Turbulent flow and pressure fluctuation prediction of the impeller in an axial-flow pump based on LES

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Abstract. The Large Eddy Simulation method with sliding mesh technique has been used for analyzing the unsteady flow in an axial-flow pump at five different flow rates. The tip leakage flow in the tip-gap region and the pressure pulsations on the blade surface were examined. The results indicate that the agreement between predicted pump performance and experimental data was reasonably good. The dominant tip-leakage vortex (TLV) extended to the pressure side of the neighboring blade for all five investigated flow rates. As the flow rate increases from 0.7Qd to 1.2Qd, the angle between the dominate TLV and the blade reduced from 20 deg to 14 deg. The results also showed that the amplitude of pressure fluctuation on the near-tip zone of the blade surface increases as the flow rate farer from the design flow rate, especially on the pressure side of the blade. At the 0.7Qd operation condition, the pressure fluctuation amplitude of the monitoring point PP3 (at the near-tip zone on the pressure side of the blade close to the blade leading edge) was 8.5 times of the one at design flow rate, and the high-frequency (18f_r) pulsation occurred due to tip leakage vortex. When the flow rate was more than 1.0Qd, the pressure fluctuations of PP3 was dominated by the rotation frequency (f_r).

1. Introduction

The tip clearance flow in an axial-flow pump is ineluctable for its operation which interacts with mainstream, boundary layer and blade wake, forming complex vortex structures in the impeller which induce vibration noise, hydraulic vibration and tip clearance cavitation, and then influences the operation stability of pump unit[1]. Therefore, the analysis of tip clearance flow and pressure fluctuation which affected by TLV in the impeller becomes significantly important for improving hydraulic design and stable operation of axial-flow pump.

With the inconvenience of installation, high cost, long period and scale effect, test is difficult in some degree. Thanks to the development of computational fluid dynamics (CFD), lots of research have been conducted to analyze characteristics of tip clearance flow[2,3]. The Reynolds Average Navier-Stokes (RANS) equations is commonly used at present. However, the instantaneous flow is treated by using time average technique and could not provide more information regarding the unsteady pressure field in gap region. So there is some limitation in solving the unsteady flow field. Large Eddy Simulation (LES) seeks to directly solve large spatial scales, while modeling the smaller scales, which could seize more details of unsteady flow and has been proved that predicts accurately on unsteady turbulent flow in hydraulic machinery[4-6]. Ghias et al[7] used LES solver to simulate the tip-flow of a rotor in hover, and the vortices contours in the wake near the tail of the airfoil were obtained at Re=100,000. YOU et al[8] predicted the effects of tip-gap size on the TLV structures and
velocity and pressure fields by using LES method. They concluded that larger tip-gap sizes were more inductive to tip clearance cavitation. By using LES method, the analysis of turbulent flow in an axial-flow pump impeller under low flow rate condition was performed by Liang et al\cite{9}. However, these studies which focused on tip leakage flow did not provide information regarding the effects of operating condition on TLV and other secondly TLV vortices. Moreover, the previous studies provided little information regarding the pressure pulsations which are influenced by TLV on the near-tip blade surface.

In this paper, LES method is applied to simulate the unsteady tip clearance flow in axial-flow pump. Firstly, the pump performance of CFD is compared to test data. Secondly, the detailed characteristics of tip clearance flow and TLV under various operation conditions will be given. Thirdly, the effects of TLV on pressure pulsations in impeller will also be discussed. At last, conclusions are made.

2. Computational Methodology and Mesh Generation

The object for this paper is a single-stage axial-flow pump (\(n = 1450\) r/min, \(Q_d = 0.35\) \(\text{m}^3/\text{s}\), \(H = 11\) m) with 6 blades rotor and 11 blades stator. The corresponding specific speed is \(n_q = 142\). The rotor tip diameter is 299.7 mm, with the tip-gap 0.3 mm.

The CFD simulations are setup for the whole computational domain includes suction chamber, rotor, stator and discharge bend pipe, as shown in the Fig. 1. The multi-block structured grid method is adapted to rotor, and the grid of C-H type is used to tip-gap region. The other three domains used the hexahedral mesh. For the purpose of obtaining an accurate prediction of the flow in boundary layer region, local mesh densification is carried out in gap region (\(y^+ \) close to 10) with the height of the first cell close to near-wall is equal to \(6 \times 10^{-5}\) m. The corresponding overall number of cells for the whole computational domain is equal to about 4,210,000.

