Fluid structure interaction dynamic analysis of a mixed-flow waterjet pump

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Abstract. In order to avoid resonance of a mixed-flow waterjet pump at run time and calculate the stress and deformation of the pump rotor in the flow field, a one-way fluid structure interaction method was applied to simulate the pump rotor using ANSYS CFX and ANSYS Workbench software. The natural frequencies and mode shapes of the pump rotor in the air and in the flow field were analyzed, and the stress and deformation of the impeller were obtained at different flow rates. The obtained numerical results indicated that the mode shapes were similar both in the air and in the flow field, but the pump rotor’s natural frequency in the flow field was slightly smaller than that in the air; the difference of the pump rotor’s natural frequency varied lightly at different flow rates, and all frequencies at different flow rates were higher than the safe frequency, the pump rotor under the effect of prestress rate did not occur resonance; The maximum stress was on the blade near the hub and the maximum deformation on the blade tip at different flow rates.

1. Introduction

A waterjet propulsion system does not directly produce thrust, but it passes reactive force of waterjet pump which jets water at the outlet to push the ship forward, and it is a special kind of ship propeller unlike conventional screw propellers. Nowadays, there is a growing interest on waterjet propulsion which has many benefits such as high propulsion efficiency, good maneuverability, little vibration, low noise, strong ability of anti-cavitation, simple transmission system and good reliability of blades. These advantages have increased the demand for waterjet propulsion systems in a variety of marine vehicles including high-speed ship [1].

The internal flow of a mixed-flow waterjet pump is very complex, and pressure fluctuation caused by unsteady flow is the main reason of vibration and noise, and the pressure fluctuation may result in a resonance to generate the exciting force. A modal analysis of the pump rotor can avoid resonance based on one-way fluid structure interaction (FSI) method, and a structure analysis provides the basis for the impeller design. Application of FSI method has been accepted extensively in the flow field, and it is important to reliability research of the hydraulic turbine [2-4]. Application weak coupling and strong coupling optimized the vibration characteristic of single channel pump by Benera et al. [5]. A combined calculation for turbulent flow and structure response of impeller was first established using two-way coupling method to study the effect of FSI of impeller in centrifugal pump by Pei J et al. [6-7]. The external characteristics and internal flow features of a diffuser pump were analyzed with the two-way flow structure coupling method to study the inner mechanism of the FSI effect on the external characteristic by Liu H L et al. [8]. Recently, FSI method is mainly applied to the internal flow field and structure analysis, and it is rarely used in the modal analysis of the pump rotor [9]. Therefore, in order to improve the operation stability of a mixed-flow waterjet pump, it is very necessary to analyze FSI dynamic characteristic of the pump rotor.
2. Numerical calculation

2.1. Geometry model
In this paper, the model is a mixed-flow waterjet pump, including impeller, guide vane and nozzle, of which specific speed is 455. Three-dimensional diagram is shown in Figure 1. The design parameters and geometric dimensions are as follows. The rotational velocity is 4500 r/min, with design flow rate of 500 m$^3$/h and head of 32 m. The impeller is enclosed with 4 blades, and the guide vane with 7 blades. The inlet diameter of impeller is 156 mm, and nozzle diameter is 80 mm.

![Figure 1. Three-dimensional diagram.](image1)

2.2. Mesh generation
Fluid calculation domain of a mixed-flow waterjet pump generates structured mesh using ICEM CFD, as shown in Figure 2. Inter-region is connected to the interface technology by region mesh generation method. The model is divided into calculate region mesh, which include inlet extension, impeller, guide vane, nozzle and outlet extension. A total of fluid calculation domain mesh is 1,476,053. The structure calculation domain mesh is generated automatically by Workbench mesh functions, and a total of structure calculation domain is 301,286, as shown in Figure 3.

