Investigation of tip leakage vortex characteristics around tip clearance in a mixed flow pump

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Abstract. A systematical investigation on hydraulic performance and unsteady flow characteristics around tip clearance in a mixed-flow pump is conducted based on experimental results and steady simulation. The 1mm increase of tip clearance size results in that the pump head reduces by 1.79m, and pump efficiency decreases by 5.49% at design flow rate. At large flow rate, the performance drop remarkably increases compared with low flow rate and design flow rate. The unsteady simulation reveals the spatio-temporal evolution of TLV (tip leakage vortex), and interaction process of primary TLV and secondary TLV is also observed. The results show that at low and design flow rates, the start points of primary TLV are near leading edge, and the separation angle is always 25°. At large flow rate, the start point of primary TLV shifts towards trailing edge, and the separation angle decreases to 15°. Based on the statistical method, pressure fluctuations in tip clearance and on blade near leading edge are also analysed. The results show that strongest pressure fluctuations appear at regions near leading edge, and the flow instability is significantly intensified under low flow rate.

1. Introduction.
Hydraulic machinery is widely used in fields of electric power, water conservancy, chemical industry and so on. It is the core component in energy conversion for hydraulic power, which is one of the most reliable and renewable energy source worldwide [1, 2]. What’s more, it can also be used as general machinery with quite large energy consumption [3]. Mixed-flow pump, with wide operation range and high efficiency, is one of the most universal hydraulic machinery in current industry. So it is of great significance to improve the energy conversion efficiency and operation stability of mixed-flow pump. Tip clearance flow, induced by the tip clearance between blade tip and shroud in impeller, can generate remarkable impact on the performance of pump [4]. The flow characteristics in pump are very complicated due to the tip leakage vortex and its interaction with main flow, thereby influencing the work ability and efficiency of pump.

Lots of researchers have paid efforts to study the hydraulic performance in different types of hydraulic machineries. The simulation results in [5] revealed that tip clearance led to dramatic drop of pump head and efficiency; there was a nearly linear relationship between pump efficiency and head and tip clearance size, which agreed with the conclusion of experiment test in [6, 7]. For both the impulse turbine [8] and Wells turbine [9], tip clearance resulted in the remarkable performance drop, but the overall performance might be improved when there was a reasonable collocation of non-uniform tip
clearance the in the Wells turbine. Based on the experimental test on an axial flow waterjet [10], there was a 25% reduction of system efficiency when the tip clearance raised from 0.7% of impeller diameter to 1.5% of that.

It can be seen that tip clearance generates significant and complex impact on pump performance, and this is closely related to the flow characteristics resulted from the tip clearance. The vortexes induced by tip clearance were divided into three categories: tip leakage vortex, tip separation vortex, and induced vortex, which were different in formation mechanism, inception region, and motion law [11, 12]. The experiment measurement [13-15] revealed that the tip leakage vortex started at about 30% chord near blade tip, and gradually developed towards adjacent blade. Vortexes shed from blade tip would be entrained into the primary tip leakage vortex, thus increasing the size of primary tip leakage vortex, but when the tip leakage got close to the adjacent blade it would break down and disperse into some small vortexes. Under cavitation condition, the tip leakage vortex cavitation cloud would interact with the sheet cavitation cloud in flow passage, and a cavitation vortex perpendicular to suction side of adjacent blade was formed near leading edge [16-18]. The complex vortex pattern shown in above analyses also endangers the operation stability. Researches on pressure fluctuation under non-cavitation [19, 20] and cavitation condition [21] all indicated that pressure fluctuations in impeller, especially in tip clearance regions, were significantly intensified when tip clearance increases, and got even stronger under cavitation condition. Besides, the radial force fluctuations were greatly intensified under unsymmetrical tip clearance [22, 23]. Above analyses indicate that the flow characteristic associated with tip clearance flow is very complicated, and it will generate tremendous influence on the hydraulic performance and operation stability of hydraulic machineries.

Though there are already varies studies on the above issues related to tip clearance, the association among energy performance and tip clearance is not well established, and there is a lack of in-depth analysis on the characteristics of TLV and pressure fluctuations. In the present research, the energy performance, TLV structures and pressure fluctuation characteristics at different flow rates are thoroughly researched based on the steady and transient simulation results.

2. Physical model and mesh arrangement.

2.1. Physical model.
Figure 1 shows the physical model of the researched object, consisting of suction pipe, impeller with 5 blades, guide vanes with 6 blades and outlet pipe. The mixed-flow pump is designed based on the direct and inverse iterative method. The key parameters and corresponding design values for this pump are listed in Table 1. The tip clearance size is set as 1mm, 1% of mean blade span.
2.2. Mesh arrangement.
To discretize the whole computation domain, structured hexahedral meshes are generated for each domain separately with ANSYS-Tubogrid and ICEM. As shown in Figure 2, the mesh near blade surface is well locally refined based on O-block topology. Similar local mesh refinement is applied around guide vane blades. In tip clearance region, 15 nodes are arranged from blade tip to shroud; 20 nodes are arranged from PS (pressure side) to SS (suction side). These mesh arrangements provide strong support for capturing details of tip clearance flow.

![Mesh refinement near blade](image1)

![Mesh refinement in tip clearance](image2)

Figure 2. Mesh refinement in impeller

3. Numerical methods and settings.

3.1. Numerical methods.
In present research, the computation fluid dynamics code of ANSYS-CFX 14.5 is applied to conduct the numerical calculation. The turbulence model in steady simulation is set as the SST $k$-$\omega$ model, considering its advantage in accurately predicting flow pattern in near-wall regions. As for the boundary conditions, total pressure inlet condition is used at pump inlet boundary, mass flow outlet condition is applied at pump outlet boundary, and no slip wall is set at walls. The frozen rotor interface is applied to couple the rotational and stationary domains in steady calculation [24, 25].