![Figure 1. Sketch of computational domain](image)

![Figure 2. Computational mesh of impeller and tip clearance](image)

3. Boundary conditions and Numerical methods

The boundary and calculation conditions are as follows:

- The velocity is given uniform according to flow rate at the inlet of suction chamber;
- The gradients of all the variables are assumed to be 0 at the outlet;
- Solid surfaces: No-slip wall, Interface is applied to connect different domains and grids;
- Dynamic and static interface: Sliding mesh technique;
- Discretization scheme: The finite volume method is used to discretize the governing equations; The second-order upwind differencing scheme is adopted for convection item and bounded central differencing scheme for dissipation item;
- Pressure-velocity coupling: SIMPEC algorithm;
- Turbulence model: LES with Smagorinsky-Lilly SGS model;
- Time step size: $\Delta t = 1 \times 10^{-4}$ s (0.87°).

The commercial code FLUENT is adopted to carry out the simulations. The unsteady flow is analyzed under different conditions ($0.7Q_d$, $0.8Q_d$, $1.0Q_d$, $1.1Q_d$ and $1.2Q_d$), wherein, $1.0Q_d$ represents the design condition. In order to analyze the effects of TLV on pressure pulsations in impeller, three monitoring points are arranged both on pressure surface and suction surface, as shown in Fig. 3.

4. Calculation analysis

4.1. Pump Performance

The performance of the pump as predicted from LES method is compared to test data. As is showed in Fig. 4 and Fig. 5, the CFD results of head and efficiency agree well with the measurement data. And the maximal relative error of head and efficiency is 4.32% and 2.34%, respectively. The good agreement between CFD and test shows that LES method can provide the unsteady flow of pump accurately.

4.2. Analysis on internal flow

Figure 6 shows the contour plots of mean pressure distribution on blade-to-blade surface at $r/R_s=0.99$ for different flow rates. The differential pressure of pressure surface and suction surface decreased with the flow rate growth due to the head decreased at a larger flow rate. The region of low-pressure existed due to TLV, and the intensity and range of low-pressure decreased with the flow rate growth. The main TLV was gradually away from the suction side of the blade, and extended to the pressure side of the adjacent blade under different flow rates. The trajectory of the TLV is significantly influenced by the change of operation conditions. The length of trajectory is shorter under the condition of lower flow rate, especially at $0.7Q_d$ that the TLV breaks at the leading part of near-tip pressure surface. As showed in Fig. 6, the angle between TLV and the blade decreases according to the increase of flow rate, $0.7Q_d$ and $1.2Q_d$ were 20° and 14° respectively. And the origin of TLV is delayed further downstream from the leading edge with the flow rate growth.
The TLV structures are visualized using the Q vortex identification method[10] and is shown in Figure 7. In this vortex identification method, Q is equal to \(1/2(|\Omega|^2-|S|^2)\), where \(\Omega\) and \(S\) are the antisymmetric and symmetric parts of velocity gradient tensor respectively. The view of perspective is showed in Fig. 7(a). For all five operating conditions considered in this paper, the main TLV dominates the tip-gap vortex structures, which decreases in strength and lateral scale as the flow rate growth. The tip-separation vortex and induced vortex are found at the tip-gap region and rear part of the blade respectively. The trajectory of main TLV presents the state of multi-curve when the flow rate is less than \(1.0Q_d\), especially at \(0.7Q_d\) that the vortex rotates violently and breaks down early at the leading part of near-tip pressure surface, and the tip-gap region is full of distributed vortex structures. The secondly TLV is formed at the rear part of the blade under different flow rates, especially at \(0.7Q_d\) and \(0.8Q_d\) that the secondly TLV even gets in touch with the main TLV at the leading part of blade, as showed in Figs. 7 (b)-(c).

4.3. Analysis on pressure fluctuation in impeller
The relative peak-to-peak value of mixed frequencies is used for describing the amplitude of pressure fluctuation, with 8 times of a rotating cycle for spectrum analysis by Fast Fourier Transformation (FFT). The coefficient of pressure pulsation\((C_s)\) is introduced for comparison:
\[ C_s = \frac{p - \overline{p}}{2\rho U^2} \]

where \( p \) is instantaneous static pressure, \( \overline{p} \) is average static pressure in 8 times of a rotating cycle, \( U \) is the circumferential velocity of blade tip.