![Figure 2. Fluid domain mesh.](image2)

![Figure 3. Structure domain mesh.](image3)

2.3. Numerical approach and boundary conditions
The internal flow of a mixed-flow waterjet pump is calculated numerically by a standard k-ε turbulence model and a scalable wall function. The governing equations are described by finite volume method based on the finite element, and the convective terms use a high resolution format. The fluid domain of a mixed-flow waterjet pump is solved by means of the commercial CFD code ANSYS CFX 12.1 with three dimensional, unsteady incompressible Reynold-averaged Navier-Stokes equation. A constant speed value is imposed at the fluid inlet and a constant pressure value is imposed at the pump outlet. A no slip boundary condition is imposed at the impeller blades, guide vanes and pump casing wall, and those walls formed by impeller are defined as a rotating boundary condition. A rotational speed of 4500 r/min is imposed to the blade impeller. Flow parameter
conversion of rotating and static coordinate system deals with the Frozen Rotor. The mesh parallel uses GGI method.

The structure domain of a mixed-flow waterjet pump is calculated by FSI method using ANSYS Workbench. The material of impeller and shaft are cast aluminum and stainless steel respectively, and the properties of the pump rotor material are shown in Table 1. This model provides three interfaces of the fluid and structure domains, including an interface of the pump rotor and the impeller blade, an interface of the pump rotor and the impeller hub, an interface of the pump rotor and the pump shaft.

| Name          | Density $\rho/(kg/m^3)$ | Elastic modulus $E/\text{Pa}$ | Poisson ratio $\mu$ | Yield strength $\sigma_s/\text{MPa}$ |
|---------------|-------------------------|-------------------------------|---------------------|-------------------------------------|
| Stainless steel | 7850                    | $2.0 \times 10^{-11}$         | 0.3                 | 250                                 |
| Cast aluminum  | 2770                    | $7.1 \times 10^{-10}$         | 0.33                | 280                                 |

3. Results and analysis
Under different conditions, comparison simulated and experimental results of head and efficiency are shown in Figure 4, when a mixed-flow waterjet pump operates at the steady flow. The results show that the maximum head difference which the pump is at same condition is 1.56m and the relative error is 2.8%, and then the maximum efficiency difference which the pump is at same condition is 1.67% and the relative error is 2.8%. Overall, the variation trend of simulated performance curve is basically consistent with experimental result, indicating that the numerical simulation method can accurately calculate this model external characteristic. At the same time, the results show that the numerical simulation method is reasonable to fluid structure interaction dynamic analysis.

Figure 4. Performance curve of a mixed-flow waterjet pump.

3.1. Modal analysis of the pump rotor in the air
In this paper, the design rotational velocity is 4500 r/min, and the rotational velocity of 0 r/min, 1000 r/min, 2000 r/min, 3000 r/min, 4000 r/min, 4500 r/min are calculated respectively because the rotational velocity of a mixed-flow waterjet pump increases from 0 r/min to 4500 r/min on startup. Six lowest modal frequencies are compared, and the results can meet demands in the engineering application. Figure 5 is a Campbell diagram of the pump rotor, and X-axis is the rotational velocity, and Y-axis is the frequency. A straight line of larger slope is the excitation frequency of the pump rotor. The first order natural frequency of the pump rotor should higher than the safe frequency of 330Hz which is acquired by blade frequency of 300Hz and safety factor of 10%. As can be seen from Figure 5, the rotational velocity of the pump rotor has little effect on the frequency, and the excitation
frequency does not have an intersection with the natural frequency, so the pump rotor does not occur resonance.

![Campbell diagram of the pump rotor.](image)

**Figure 5.** Campbell diagram of the pump rotor.

Six lowest order modal shapes of the pump rotor in the air are shown in Figure 6. The first and second order mode shapes show the bending vibration of the pump shaft, and the maximum deformation is at middle part of the pump shaft. The third order mode shape shows the torsional vibration of the impeller and pump shaft, and the maximum deformation is at the impeller blade tip. The fourth and fifth order mode shapes are similar to that of the third order, but the maximum deformation is at two groups of symmetry blades. The sixth order mode shape shows the torsional vibration of the impeller and the bending vibration of the pump shaft.

