Numerical Analysis of Stress and Deformation of Impeller in the Mixed Flow Pump Based on Fluid-structure Interaction

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Abstract. To improve the operational reliability of mixed flow pumps, the two-way coupling fluid-structure interaction method is applied to investigate stresses and deformations of the impeller. The Reynolds-averaged Navier–Stokes equation, coupled with the SST k-ω turbulent model, is solved in the fluid domain by using ANSYS CFX, while transient structure dynamics in the structural domain are calculated with the finite element method. The accuracy of the numerical results has been verified by experimental pump performance. The results show that the flow rate definitely has a noticeable effect on both deformations and stresses. With the flow rate increasing, the impeller deformation decreases, as well as the stress, which is completely different from centrifugal pumps, but similar to axial pumps. Moreover, a local maximum deformation occurs in the middle of blade inlet, and the maximum stress of the entire impeller fluctuates periodically. On the same shroud, the stress on the suction side is higher than the pressure side along the intersection path; and on the same side, the mean stress on the rear shroud is larger than the front shroud. This study can help understand the distribution of both deformations and stresses in the impeller and provide guidance for improving the operation safety of mixed flow pumps.

Introduction

Mixed flow pumps are characterized by relatively large flow rate and high head in comparison with centrifugal pumps and axial flow pumps, which are widely used in nuclear power plants, seawater desalination systems, et al. The complex internal flow in mixed flow pumps, such as rotor-stator interaction and cavitation, generates hydraulic excitation forces and pressure fluctuations, which may result in mechanical vibrations and alternating stresses on pump components [1]. These flow-induced vibrations increase the working risk of mixed flow pumps, thus it is important to investigate the interaction between unsteady flow fields and stationary structures.

Previously, researchers have conducted some investigations involving the fluid-structure interaction (FSI) for pumps. Benra [2] investigated the flow-induced vibrations of a single-blade pump by using numerical simulations and experimental methods. Kobayashi [3] evaluated the largest stress of a mixed flow pump with unshrouded impeller by using one-way coupling fluid-structure simulation. Campbell et al. [4] analyzed the deformation of an airfoil in turbomachinery based on FSI, and then compared experimental and simulated results. Pei et al. [5] applied two-way coupling FSI to study the effects of impeller deformation on flow field in a centrifugal pump and the dynamic behavior of rotor. However, few research papers on FSI in mixed flow pumps have been published, and their structural responses are still unclear. In this study, the two-way coupling of the flow calculation (CFD) and the solid calculation (CSD) is applied to capture the interaction process. Since the impeller is the only rotating component with high risk, the deformation and stress of impeller at different working conditions are analyzed in detail for improving the operation safety.

Numerical Simulation and Experimental Validation

Pump Model

Table 1 lists the main geometric parameters of the mixed flow pump. The rotating speed is \( n = 1480 \) r/min and the specific speed \( n_s = 3.65nQ^{0.5}/H^{0.25} \) is 386 at the design condition, which is known as a mixed flow pump. The 3-D model is showed in Figure 1.
Table 1. Main geometric parameters.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Impeller inlet diameter, $D_1$ (mm)</td>
<td>717.5</td>
</tr>
<tr>
<td>Impeller outlet diameter, $D_2$ (mm)</td>
<td>804</td>
</tr>
<tr>
<td>Impeller outlet width, $b_2$ (mm)</td>
<td>212</td>
</tr>
<tr>
<td>Number of impeller blades, $Z_i$</td>
<td>4</td>
</tr>
<tr>
<td>Diffuser inlet width, $b_1$ (mm)</td>
<td>241</td>
</tr>
<tr>
<td>Number of diffuser blades, $Z_d$</td>
<td>11</td>
</tr>
<tr>
<td>Circular casing inlet diameter, $D_5$ (mm)</td>
<td>1340</td>
</tr>
</tbody>
</table>

Figure 1. 3-D MODEL.

**CFD Settings**

The fluid domains of the whole flow passage are composed of suction pipe, impeller, diffuser and circular casing, whose structure meshes are generated by using the software ICEM-CFD, and approximately 10.7 million grids are selected to conduct the calculation through grid independency analysis, as Figure 2 (a) shows. The Reynolds-averaged Navier-Stokes equation is solved with the shear stress transport (SST) $k$-$\omega$ turbulence model by using ANSYS CFX. The total pressure and mass flow rate are employed in the inlet and outlet respectively. The time step is chosen to correlate to an impeller rotation of 1 degree and the total calculation time is 10 revolutions.

