%$ two-phase flow. The axial force of $z=6$ and $z=8$ has almost the same variation at $0.6Q_{bep}$ to $1.4Q_{bep}$ under two-phase flow; and the axial force under solid-liquid two-phase flow is greater than the axial force under clean water at the same flow rate, and the direction of the axial force is always in the impeller inlet backward direction at $0.6Q_{bep}$ to $1.4Q_{bep}$. When the number of pump-out vanes is six, and the direction of the axial force is in the impeller inlet backward direction; The axial force values of $z=6$ is less than that of $z=8$ at the same flow rate, which indicates that the axial force has increasing trend with the increase of the number of pump-out vanes. The axial force under solid-liquid two-phase flow and under water medium have a similar trend for scheme four; the axial force decreases with the increase of flow rate at $0.4Q_{bep}$ to $0.6Q_{bep}$, which direction is the impeller inlet direction; however, the
axial force increases with the increase of flow rate at 0.6Q_{bep} to 1.4Q_{bep}, which direction is in the impeller inlet backward direction. The axial force value is 0.27N at 0.6Q_{bep}, the impeller almost does not bear the axial force effect, and this state is the best state for balancing the axial force to the pump-out vanes. The scheme five is the optimal scheme in the three schemes in this study.

5.2. Effects of the width of pump-out vanes on the axial force

Figure 6 shows the calculated axial force at different flow rate when the width of pump-out vanes is 2mm, 3mm and 4mm corresponding to the scheme one, two and three respectively under water and the solid volume fraction C_{v}=10% two-phase flow. The trend of the axial force with the flow rate under solid-liquid two-phase flow is basically identical with that under clean water, and the axial force increases with the increase of the flow rate at 0.6Q_{bep} to 1.2Q_{bep}, and the direction of the axial under the two mediums is in the same direction. The variations of the axial force at different width of the pump-out vanes are basically similar at 0.4Q_{bep} to 1.4Q_{bep} under solid-liquid two-phase flow. As small flow rate (0.4Q_{bep}), the axial force under two-phase flow is larger than that under clean water at the same flow rate, and the axial force increases with the increase of the flow rate at 0.6Q_{bep} to 1.2Q_{bep}, but the axial force has the decreasing trend at 1.2Q_{bep} to 1.4Q_{bep}, and the direction of the axial force is in the impeller inlet backward direction. The results show that the balancing effect corresponding to b=3mm is better than that corresponding to b=4mm. The axial force corresponding to b=2mm under water and two-phase mediums has the same trend; the magnitude of the axial force under solid-liquid two-phase flow is less than that under clean water at 0.4Q_{bep} to 0.6Q_{bep}, but the magnitude of the axial force under solid-liquid two-phase flow is greater than that under clean water at 0.8Q_{bep} to 1.4Q_{bep}; the direction of the axial force changes to the impeller inlet backward direction at 60% flow rate. From the comparison of the above three schemes of different width of pump-out vanes, it is easy to find that there exists best width of the pump-out vanes for balancing the axial force.

5.3. Effects of the solid concentration on the axial force

Figure 7 is the variation of the calculated axial force corresponding to z=6 and b=3mm at different flow rate under clean water and different solid volume fractions (5%, 10%, 15%). The Figure shows that the variation of axial force under different solid volume fraction is basically identical with that under clean water. The axial force increases with the increase of the flow rate between 0.6Q_{bep} to 1.2Q_{bep}, but the axial force has decreasing trend between 1.2Q_{bep} to 1.4Q_{bep}, and the direction of the
axial force is in the impeller inlet backward direction at $0.6Q_{bep}$ to $1.4Q_{bep}$, and the axial force increases with the increase of the solid volume fraction at the same flow rate.

![Figure 7. Variation of axial force with different solid volume fraction](image)

6. Conclusions

- The effect of pump-out vanes on balancing the axial force is obvious; the pump-out vanes not only can change the magnitude of the axial force but can change the direction of the axial force; there exists optimal values of the number and width of pump-out vanes for balancing the axial force.
- The axial forces under different solid volume fractions have almost the same variation with that under clean water, and the axial force increases with the increase of the solid volume fraction.

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