The Numerical Analysis of Screw-type Centrifugal Pumps with Different Rotary Impellers

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Abstract. Screw-type centrifugal pumps with different rotary layers have been numerically simulated. Internal flow fields of single-rotary impeller and double-rotary impeller are simulated by Reynolds Navier-Stokes function and \(k-\varepsilon\) viscous model. Stream lines, velocities, and pressure variations are calculated. The stream lines of impellers of two types show that the flow directions are significantly different. The flow distribution inside the pump chamber is approximately the same, and the flow rate is quite different. The pressure distribution of the single-rotary impeller is more uneven than the double-rotary impeller. With different rotary layers, the pressure range has a big difference. The mechanism that the internal flow field of single-rotary impeller pump differs from double-rotary impeller pump rotating layers causes such a large difference in the flow field is discussed.

1. Introduction

The invention of the screw-type centrifugal pump can be traced back to the 1960s. It was first used for port operations such as transporting live fish, and it is the most used fish pump now. It has strong anti-winding and anti-blocking ability and non-destructive ability to the transport medium. Nowadays, the screw-type centrifugal pump expands its range of applications, for example, it has been gradually applied in the fields of environmental protection, papermaking, aviation, etc. [1].

As the application of screw-type centrifugal pumps is continually expanding, the computational fluid dynamics (CFD) method is introduced into the research of the pump. A lot of research and discussion have been carried out by combining the characteristics of the internal flow field, pressure and, vorticity obtained by different calculation methods with the external characteristics and performance curves obtained by experimental means [2-7]. These researches mainly focused on a specific structural impeller of screw-type centrifugal pump, however, there are few works studies the effect of different structural impellers on the pump performance, which may be important in the development of different pumps. In this paper, a single-rotary impeller and a double-rotary impeller are introduced. The characteristics of
the internal flow field, such as streamlines, velocities, and pressure of the pumps with the two types of impellers are simulated by CFD method. The mechanism that how different structural impellers influence the characteristics is discussed.

2. Computational Methods

2.1. Control Equations

2.1.1. Governing Equation of Fluid Flow. The laws of mass and momentum are applied for compressive laminar and turbulence flow. All control equations are represented by a conservation law. The coordinate system used to describe the problem is the Cartesian coordinate system in the Euler description. When the flowing fluid is defined as a continuum, the governing equations used to solve are as follows.

Continuity equation (1):
\[
\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j} (\rho u_j) = 0
\] (1)

\( u \) is the instantaneous velocity in the \( j \) direction and \( \rho \) is the density of the fluid.

Momentum equation (2):
\[
\frac{\partial}{\partial t} (\rho u_j) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = - \frac{\partial p}{\partial x_j} + \frac{\partial \tau_{ij}}{\partial x_j} + \rho f_i
\] (2)

\( P \) is the static pressure, \( \tau_{ij} \) is the viscous stress tensor and \( f_i \) is the volume force. \( \tau_{ij} \) can be expressed as:
\[
\tau_{ij} = (\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}) - \frac{2}{3} \mu \left( \frac{\partial u_m}{\partial x_m} \right) \delta_{ij}
\] (3)

\( \mu \) is the hydrodynamic viscosity, \( \delta \) is the Kronecker function.

Bring (2) into (1) to get the Navier-Stokes equation:
\[
\frac{\partial}{\partial t} (\rho u_j) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = - \frac{\partial p}{\partial x_j} + \frac{\partial \tau_{ij}}{\partial x_j} + \frac{\partial u_m}{\partial x_m} \delta_{ij} + \rho f_i
\] (4)

2.1.2. Standard \( k-\varepsilon \) Turbulence Model. The most commonly used model in this study is the standard \( k-\varepsilon \) turbulence model. To estimate the velocity and length dimension in the standard \( k-\varepsilon \) turbulence model, turbulent flow energy transfer and dissipation equations are introduced, as shown below:

\[
\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_j} (\rho u_j k) = \rho P - \rho \varepsilon + \frac{\partial}{\partial x_j} \left( \mu + \mu_t \frac{\partial k}{\partial x_j} \right)
\] (5)

\[
\frac{\partial}{\partial t} (\rho \varepsilon) + \frac{\partial}{\partial x_j} (\rho u_j \varepsilon) = C_{e_1} \frac{\rho P \varepsilon}{k} - C_{e_2} \frac{\mu_k}{\varepsilon} + \frac{\partial}{\partial x_j} \left( \mu + \mu_t \frac{\partial \varepsilon}{\partial x_j} \right)
\] (6)

\( P \) is defined by equation (7):
\[ P = v_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \frac{\partial u_m}{\partial x_m} \delta_{ij} \right) \frac{\partial u_i}{\partial x_j} - \frac{2}{3} k \frac{\partial u_m}{\partial x_m} \delta_{ij} \] (7)

2.2. Geometric Model Parameters and Meshing

The parameters of the simulated pumps with single-rotary impeller and double-rotary impeller are volume flow \( Q = 80 \text{ m}^3/\text{h} \), head \( H = 10 \text{ m} \) and rotating speed \( n = 1450 \text{ r/min} \). The main parameters of the pumps with single-rotary impeller are listed as below.

**Table 1. Parameters of pumps with single-rotary impeller and double-rotary impeller.**

| Parameters                     | Single-rotary impeller | Double-rotary impeller |
|-------------------------------|------------------------|------------------------|
| Inlet diameter \( D_1 \)      | 160[mm]                | 110[mm]                |
| Outlet diameter \( D_2 \)     | 200[mm]                | 200[mm]                |
| Axial length \( L \)          | 180[mm]                | 160[mm]                |
| Rim side blade inclination \( \alpha \) | 40[°]                   | 40[°]                   |
| Hub side blade inclination \( \beta \) | 40[°]                   | 40[°]                   |
| Impeller outlet edge angle \( \theta \) | 15[°]                   | 15[°]                   |

An unstructured tetrahedral mesh is used through the domain. In the single-rotary model, the total nodes of channel 132387 and the total number of grids is 775472. In the double-rotary model, the total nodes of channel 124611 and the total number of grids is 712432. A grid independence check is performed. The calculating grids are as shown in figure 1.