![Figure 2](image)

**CSD Settings**

The impeller structure is modeled with isotropic material behavior and made from duplex stainless steel, showed in Table 2. The mesh in the structural domain contains about 0.3 million quadratic tetrahedral elements, as Figure 2 (b) shows. Fixed support are applied to the impeller hub. The fluid-structure interfaces for all wetted surfaces, blades and shrouds, are defined. The total FSI calculation time is chosen to correlate 3 revolutions.

![Figure 2](image)

**Experimental Validation**

Figure 3 shows the open test rig of the mixed flow pump, whose testing precision is superior to the national grade 1 in China. The model pump is manufactured instead of the prototype, and the linear
ratio of the prototype to the model is 5.74 according the affinity law. In order to verify the accuracy of numerical simulations at the same scale, the flow rate coefficient \( \varphi \) and head coefficient \( \psi \) are defined as the following equations showed.

\[
\varphi = \frac{4Q}{\pi^2 nd_2^3}
\]

\[
\psi = \frac{2gH}{\pi^2 d_2^2 n^2}
\]

Here, \( Q \) is the flow rate, \( n \) is the rotating speed of impeller, \( d_2 \) is the outer diameter of impeller, \( H \) is the head. Figure 4 shows the comparison of experimental and simulated head curves. The simulated head presents a good agreement with the experiments, and the head deviation at the design condition (dimensionless flow rate is 0.19) is 4.08%. Overall, the maximum relative deviation is less than 5%. Therefore, the numerical simulation results are reliable.

Results and Discussions

Deformation Analysis

The dimensionless displacement coefficient \( d^* \) and the dimensionless equivalent stress coefficient \( \sigma^* \) are defined as the following equations showed:

\[
d^* = \frac{d_{def}}{d_2}
\]

\[
\sigma^* = \frac{\sigma_{eqv}}{0.5 \cdot \rho \cdot u_2^2}
\]

Where \( d_{def} \) is the absolute value of displacement, \( \sigma_{eqv} \) is the equivalent stress, \( \rho \) is the density of working mediums, \( u_2 \) is the impeller outlet velocity.
Figure 5 shows the impeller deformation at three working conditions. On the left-hand side, the deformation is showed as viewed on the impeller inlet, and on the right as viewed on the rear shroud. Overall, the maximum of displacement decreases with the increasing flow rate; the maximum displacement at the part load is largest, nearly twice the maximum displacement at the overload condition, which is totally different from centrifugal pumps but similar to axial pumps. As the flow rate rises, the displacement field trends to be obviously uneven. At each working condition, a local maximum occurs in the middle of blade inlet; the displacement on the blade enlarges from the rear shroud to the front shroud; the displacement of front shroud is much larger than rear shroud, and the rear shroud presents to be almost changeless.

Stress Analysis

A qualitative investigation of impeller stresses indicates that the critical regions of large stress are the intersection paths between blades and shrouds. Figure 6 shows the intersection paths. The intersection path between the pressure side (PS) of front shroud and the blade is marked as Front-ps, et al. Besides, 0 presents the leading edge and 1 is the trailing edge.

Figure 7 shows the dimensionless stress distribution along the intersection paths. Overall, the stress along the same path decreases with the flow rate rising, but in the similar trend; on the same shroud, the stress on the suction side (SS) is higher than the PS, especially the local maximum stress; on the same side, the mean stress on the rear shroud is larger than the front shroud, but the local maximum stress on the rear shroud is smaller than the front shroud. On the front shroud, the stress along the Front-PS peaks at the leading edge, but along the Front-SS it peaks at the trailing edge. By contrast, on the rear shroud, the maximum stress occurs at the latter path.
Figure 7. Dimensionless stress distribution along intersection paths between blades and shrouds.
Figure 8 shows the time-history results of dimensionless maximum stress $\sigma_{\text{max}}$ of the whole impeller during one revolution. In general, the maximum stress decreases with the increasing flow rate, which is the same as above. Moreover, obvious periodic fluctuations of the maximum stress can be found, and there are 11 peaks and 11 valleys during one revolution, corresponding to the number of diffuser blades.

![Figure 8. Time-history results of dimensionless maximum stress during one revolution.](image)

**Conclusions**

In this study, the deformations and stresses of impeller in the mixed flow pump are calculated by means of two-way coupling fluid-structure interaction. And the accuracy of the numerical results has been verified by experimental pump performance. Some conclusions can be drawn:

(a) The impeller deformation decreases with the flow rate increasing, and a local maximum deformation occurs in the middle of blade inlet.

(b) As the flow rate rises, the stress along the intersection paths between shrouds and blades decreases, but in the similar trend.

(c) The maximum stress of the entire impeller fluctuates periodically and reduces with flow rate rising.

**Acknowledgments**

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**Reference**