Peak-to-peak pressure fluctuation at each monitoring point for different flow rates is shown in figure 8. From the simulation results we can clearly observe that the amplitude of pressure fluctuation on the near-tip blade surface became higher when the flow rate was farer from the design flow rate, especially under \( 0.7Q_d \) and \( 1.2Q_d \), but it is not obvious from \( 0.8Q_d \) to \( 1.1Q_d \). The increasing amplitude of pressure fluctuation is more violent at each monitoring point on pressure surface and PS3 on suction surface. In Fig. 7, we can see that the main TLV is gradually away from the suction side of the blade, and extended to the pressure side of the adjacent blade. The near-tip blade surface and the leading part of near-tip suction surface are affected severely by TLV. The high-amplitude of pressure pulsation occurs on near-tip blade surface due to TLV.

![Figure 8. Peak-to-peak pressure fluctuation at each monitoring point for different flow rates](image)

To further study of the characteristics of pressure fluctuation at location PP3, the pressure fluctuation in time for PP3 is showed in Fig. 9. The pressure fluctuation curves of \( 1.0Q_d \), \( 1.1Q_d \) and \( 1.2Q_d \) are similar to each other and the pressure fluctuates periodically with time. However, the waves of \( 0.7Q_d \) and \( 0.8Q_d \) are irregular, especially at \( 0.7Q_d \). As Fig.9 shows, the peak-to-peak pressure fluctuation(\( \Delta C_s \)) at location PP3 has a increasing tendency at first and then decreases with the flow rate growth, the maximum and minimum amplitude of pressure fluctuation occurs at \( 0.7Q_d \) and \( 1.0Q_d \) respectively, 8.5 times and 1.0 times of design condition.

![Figure 9. Pressure fluctuation at location PP3 for different flow rates](image)

Frequency spectra of pressure fluctuation at location PP3 for different flow rates is showed in Fig. 10. The separated flow on blade surfaces and TLV hardly affect the frequency of pressure fluctuation at location PP3 when the flow rate is more than \( 1.0Q_d \), and the rotating frequency(\( f_r \)) dominates the
pressure fluctuation of PP3. However, 11 times of \( f_r \) dominates the pressure fluctuation of PP3 under \( 0.8Q_d \). At the flow rate of \( 0.7Q_d \), masses of high-frequency pulsations occur, and the 18 times of \( f_r \) dominates the pressure fluctuation of PP3. From the above results, we can observe that the frequency of pressure fluctuation at location PP3 is severely affected by the genesis and development of TLV due to the main TLV extends to the leading part of near-tip pressure surface and breaks down, as is showed in Fig. 7.

![Figure 10. Frequency spectra of pressure fluctuation at location PP3 for different flow rates](image)

5. Conclusion
The unsteady characteristics of tip clearance flow and the pressure pulsations on the near-tip blade surface under various conditions were obtained based on LES method. The main conclusions are summarized as follows:

1) The CFD results of head and efficiency show good agreement with test data.
2) The dominate TLV extends to the pressure side of the adjacent blade for all five investigated flow rates. The angle between the dominate TLV and the blades decreases as the flow rate increases. For \( 0.7Q_d \) and \( 1.2Q_d \), the corresponding angle is 20° and 14°, respectively.
3) The amplitude of pressure fluctuation in the near-tip region on the blade becomes higher when the flow rate is farther from \( Q_d \), especially on the pressure side of the blade. The amplitude at the leading part of near-tip pressure surface is 8.5 times of design flow rate under \( 0.7Q_d \). At the near-tip zone on the pressure side of the blade (PP3), the amplitude of pressure fluctuation of \( 0.7Q_d \) is 8.5 times of the one at \( 1.0Q_d \).
4) When the flow rate is more than \( 1.0Q_d \), the dominate frequency of the pressure fluctuation of PP3 is the rotating frequency(\( f_r \)). At \( 0.8Q_d \), the dominate frequency increases to 11\( f_r \). At \( 0.7Q_d \), 18\( f_r \) dominates the pressure fluctuation of PP3, and high-frequency bands occur due to the effect of the TLVs.

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