Numerical study on pressure fluctuation in a multiphase rotodynamic pump with different inlet gas void fractions

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Abstract. Pressure fluctuation in the single-phase pump has been widely discussed, while less attention was paid for that in the multiphase pump. Therefore, this study investigated the pressure fluctuation in a multiphase rotodynamic pump. Unsteady simulations based on Euler two-fluid model were done with ANSYS-CFX software when the inlet gas void fractions (IGVF) were 0.0%, 3.0%, 9.0%, and 15.0%. Under pure water (IGVF=0.0%) and gas-liquid two-phase (IGVF=3.0%, 9.0%, 15.0%) conditions, the reliability of numerical method was verified by comparison with the experimental data. The numerical results showed that the dominant frequency in the impeller and guide vane corresponded to the blade numbers of guide vane and impeller, respectively. This illustrated that the rotor-stator interaction was the main cause for the generation of fluctuation in such pump. Meanwhile, the maximum fluctuation amplitude appeared near the impeller outlet at IGVF=0.0%, while it appeared near the guide vane inlet under two-phase conditions. Furthermore, the maximum fluctuation amplitudes at IGVF=3.0%, 9.0%, 15.0% are 1.9, 2.9, and 3.3 times as large as that at IGVF=0.0%.

1. Introduction

Pressure fluctuation in the single-phase pump has been widely discussed. It not only aggravates the vibration and noise of unit, but also causes the cavitation. Therefore, the unit will operate instability, or even be destroyed[1-3]. Compared with the single-phase pump, the gas-liquid two-phase pump is often accompanied with the coalescence and breakup of bubbles, and the separation and mixing among the phases, thus the pressure fluctuation will come into being[4].

The experimental and numerical methods are adopted to investigate the pressure fluctuation in the pump. For the single-phase pump, the relevant experimental study is usually carried out by monitoring the pressure variation in the shroud and hub of pump[5]. However, for the two-phase pump, the experimental study is mainly focused on the external and internal characteristics[6]. Therefore, the shell of impeller and guide vane is usually made of organic glass[7], which makes some difficulties for monitoring the pressure variation in the multiphase pump.

With the technological development of Computational Fluid Dynamics(CFD), CFD simulation has become a convenient and effective method to investigate the pressure fluctuation in the multiphase pump[8]. Inlet gas void fraction (IGVF) is one of the important factors that affect the complex flow in such pump. What’s more, with the increase of IGVF, various flow patterns, the gas accumulation and flow separation will come into being[9], thus resulting in the variation of pressure fluctuation. Zhang et al investigated the pressure fluctuation in a multiphase pump and found IGVF has a great influence on
the dominant frequency, but little effect on the second dominant frequency[10]. Our group has also analyzed the pressure fluctuation in such pump, and obtained that the pressure fluctuation in guide vane is greater than that in impeller[11, 12]. However, the relationship between IGVF and pressure fluctuation, as well as the difference of pressure fluctuation between single-phase pump and multiphase pump, is still not clear enough.

On the whole, few studies have focused on the pressure fluctuation in the multiphase pump. Therefore, the characteristic and mechanism of pressure fluctuation in such pump still need to be explored. Based on Euler two-fluid model, the unsteady flow in a multiphase rotodynamic pump was simulated with ANSYS-CFX software. The characteristics of pressure fluctuation between pure water and two-phase flow conditions were analyzed. Meanwhile, the influence of IGVF was investigated to reveal the generation mechanism of pressure fluctuation in such pump.

2. Model parameters
The designed parameters of the pump with air-water two-phase flow are as follows: the rotational speed is 2950rpm; the head is 15m; and the flow rate is 50m$^3$/h. Figure 1 shows the test model including the inlet and outlet pipes, impeller with four blades, and guide vane with eleven blades. Furthermore, the shell of impeller and guide vane is made of organic glass to observe the internal flow in the pump.

3. Numerical methods
3.1. Governing equations
Two-fluid model is widely used to simulate the two-phase flow in the multiphase pump because of its high computational accuracy[13]. In this model, each medium has its own conservation equation and movement law. Moreover, their equations are coupled through the phase interaction. Therefore, the two-fluid model is adopted in this study.

The unsteady Reynolds-averaged Navier-Stokes (RANS) equations are solved using ANSYS-CFX. The governing equations for incompressible flow in Cartesian coordinate system are written below[14].

Continuity equation:

$$\frac{\partial}{\partial t} (\alpha_k \rho_k) + \nabla \cdot (\alpha_k \rho_k U_k) = 0$$

(1)

Momentum equation:

$$\frac{\partial}{\partial t} (\alpha_k \rho_k U_k) + \nabla \cdot (\alpha_k \rho_k U_k U_k - \alpha_k \tau) = -\alpha_k \nabla p + M_k + \alpha_k \rho_k f_k$$

(2)

where the subscript $k = l(g)$ represents the liquid (gas) phase; $\alpha_k$ is the void fraction, $\alpha_l + \alpha_g = 1$; $M_k$ is the interphase force; and $\tau$ is the viscous stress tensor.

