The Distribution of Pressure and Vorticity of Impellers with Single-Rotary and Multi-Rotary

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Abstract. The screw-type centrifugal pumps with different impellers are studied. The structure of single-rotary and multi-rotary impellers are designed by the CAD method. The governing equation of fluid flow and SST k-ω model are applied to simulate the internal flow field by computational fluid dynamics method. The pressure distribution and vorticity distribution are calculated and analyzed. It is found that the pressure distribution and vorticity distribution of the two types of impellers are significantly different. The pressure distribution of single-rotary impeller is relatively uniform. The vorticity of single-rotary impeller is larger than multi-rotary impeller.

1. Introduction
In the 1960s, Swiss engineer Martin Sthale invented a screw-type centrifugal pump equipped with a screw-type impeller and used it to transport live fish in the port [1]. The impeller of the screw-type centrifugal pump consists of two parts, the screw-type section and the centrifugal section. The inlet part is a screw-type impeller and the outlet part is similar to a mixed flow impeller. Its unique structure combines the screw propelling of the impeller with the centrifugal effect to enable the medium to obtain energy. It has both advantages of the screw-type pump and the centrifugal pump [2]. It is widely used in many different fields of application.

As the application of screw-type centrifugal pumps is continually expanding, the computational fluid dynamics (CFD) method is applied to study the internal characteristics of the pump [3, 4]. As the external characteristics can be obtained by the experimental methods, the internal characteristics are mostly obtained by the simulation methods. Han [5] analyzed the radial thrust and axial thrust in the screw centrifugal pump and found that the occurrence of minimum pressure value is at the timing that the maximum impeller radius just turned the tongue of the volute. Cheng [6] studied the parameter equation of the screw-type centrifugal pump, and give the regularity for change of pitch in the variable-pitch parametric equation group. Kim [7] analyzed the effects of the impeller shapes on the Screw-type Centrifugal Pump Performances. They found that as the shapes of vanes change, the total head, efficiency, and power of the screw-type pump show no significant change. Zubanov [8] optimized a two-stage centrifugal pump, which consists of low-pressure screw centrifugal pump and high-pressure centrifugal pump.
In this study, the internal characteristics of impellers of screw-type centrifugal pump with single-rotary and multi-rotary are simulated by the CFD method. The pressure distribution and vorticity distribution are calculated and analyzed by the governing equation of fluid flow and SST $k$-$\omega$ model. The differences between the single-rotary and multi-rotary impellers are 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:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j} (\rho u_j) = 0 \tag{1}$$

$u_j$ is the instantaneous velocity in the $j$ direction and $\rho$ is the density of the fluid.

Momentum equation:

$$\frac{\partial}{\partial t} (\rho u_j) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = -\frac{\partial P}{\partial x_j} + \frac{\partial}{\partial x_j} (\mu \frac{\partial u_i}{\partial x_j}) + \rho f_i \tag{2}$$

$P$ is the static pressure, $\tau_{ij}$ is the viscous stress tensor and $f_i$ is the volume force. $\tau_{ij}$ can be expressed as:

$$\tau_{ij} = \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu \frac{\partial u_m}{\partial x_m} \delta_{ij} \tag{3}$$

$\mu$ is the hydrodynamic viscosity, $\delta$ is the Kronecker function.

Bring (2) into (1) to get the Navier-Stokes equation:

$$\frac{\partial}{\partial t} (\rho u_j) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = -\frac{\partial P}{\partial x_j} + \frac{\partial}{\partial x_j} (\mu \frac{\partial u_i}{\partial x_j}) - \frac{2}{3} \mu \frac{\partial u_m}{\partial x_m} \delta_{ij} + \rho f_i \tag{4}$$

2.1.2. SST $k$-$\omega$ Model. SST $k$-$\omega$ Model was proposed in 1993 by F.R. Menter [9]. As the Wilcox $k$-$\omega$ model is accurate in the near-wall region and the $k$-$\varepsilon$ model is suitable for freestream, this model combines these advantages of the Wilcox $k$-$\omega$ model and the $k$-$\varepsilon$ model. The $k$-$\varepsilon$ model is transformed into a $k$-$\omega$ model. The two models are weighted averaged by function $F$ and then added to obtain the BSL $k$-$\omega$ model. Based on the BSL $k$-$\omega$ model, the SST $k$-$\omega$ model modified the definition of the eddy-viscosity for adverse pressure gradient boundary-layer flows.

