Characteristic analysis on the pressure fluctuation in the impeller of a low specific speed mixed flow pump

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Abstract. To explore the pressure fluctuation characteristics of a low speed specific speed mixed flow pump caused by rotor-stator interaction, the unsteady flow was simulated with CFX for the whole flow passage of a mixed flow pump with a specific speed of 148.8. The structured mesh of the computation domain was generated with ICEM CFD and TurboGrid, and mesh-independent analysis was done in the design condition. Through the comparison with the experiment data, the reliability of the simulation was verified. In different locations of the impeller passage, monitoring points were set. With Fast Fourier Transform (FFT), the characteristic analysis on the pressure fluctuation in the impeller passage was done for three flow rate conditions (0.75Q_d, Q_d, 1.25Q_d). The results show that the pressure fluctuation amplitude increases from the inlet to the outlet. And the maximum values in different flow rates exist near the hub of the outlet; The pressure fluctuation is small in the design condition, but the largest in the small flow rate condition, accompanied by the secondary dominant frequencies with large amplitudes; In the small flow rate condition and design condition, the dominant frequency varies from the inlet to the outlet because the combine action of the impeller and guide vane; while in the large flow rate condition, the pressure fluctuation in the whole impeller passage is affected significantly by the guide vane, and the domain frequency is 8 times the rotational frequency of impeller. In addition, the change of pressure fluctuation from the pressure surface to the suction surface in the off-design conditions is investigated, and the results demonstrates that the intensity of the pressure fluctuation in the impeller passage is closely related with the impeller as well as the distribution of the vorticity and the pressure.

1. Introduction
The performance and structure of the mixed flow pump are between the centrifugal pump and the axial flow pump. Because of the features of compact structure, easy starting and high efficiency, mixed flow pumps are widely used in farmland irrigation, waterlogging drainage, sewage treatment, power plant cooling etc. Usually, low specific speed mixed flow pumps refer to those with a specific speed less than 350. In recent years, domestic and foreign scholars have done a lot of research on the design, experiment and numerical calculation of the mixed flow pump. Shi weidong et al. studied the characteristic on unsteady pressure fluctuation of the high specific speed mixed flow pump in different conditions by using time-average Renoylds equations; Bing hao et al. proposed parametric approach of velocity moment distribution law, designed a series of mixed flow pump impeller and predicted its performance by numerical simulation; Li Yibin et al. analyzed the turbulent flow in a mixed flow
pump with the RNG k-ε model and sliding mesh technique; Jin Shuanbao et al.[4] simulated the unsteady flow in a mixed-flow pump based on RANS solver embedded with SST turbulence model and SIMPLEC algorithm; in order to improve hydraulic performance of a mixed-flow pump with guide vanes, Chang Shuping et al.[5] explored the effects of several factors on the performance including the tip clearance, blade number, blade angle and blade thickness; ZHANG Desheng et al.[6] received an abruptly descending characteristic curve by the optimization of the wrap angle and the outlet blade angle, which could meet the need of acceleration at different working conditions; Chisachi Kato et al.[7] investigated the unsteady flow characteristics for a high specific speed mixed flow pump in the small flow rate condition based on LES method; Masahiro Miyabe et al.[8] found that the unsteady characteristic in low specific speed mixed flow pumps is caused by the large scale back flow between the outlet of impeller and the guide vane.

Comparatively speaking, less research is done on the design and the internal flow of low specific speed mixed flow pumps, which limits the scope of such pumps. In some cases, low specific speed mixed flow pumps are replaced unreasonably by centrifugal pumps[9]. In this paper, unsteady simulation was done with CFX for the internal flow of a mixed flow pump with a specific speed of 148.8. Monitoring points were set on different sections in the impeller passage to capture the change of the pressure, then analysis was done with the fast Fourier transform (FFT) to evaluate the characteristics of the pressure fluctuation in the low specific speed mixed flow pump.

2. Structure and parameters of the pump
The main parameters of the mixed flow pump are as below: impeller diameter $D$ is 150mm, blade number $Z_1$ is 6, guide vane number $Z_2$ is 8, design flow rate $Q_d$ is 1.39$m^3$/min, rotational speed $n$ is 2000 r/min, design head $H_d$ is 14.59m, and power $P$ is 5.5kw. The 3-D geometric model of the whole flow passage is constructed with UG NX 6.0 (see figure 1).