In this study, the SST $k-\omega$ turbulence model is adopted with considering the complex gas-liquid two-phase flow and the adverse pressure gradient in the multiphase pump. In this model, the $k-\omega$ and $k-c$
models are employed for the near-wall and mainstream regions, respectively, thus it can give a high accuracy in predicting flow separation under the adverse pressure gradient[15].

3.2. Structured mesh and numerical solution
The structured mesh is adopted for the computational domain. The mesh for the impeller and guide vane is generated using TurboGrid, while the mesh for the inlet and outlet pipes is generated using ICEM-CFD. Moreover, after the mesh independence analysis, the total mesh number of 3.68 million is adopted finally[11].

At the inlet of the computational domain, the mass flow rate and the corresponding void fraction are set, respectively. Meanwhile, the bubble diameter is set as 0.4mm based on the experimental result. At the outlet, the average static pressure is specified. The transient rotor-stator method is imposed for the rotor-stator interaction region. The second order backward Euler method is used for the transient scheme and the RSM residual is set $1 \times 10^{-4}$. The minimum and maximum iteration steps are set 20 and 40, respectively. Moreover, after the time step independence analysis, the computational time step is set to $1.13 \times 10^{-4}$s, corresponding to the time for the impeller rotation of 2°, and 10 cycles are calculated.

3.3. Monitoring points
In order to investigate the pressure fluctuation in the pump, eight monitoring points denoted $R_i$ and $S_i$ (i=1, 2, 3, 4) are arranged along the flow direction and located at mid-height in the radial direction, as shown in Figure 2.

![Figure 2. Locations of monitoring points.](image)

4. Results and discussions
4.1. Model validation
The performance curves for simulation and experiment at $IGVF=0.0\%$ is shown in Figure 3. The numerical efficiency and head at $IGVF=0.0\%$ agree well with the experimental data, especially for the designed flow rate with the errors of efficiency and head which are only 0.9% and 3.0%, respectively. In addition, when the $IGVF$s are 3.0%, 9.0%, and 15.0% at $Q=50\text{m}^3/\text{h}$, the errors of head between simulation and experiment are 3.0%, 1.9%, and 3.3%, respectively, as shown in Table 1. The above analysis indicates that the numerical method adopted in this study is reliable.
Figure 3. Performance results for simulation and experiment at IGVF=0.0%.

Table 1. Comparison of pump head between simulation and experiment at different IGVFs.

| IGVF  | H_CFD | H_EXP | Error (%) |
|-------|-------|-------|-----------|
| 3.0%  | 16.5  | 16.0  | 3.0       |
| 9.0%  | 16.2  | 15.9  | 1.9       |
| 15.0% | 15.5  | 15.0  | 3.3       |

4.2. Pressure fluctuation in impeller

Table 2 shows the dominant frequency and corresponding fluctuation amplitude of monitoring points R1~ R4 at IGVFs =0.0%, 3.0%, 9.0%, 15.0%. The fn is the rotation frequency of impeller and fn=49.17 Hz. As shown in Table 2, the dominant frequencies of points in the impeller are mainly 11fn that corresponds to the blade numbers of guide vane. However, 0.5fn appears for point R1 at IGVF=3.0%, 15.0%, which is believed to be related to the flow separation and the vortex shedding. Point R1 is located at the impeller inlet and affected with a weak rotor-stator interaction, thus the fluctuation caused by the flow separation will play a key role at some conditions.

Table 2. Domain frequency (DF) and amplitude (A) of points R1~ R4 in impeller.

| IGVF  | R1  | R2  | R3  | R4  |
|-------|-----|-----|-----|-----|
|       | DF  | A/kPa | DF  | A/kPa | DF  | A/kPa | DF  | A/kPa |
| 0.0%  | 11fn | 0.10   | 11fn | 0.32   | 11fn | 0.73   | 11fn | 3.00   |
| 3.0%  | 0.5fn | 0.36   | 11fn | 0.92   | 11fn | 0.68   | 11fn | 0.74   |
| 9.0%  | 11fn | 0.18   | 11fn | 1.22   | 11fn | 1.12   | 11fn | 0.64   |
| 15.0% | 0.5fn | 0.23   | 11fn | 1.00   | 11fn | 0.91   | 11fn | 0.22   |

From Table 2, the pressure fluctuation in impeller increases gradually along the flow direction at IGVF=0.0%, while at IGVF=3.0%, 9.0%, 15.0, the fluctuation increases firstly and then decrease. Moreover, the amplitude of point R4 at IGVF=0.0% is 4.1, 4.7 and 13.6 times as great as that at IGVF=3.0%, 9.0%, 15.0%, respectively.