Original $k$-$\omega$ model and transformed $k$-$\varepsilon$ model are weighted and averaged to obtain the following model:

$$\frac{\partial}{\partial t} (\rho k) = \tau_{ij} \frac{\partial u_i}{\partial x_j} - \beta^* \rho \omega k + \frac{\partial}{\partial x} \left[ (\mu + \sigma_{k1} \mu_t) \frac{\partial k}{\partial x} \right] \tag{5}$$

$$\frac{\partial}{\partial t} (\rho \omega) = \frac{\gamma}{\nu_t} \tau_{ij} \frac{\partial u_i}{\partial x_j} - \beta \rho \omega^2 + \frac{\partial}{\partial x} \left[ (\mu + \sigma_{\omega} \mu_t) \frac{\partial \omega}{\partial x} \right] + 2 \rho \left( 1 - F_t \right) \sigma_{\omega} \omega \frac{1}{\omega} \frac{\partial k}{\partial x} \frac{\partial \omega}{\partial x} \tag{6}$$

Where,
\[ \mu_t = \frac{\rho a_1 k}{\max (a_1, \omega F_2)}, \quad v_t = \frac{\mu_t}{\rho} = \frac{a_1 k}{\max (a_1, \omega F_2)}, \quad F_2 = \text{tangarg}^2, \quad \text{arg}_2 = \max \left( \frac{\sqrt{1}}{\beta \omega \omega' \omega''}, \frac{500 \nu}{d_2 \omega} \right), \quad F_1 = \text{tanharg}^4, \quad \text{arg}_1 = \min \left[ \max \left( \frac{\sqrt{1}}{\beta \omega \omega' \omega''}, \frac{500 \nu}{d_2 \omega} \right), \frac{\rho a_2 k}{C_{D, \omega}} \right], \quad C_{D, \omega} = \max \left( \frac{\rho a_2 k}{\omega}, \frac{\partial k}{\partial \omega}, \frac{\partial \omega}{\partial \omega'} \right) \times 10^{-20} \]

\[
\beta^*, \beta, \gamma, \sigma_1, \sigma_2, \text{ and } a_1 \text{ are empirical constants. For any constant } \phi, \text{ there is}
\]

\[
\phi = F_1 \phi_1 + \left( 1 - F_1 \right) \phi_2 \quad (7)
\]

The subscript 1 indicates the constant value in the inner layer, and the subscript 2 indicates the constant value in the outer layer.

### 2.2. Geometric model parameters

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 below.

| Parameters | Single-rotary impeller | Multi-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[°] |

### 3. Results and Discussion

The pressure distributions of single-rotary and multi-rotary impellers are shown in Fig. 1. It can be found that the pressure distribution of single-rotary impeller is relatively uniform comparing with the multi-rotary impeller. From the inlet of the impeller to the outlet, the pressure gradually increases, which is similar to the screw pump. Single-rotary impeller has a lower differential pressure between inlet and outlet than multi-rotary impeller, which leads to a smooth flow. The pressure near the shaft is lower than at the edge and gradually increases along the radial direction in both impellers. This is consistent with the centrifugal effect of the screw-type centrifugal pump.

**Figure 1.** The pressure distributions of single-rotary and multi-rotary impellers (in Pa)
In order to numerically discuss the variations of the pressure distribution, the distribution curve is shown in Fig. 2. The data is taken along a radial direction at the rotary near outlet. The lowest pressure of single-rotary impeller is $0.52 \times 10^4$Pa, as well as the highest pressure is $3.23 \times 10^4$Pa. The lowest pressure of multi-rotary impeller is $3.87 \times 10^4$Pa, as well as the highest pressure is $9.21 \times 10^4$Pa. The differential pressure between shaft zone and edge zone forced the pumped materials to the shaft zone, as the shaft zone is smooth enough to prevent the materials from damage. The pressure of multi-rotary is much larger than single-rotary, which means the flow has stronger impact.

![Figure 2. Pressure distributions of (a) single-rotary and (b) multi-rotary impellers](image)

The distribution of vorticities of single-rotary and multi-rotary impellers are shown in Fig. 3. The vorticity distribution is uniform in both impellers, which indicates that there is no obvious vortex. As the vorticity at the edge of multi-rotary impeller is much larger than other regions, vortex appears at the edge, while there is only a slight increase at the edge of single-rotary impeller. This means that the vortex is smaller in the single-rotary, which benefits the safety of material pumping.

![Figure 3. Vorticity distributions of (a) single-rotary and (b) multi-rotary impellers (in 1/s)](image)

4. Conclusion

SST $k-\omega$ Model is applied to simulate the characteristics of single-rotary and multi-rotary impellers of screw-type centrifugal pump. Single-rotary impeller has a lower pressure than multi-rotary impeller. The differential pressure of single-rotary impeller is smaller than that of multi-rotary impeller. The vorticity distributions are uniform in both models.
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