**Figure 1.** Calculating grid of (a) pump with single-rotary impeller and (b) pump with double-rotary impeller.

**Results and Discussion**

The streamlines of two different pumps with single-rotary impeller and double-rotary impeller are shown in figure 2. The results show that the flow field in the chamber of both pumps is rotating around the impeller shaft, and no obvious vortex is observed. The screw-type centrifugal pump features both a volumetric pump and a centrifugal pump, and the impeller has a large wrap angle and axial length, so the fluids flow at a fairly gentle manner in the chamber. As the fluids flow around the shaft in the chamber, it becomes a vertical flow in the outlet channel. A vortex occurs in the outlet channel of the pump with the single-rotary impeller, which is not observed in the channel of the double-rotary impeller. Compared to the double-rotary impeller, the flow channel of the single-rotary impeller is relatively short.
and the rectifying ability to the liquid is relatively weak. Velocity in the chamber is larger than in the inlet and outlet with both pumps. This difference is smaller in the single-rotary impeller and more pronounced in the double-rotary impeller. To explain this phenomenon, the flow rate distribution of a particular horizontal plane in the pump chamber is taken.

![Figure 2. Streamlines of (a) pump with single-rotary impeller and (b) pump with double-rotary impeller.](image)

The flow rate distributions corresponding to a horizontal plane in the pumps with single-rotary impeller and double-rotary impeller are shown in figure 3. As can be seen, the maximum velocities of the flow field in the chamber are 18.7 m/s and 19.1 m/s of pumps with single-rotary impeller and double-rotary impeller. There is no significant between the two data. But as shown in figure 3(a), it indicates that the maximum velocity only appears at the edge of the impeller, and the velocity is much smaller in the other part of the chamber. There is a similar phenomenon in the pump with the double-rotary impeller, although the difference of velocities between the edge and chamber is relatively small. The mean velocity in the chamber of the pump with single-rotary impeller is 4.51 m/s, as well as the mean velocity in the chamber of the pump with double-rotary impeller is 9.76 m/s. Larger flow velocity means greater pumping capacity, which is more beneficial in practical applications.

![Figure 3. Velocity distribution in a horizontal plane of (a) pump with single-rotary impeller and (b) pump with double-rotary impeller.](image)
Pressure distribution of pump with single-rotary impeller and double-rotary impeller are shown in figure 4. The maximum pressure of single-rotary type is $9.94 \times 10^4$ Pa, and the maximum pressure of double-rotary type is $3.90 \times 10^4$ Pa, which is much smaller than the single-rotary type. Higher pressure represents a stronger pumping capacity, which is more conducive to the delivery of complex and viscous media. In contrast, higher pressure makes the transported material more susceptible to damage. As mentioned earlier, the velocity distribution inside the double-rotary type is more uniform than that of the single-rotary. Similarly, the pressure distribution inside the pump with the double-rotary impeller is more uniform than that of the single-rotary. The results may be due to that the flow channel of double-rotary type is longer than that of single-rotary type, therefore, the flow channel has a stronger binding force to the fluid.

**Figure 4.** Pressure distribution in a horizontal plane of (a) pump with single-rotary impeller and (b) pump with double-rotary impeller.

**Conclusion**

In this study, the internal flow fields of screw-type centrifugal pumps with single-rotary and double-rotary impellers were simulated by CFD method. Streamlines, velocity, and pressure distributions were figured and discussed. It indicates that the velocities and pressure of the pump with the single-rotary impeller are both smaller than that with the double-rotary impeller. A larger flow velocity means greater pumping capacity, however, a higher pressure may lead to damages on the pumping materials. Both pumps have a relatively stable internal flow fields and are suitable for the transport of living things.

**References**

[1] Tatebayashi Y, Tanaka K and Kobayashi T 2005 *J. Turbom.* Pump Performance Improvement by Restraining Back Flow in Screw-Type Centrifugal Pump, **127**(4) 755-62.

[2] Han W, Ma W, Li R and Li Q 2012 *Proced. Eng.* The Numerical Analysis of Radial Thrust and Axial Thrust in the Screw Centrifugal Pump, **31** 176-81.

[3] Cheng X and Li R 2012 *Proced. Eng.* Parameter equation study for screw centrifugal pump, **31** 914-21.

[4] Kim D J, Suh S H and Sung S K 1998 *The KSFM J. of Fluid Machin.* Effects of the Impeller Shapes on the Non-Clogging and the Screw-type Centrifugal Pump Performances, **1** 81-9.

[5] Hui Q, Li R, Su Q, Han W and Wang P 2015 *Adv. Mech. Eng.* Analysis on Energy Conversion of Screw Centrifugal Pump in Impeller Domain Based on Profile Lines, **5** 512-23.

[6] Zubanov V, Volkov A, Matveev V, Popov G and Baturin O 2018 *Proceed. Asme Turbo Expo: Turbomachin. Technic. Conf. Expos.* Optimization of Fuel Two-Stage Screw Centrifugal Pump of Rocket Powerful Turbopump Unit, **2B** V02BT44A020.
[7] Zubanov V M, Shabliy L S and Krivcov A V 2014 PROCEEDINGS OF THE ASME GAS TURBINE INDIA CONFERENCE CFD-Modeling of Powerful Screw Centrifugal Kerosene Pump, 1 V001T01A002.

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