The time and frequency domain characteristics of point R4 are shown in Figure 4. Figure 4(a) shows that the average pressure of point R4 in one cycle decreases gradually with the increase of IGVF, which illustrates that the work capacity of impeller decreases. As the IGVF increases, the fluctuation amplitude of point R4 decreases gradually, as shown in Figure 4(b). This is closely related to the gas accumulation and velocity distribution therein. At the impeller outlet, the degree of gas accumulation and the gas velocity enhance with the increase of IGVF, as shown in Figure 5 and 6. Therefore, it can be obtained that the complex gas-liquid flow in the vicinity of the impeller outlet weakens the fluctuation therein.
4.3. Pressure fluctuation in guide vane
The dominant frequency and corresponding amplitude of monitoring points $S_1$~$S_4$ at different IGVF conditions (IGVF=0.0%, 3.0%, 9.0%, 15.0%) are listed in Table 3. The dominant frequencies of points $S_1$~$S_4$ are all $4fn$ at these four IGVFs, which corresponds to the blade numbers of impeller. Meanwhile, the pressure fluctuation at different IGVF's overall decreases from the inlet to outlet of guide vane.

| IGVF  | $S_1$ | $S_2$ | $S_3$ | $S_4$ |
|-------|-------|-------|-------|-------|
| 0.0%  | 4fn   | 1.70  | 4fn   | 1.71  | 4fn   | 0.66  | 4fn   | 0.40  |
| 3.0%  | 4fn   | 5.81  | 4fn   | 3.98  | 4fn   | 2.06  | 4fn   | 0.99  |
| 9.0%  | 4fn   | 8.69  | 4fn   | 5.34  | 4fn   | 2.62  | 4fn   | 1.52  |
| 15.0% | 4fn   | 9.89  | 4fn   | 5.55  | 4fn   | 2.74  | 4fn   | 1.48  |

As the increase of IGVF, the fluctuation of corresponding points shown in Table 5 overall increases, which ascribes to the disordered flow in guide vane, as shown in Figure 7. When the IGVFs are 3.0%, 9.0%, and 15.0%, the gas accumulation and flow separation are happened. Furthermore, the degree of gas accumulation and the range of vortexes enhance gradually with the increase of IGVF.
Combined with the fluctuation in impeller which is shown in Table 2, it can be concluded that the pressure fluctuation in guide vane is relatively large. Overall, the dominant frequencies of points $R_1 \sim R_4$, $S_1 \sim S_4$ are $11fn$ and $4fn$ respectively, which corresponds to the blade numbers of guide vane and impeller. Therefore, the rotor-stator interaction is the main reason for the generation of fluctuation in the multiphase rotodynamic pump. However, the maximum fluctuation amplitude appears near the impeller outlet (point $R_4$) at $IGVF=0.0\%$, while it appears near the inlet of guide vane (point $S_1$) under two-phase flow condition. Furthermore, the maximum fluctuation amplitudes at $IGVF=3.0\%$, $9.0\%$, $15.0\%$ are $1.9$, $2.9$, and $3.3$ times as large as that at $IGVF=0.0\%$.

The time and frequency domain characteristics of point $S_1$ which is located in the inlet of guide vane are shown in Figure 8. As the $IGVF$ increases, the average pressure of point $S_1$ decreases gradually, while the fluctuation increases gradually. Figure 7(b) shows that, aside from the dominant frequency $4fn$, the second dominant frequency $8fn$ with a relatively large amplitude also appears.

5. Conclusions
In this study, on the basis of Euler two-fluid model, the characteristics of pressure fluctuation in the multiphase pump were analyzed. The main conclusions can be drawn as follows:

1. Compared to the single-phase flow, the dominant frequencies of points $R_1 \sim R_4$, $S_1 \sim S_4$ are mainly $11fn$ and $4fn$ respectively, corresponding to the blade numbers of guide vane and impeller. Therefore, the rotor-stator interaction is the main reason for the generation of fluctuation in the multiphase rotodynamic pump.
(2) From the inlet to outlet of impeller, the pressure fluctuation increases gradually at $IGVF=0.0\%$, while it increases firstly and then decrease at $IGVF=3.0\%, 9.0\%, 15.0\%$. Meanwhile, the fluctuation in guide vane decreases gradually at these four $IGVFs$, but is larger than that in impeller.

(3) The maximum fluctuation amplitude appears near the impeller outlet (point $R_3$) at $IGVF=0.0\%$, while it appears near the inlet of guide vane (point $S_1$) under two-phase flow conditions. Furthermore, the maximum fluctuation amplitudes at $IGVF=3.0\%, 9.0\%, 15.0\%$ are 1.9, 2.9, and 3.3 times as large as that at $IGVF=0.0\%$.

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