![Figure 1. Configuration of the mixed flow pump](image)

3. Numerical methods
3.1. Governing equations
The conservation equations of mass and momentum for incompressible flow are written in Cartesian coordinates as follows:

$$\frac{\partial u_i}{\partial x_i} = 0$$

(1)

$$\frac{\partial u_i}{\partial t} + (u_j u_j) \frac{\partial u_i}{\partial x_j} = - \frac{1}{\rho} \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + f_i$$

(2)

where $f_i$ is the Coriolis force; $\rho$ is the fluid density; $u$ is the relative velocity, $p$ is the pressure;
\( \mu \) is the viscosity of the fluid; \( \mu_t \) is the turbulent viscosity. The turbulence is modeled by the shear stress transport (SST) model. This model can give a more accurate prediction for flow separation under adverse pressure gradient. In this model, the eddy viscosity is computed by:

\[
\mu_t = \frac{\rho a_k}{\max \left( a_1 \omega, SF_2 \right)}
\]

where \( a_1 \) is the model constant and \( a_1 = 5/9 \); \( S \) is the invariant measure of strain rate; \( F_2 \) is the blending function; \( k \) and \( \omega \) are the turbulent kinetic energy and the turbulence frequency, respectively.

### 3.2. Mesh and boundary conditions

The computation domain includes four parts, i.e. the inlet pipe, the outlet pipe, the impeller and the guide vane. The structured meshes of the former two parts are generated by ICEM_CFD, while the meshes of the latter two are created with TurboGrid. The \( y^{+} \) values of the wall mesh of the impeller and the guide vane are between 20 and 30. Through the analysis of mesh independence in the design condition, the mesh numbers for a single passage of the impeller and the guide vane are determined to be 202720 and 200124, while the mesh number for the whole domain is 3979933. The meshes of the four parts are shown in figure 2.

![Structured mesh for flow analysis](image)

**Figure 2.** Structured mesh for flow analysis

At the inlet of the computation domain, uniform velocity was specified according to experimental value; at the outlet, free outflow condition was adopted. At all the wall boundaries, the non-slip condition of viscous fluid is used, and logarithmic wall function approach was applied in the near wall region. Transient Rotor-stator approach was adopted for the data exchange in the rotor-stator interaction region (namely the impeller/inlet pipe interface and the impeller/guide vane interface).
3.3. Other numerical settings

In order to achieve the unsteady flow information in a pump with a high resolution, the time step must be small enough\cite{10}. As the rotational speed is 2000r/min, the value of the rotational frequency of the impeller $f_n$ is 33.3Hz and the period of rotation $T$ is 0.03s. In this study, the time step was set as 0.0002s, thus the impeller rotates for $2.4^\circ$ in a time step. The total simulation time is 0.45s, which is equivalent to 15 periods.

![Figure 3. Location of monitoring points](image)

In order to study the characteristics of the pressure fluctuation in the mixed flow pump impeller, 5 monitoring surfaces and 45 monitoring points were set from the inlet to the outlet of the impeller, as shown in figure 3. Here, the monitoring surface 1 is near the inlet of the impeller, while the monitoring surface 5 is near the outlet; the subscripts C, N, S denote that the monitoring points are at the center, near the shroud and near the hub of the monitoring surface, respectively, while the superscripts ‘‘ and ‘‘ represent that the monitoring points are near the pressure surface and the suction surface, respectively.

![Figure 4. Performance curves from numerical simulation and experiment](image)
4. Results and discussions

4.1. Performance prediction and validation

To verify the reliability of the numerical model, the unsteady internal flow of pump were simulated in eight flow rate conditions, and comparison with the experiment was made for the head, the efficiency and the power, which is shown in figure 4. The calculation errors of the head, the efficiency and the power at each flow rate condition are less than 8.7%, among which the maximum is the efficiency error in the large flow rate condition, and this is because the cavitation is more serious in large flow rate condition. On the whole, the numerical and experimental results agree well with each other, which demonstrates that the numerical approach is reliable.