![Six lowest order mode shapes of the pump rotor in the air.](image)

**Figure 6.** Six lowest order mode shapes of the pump rotor in the air.

3.2. Modal analysis of the pump rotor in the flow field

**Table 2.** The frequencies of the pump rotor in the air and in the flow field

| Mode | \(f_{\text{air}}\) (Hz) | \(f_{0.68Q}\) (Hz) | \(f_{0.85Q}\) (Hz) | \(f_{1Q}\) (Hz) |
|------|-----------------|-----------------|-----------------|-----------------|
| 1    | 377.37          | 369.47          | 369.46          | 369.46          |
| 2    | 380.00          | 369.77          | 369.76          | 369.75          |
| 3    | 630.35          | 629.48          | 629.41          | 629.36          |
The prestress will affect the modal frequency and mode shape of the pump rotor, and the model proceeds modal analysis under different prestress. The frequencies of the pump rotor in the air and in the flow field are shown in Table 2. As can be seen from Table 2, the pump rotor’s frequency in the flow field is lower than that in the air, but the frequencies under different flow rates have very little difference, and it does not indicate that the prestress have no affect on the natural frequency. In addition, the pump rotor under prestress will not occur resonance because the frequencies at different flow rates are higher than the safe frequency of 330Hz.

Only the mode shape under design flow rate is analyzed due to space limitation, and six lowest order mode shapes of the pump rotor at $1.0Q$ condition are shown in Figure 7. Six lowest order mode shapes at $1.0Q$ condition are basically similar with these in the air.

![Figure 7. Six lowest order mode shapes of the pump rotor at $1.0Q$ condition.](image)

### 3.3. Stress and deformation of the pump rotor in the flow field

Dynamic characteristic of the impeller is analyzed under different flow flows, and the stress, strain and total deformation of the impeller are shown in Table 3. As can be seen from Table 3, the maximum stress and strain difference of the impeller is large under different flow rates, and the maximum stress at $0.68Q$ condition is 0.64 times of $1.0Q$ condition, and the maximum strain at $0.68Q$ condition is 1.83 times of $1.0Q$ condition, and the maximum total deformations are basically same under different conditions. The maximum stress is 57.04MPa at $0.68Q$ condition, and it is less than allowable stress of the impeller material. In addition, the maximum total deformation is 0.21 mm at $0.85Q$ condition, and the tip clearance is 0.5 mm in this model, so the impeller cannot contact with pump casing due to deformation.

| Flow  | Maximum stress/MPa | Minimum stress/MPa | Maximum strain/mm | Minimum strain/mm | Maximum total deformation/mm |
|-------|-------------------|-------------------|-------------------|-------------------|-----------------------------|
| 0.68Q | 36.77             | 0.01              | 5.24e-4           | 4.80e-8           | 0.20                        |
| 0.85Q | 45.46             | 0.01              | 3.27e-4           | 4.82e-8           | 0.21                        |
Total deformation and equivalent stress of the impeller at different flow rates are shown in Figure 8. As can be seen from Figure 8, the maximum total deformation of the impeller is on the blade tip at different flow rates due to the action of centrifugal force. However, the maximum total deformation is on the blade outlet edge at 0.68Q condition, and on the blade inlet edge at 0.85Q condition and 1.0Q condition. In addition, the maximum equivalent stress is on the blade near the hub at different flow rates.

![Figure 8. Total deformation and equivalent stress of the impeller at different flow rates.](image)

Row 1: Total deformation. Row 2: Equivalent stress.

4. Conclusion

The natural frequency of the pump rotor is basically the same in the air and in the flow field, and the frequency in the flow field is lower than in the air. Moreover, the reduced amplitude is mainly related with the vibration modes, and it is 1% to 5% in this paper. The first and second order mode shapes and modal frequencies are similar with the third and fourth order, because the structure and boundary conditions of the pump rotor are symmetrical. Thus the two mode shapes in orthogonal direction will happen, and they have the same frequency and different phase. Deformation of the impeller at 0.68Q condition is different from those at 0.85Q and 1.0Q conditions, as the impeller blade outlet near the guide vane inlet will have greater backflow which can lead to larger deformation at the impeller blade outlet. Through FSI dynamic analysis of the pump rotor, designers have to consider how to accurately calculate the natural vibration characteristic of the pump rotor, and it is important to avoid resonance.

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