4.2. Characteristics of the pressure fluctuation in the impeller passage

The pressure fluctuation of the central monitoring points at different working conditions is presented in figure 5. Here, the pressure coefficient is defined as \( C_p = \frac{\Delta p}{0.5 \rho u^2} \), where \( \Delta p \) is the pressure difference between the monitoring points and the average pressure, \( u \) is the circumferential velocity of the impeller. It shows that the pressure fluctuation coefficient in the small flow rate condition (\( Q=0.75Q_d \)) is much greater than that in the design condition and the large flow rate condition, and the pressure fluctuation is in disorder. In the design condition (\( Q=Q_d \)) and large flow rate condition (\( Q=1.25Q_d \)), the pressure fluctuation coefficient is smaller, the fluctuation is more stable and the periodicity is more obvious. Eight peaks and valleys occurs in each cycles of the three flow rate conditions, which coincides with the blade number of the guide vane. Therefore, it can be concluded that the pressure fluctuation of point \( P_{sc} \) in the three flow rate conditions is mainly affected by the guide vane.

![Figure 5. Pressure fluctuation of the point P_{sc}](image)

The maximum values of the relative pressure fluctuation (indicated by \( \Delta p/\bar{p} \)) of the 9 monitoring points on each monitoring surface are compared in figure 6. It can be seen that the amplitude of pressure fluctuation increase from the inlet to the outlet. The pressure fluctuation in the impeller passage is smaller in the design condition (the fluctuation is greater than 2% only in monitoring
surface 5), which illustrates that the design of the low specific speed mixed flow pump is reasonable. The maximum fluctuation occurs in the small flow rate condition, which is consistent with the analysis in figure 5. The reason for this may be that surge phenomenon is easy to happen at small flow rate condition and thus the flow is more unstable. Furthermore, due to the influence of the rotor-stator interaction, the pressure fluctuation in monitoring surfaces 4 and 5 is greater, especially in monitoring surface 5. In the three working conditions (\(Q=0.75Q_d\), \(Q=Q_d\) and \(Q=1.25Q_d\)), the maximum values of the pressure fluctuation in monitoring surfaces 5 occur near the hub, with the value of relative pressure fluctuation being 21.8%, 5.5% and 17.1%, respectively.

**Figure 6.** Comparison of the monitoring points with maximum pressure fluctuation

Because the pressure fluctuation occurs mainly in monitoring surface 4 and 5, further analyses for these two monitoring surfaces are carried out. Figure 7 shows the frequency domain of the pressure fluctuation of the central monitoring points (\(P_{ac}\), \(P_{rc}\)) in the three flow rate conditions. In the small flow rate condition (\(Q=0.75Q_d\)), the design condition (\(Q=Q_d\)) and the large flow rate condition (\(Q=1.25Q_d\)), the dominant frequency of the point \(P_{ac}\) are 3\(fn\), 2\(fn\) and 8\(fn\), respectively, while the dominant frequency of the point \(P_{rc}\) are all 8\(fn\). On the other hand, according to the calculation

**Figure 7.** Frequency domain characteristics of the central monitoring points
results, all the dominant frequencies of monitoring points $P_{1c} \sim P_{sc}$ are 3fn in the $Q=0.75Q_d$ condition, while the dominant frequency of the same points are 6fn, 6fn and 2fn at $Q=Q_d$ condition, and all of them are 8fn at $Q=1.25Q_d$ condition. Therefore, the pressure fluctuation of the monitoring points $P_{1c} \sim P_{sc}$ is influenced by the interaction of the impeller and the guide vane in the small flow rate condition and the design condition, while the pressure fluctuation of $P_{sc}$ is mainly affected by the guide vane. In large flow rate conditions, the pressure fluctuation of the points $P_{1c} \sim P_{sc}$ is mainly affected by the guide vane.

It can also be found that, the amplitudes of pressure fluctuation of the points $P_{sc}$ and $P_{sc}$ are the largest in the small flow rate condition, which are 8.35 and 2.11 times as great as that in the design condition, respectively. The amplitudes in the large flow rate condition are 8.19 and 1.09 times as great as that in the design condition, respectively. In the design condition, in addition to the dominant frequency, the amplitudes of the secondary dominant frequency of the points $P_{sc}$ and $P_{sc}$ are also great, and many other frequencies are accompanied, which increases the flow instability in the pump. Therefore, off-design conditions, especially the small flow rate conditions, should be avoided for the running of low specific speed mixed flow pumps.

4.3. Pressure fluctuation from the pressure surface to the suction surface

To compare the pressure fluctuation from the pressure surface to the suction surface, we make an average of the root mean square (RMS) for the fluctuation values near the pressure surface, in the center and near the suction surface, as shown in figure 8. On the whole, the pressure fluctuation gradually increases from the inlet to the outlet in the three flow rate conditions. Therefore, the effect of rotor-stator interaction is enhanced in this direction. The fluctuation amplitude of the monitoring surface 1 keeps almost constant from the pressure side to the suction side, and this is because surface 1 locates at the inlet of the impeller, where the impact of guide vane is small for the flow. In general, the pressure fluctuation in surface 2~4 decreases from the pressure side to the suction side, with an exception for the monitoring surface 4 in large flow rate condition ($Q=1.25Q_d$), where the fluctuation at the central point is slightly greater than that near the pressure surface. For the monitoring surface 5

![Figure 8. Pressure fluctuation from the pressure surface to the suction surface](image-url)
in the small flow rate condition (Q=0.75Q₁) and the large flow rate condition (Q=1.25Q₁), the amplitude of fluctuation near the pressure side is the largest, followed by that near the suction side, and the fluctuation in the middle is the smallest. This may be related to the outlet structure of the low specific speed mixed flow pump.

The vorticity and pressure distribution in the small flow rate condition at the span of 0.5 are shown in figure 9. On the whole, the vorticity changes little in the upstream section of the impeller passage, while at the outlet region, it increases dramatically near the pressure surface and the suction surface and becomes small at the center. This is consistent with the change of the fluctuation amplitude shown in figure 8, indicating that the intensity of the pressure fluctuation in the impeller passage is closely related with the vorticity magnitude. From figure 9(b), it can be seen that the pressure gradient near the outlet of the impeller passage is obviously larger than that in the inlet and the middle section of the passage, which illustrates that the flow herein is affected significantly by the rotor-stator interaction, and thus the steeply increase of pressure fluctuation is aroused in the monitoring surface 5.

![Vorticity and Pressure Distribution](image)

**Figure 9.** Distribution of the vorticity and the pressure in the small flow rate condition (Span=0.5, t=0.448s)

5. Conclusions

Through numerical calculation, the characteristics of the pressure fluctuation in the impeller of a low specific speed mixed flow pump were analyzed in the design condition and the off-design conditions. The conclusions obtained are as follows:

1) The amplitude of pressure fluctuation increases from the inlet to the outlet of the impeller. In the small flow rate condition, design condition and large flow rate condition, the maximum values of the pressure fluctuation exist near the hub of the impeller outlet, and the values of the relative pressure fluctuation are 21.8%, 5.5%, 17.1%, respectively.

2) The pressure fluctuation in the impeller passage is small in the design condition (exceeds 2% only in the region near the impeller outlet), which means the design of the mixed flow pump is reasonable; while the pressure fluctuation is the largest in the small flow rate condition, accompanied by secondary dominant frequencies with large amplitude, and this will intensify the instability of the flow in the pump.

3) In the small flow rate condition and the design condition, the dominant frequencies varies from the inlet to the outlet because the combined action of the impeller and guide vane; while in the large flow rate condition, the pressure fluctuation in the whole impeller passage is affected significantly by the guide vane, and the domain frequency is 8 times the rotational frequency of impeller.

4) In off-design condition, the pressure fluctuation in the upstream section of the impeller passage tends to decrease from the pressure surface to the suction surface; while in the region near the outlet, the pressure fluctuation near the pressure surface is the largest; followed by that near the suction
surface, and it is smallest in the center. The intensity of the pressure fluctuation in the impeller passage is closely related with the impeller structure as well as the vorticity and the pressure.

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