In transient simulation, the LES turbulence model is selected to capture more detailed and accurate flow details and instability characteristics. The set of time step is according to the impeller rotating period $T$. After the independence validation of time step in previous research (see [6] for details), the time step adopted in the calculation is set as $2.5862 \times 10^{-4}$, which is equal to $1/160T$. The interface
adopted to couple the rotational and stationary domains has been changed to transient rotor stator in transient simulation [22, 23]. The simulation can be identified as converged when the root-mean-square residual is under \(1 \times 10^{-5}\).

3.2. Independence validation of mesh density.
To guarantee that the calculation result is not affected by the mesh density, the independence test of mesh density is conducted. Four sets of meshes with elements from 4,715,842 to 9,230,887 are selected. As can be seen from the results in Table 2, when the mesh elements are beyond 7,827,832, the pump head and efficiency rarely change with the further increasing of mesh density; the relative change of efficiency between Mesh 3 and Mesh 4 is just 0.01%. Therefore, considering the computation cost and simulation accuracy, Mesh 3 with 7,827,832 elements is selected to continue the present research.

| Component          | Mesh 1      | Mesh 2      | Mesh 3      | Mesh 4      |
|--------------------|-------------|-------------|-------------|-------------|
| Inlet pipe         | 459108      | 459108      | 459108      | 459108      |
| Outlet pipe        | 1511470     | 3032000     | 4623460     | 6026515     |
| Impeller           | 2065536     | 2065536     | 2065536     | 2065536     |
| Guide vane         | 679728      | 679728      | 679728      | 679728      |
| Whole passage      | 4715842     | 6236372     | 7827832     | 9230887     |
| \(H/H_1\)          | 1           | 0.9909      | 0.9850      | 0.9837      |
| \(\eta/\eta_1\)    | 1           | 1.0011      | 1.0013      | 1.0014      |

4. Results and discussions.

4.1. Validation of simulation accuracy.
Relying on the test apparatus in Beifang Investigation, Design &Research Co., Ltd, the experiment test on the mixed-flow pump is completed. In the test, after comprehensive analysis, the measurement error is assessed to be smaller than ±0.28%. To get the energy performance of pump in wide range, the pump at 17 flow rates from 400 kg/s to 720kg/s are calculated. As shown in Figure 3, there is a fine coincide between the performance curves in experiment and calculation; the best agreement can be observed near the design flow rate. Thus it can be seen that the numerical methods and settings illustrated above is of sufficient quality to conduct the numerical simulation work on the mixed-flow pump.
4.2. Energy performance.
Table 3 presents the head and efficiency versus flow rate for pumps with 0mm tip clearance (TC0) and 1mm tip clearance (TC1). As shown in the table, the pump head decreases with the increase of flow rate all through, while pump efficiency reaches the peak value near design point. In comparison of pump of TC0, both head and efficiency remarkably drop when the tip clearance is 1mm. The 1mm increase of tip clearance size results in that the pump head reduces by 1.79m, and pump efficiency decreases by 5.49% at design flow rate. With the increase of flow rate, the head drop $\Delta H$ reduces in general, with a relative small range. However, the efficiency drop $\Delta \eta$ significantly increases at large flow rate, demonstrating that the tip clearance generated stronger negative impact on the energy conversion process.

| Flow rate | $H_{TC0}$ | $H_{TC1}$ | $\Delta H$ | $\eta_{TC0}$ | $\eta_{TC1}$ | $\Delta \eta$ |
|-----------|-----------|-----------|------------|--------------|--------------|----------------|
| 420       | 20.60     | 18.67     | 1.92       | 80.36        | 77.02        | 3.34           |
| 440       | 19.91     | 18.00     | 1.91       | 81.65        | 78.50        | 3.14           |
| 460       | 19.45     | 17.77     | 1.68       | 83.71        | 81.46        | 2.25           |
| 480       | 18.87     | 17.31     | 1.56       | 85.19        | 83.12        | 2.07           |
| 500       | 18.44     | 16.68     | 1.77       | 87.39        | 83.77        | 3.62           |
| 520       | 17.57     | 15.90     | 1.67       | 87.84        | 83.71        | 4.13           |
| 540       | 16.60     | 14.81     | 1.79       | 87.71        | 82.22        | 5.49           |
| 560       | 15.36     | 13.67     | 1.70       | 85.99        | 80.26        | 5.73           |
| 580       | 14.18     | 12.46     | 1.72       | 84.32        | 77.75        | 6.57           |
| 600       | 12.93     | 11.27     | 1.66       | 82.03        | 75.08        | 6.95           |
| 620       | 11.50     | 10.01     | 1.49       | 78.07        | 71.51        | 6.56           |
| 640       | 10.12     | 8.56      | 1.56       | 73.80        | 65.89        | 7.91           |
| 660       | 8.66      | 7.22      | 1.44       | 68.18        | 60.23        | 7.95           |
4.3. Flow pattern in impeller.

Figure 4 shows the vortex pattern on circumferential section of 98% blade height at three flow rates in turbo perspective. As can be seen, the tip leakage vortex can be classified into two major types: primary TLV and secondary TLV. The start point of primary TLV is near blade leading edge at low flow rate and design flow rate, while it shifts to about 20%-30% chord at large flow rate. This shift leads to that there are more vortices in the middle of flow passage at 1.2Q. Besides, the primary TLV remarkably increases with flow rate varies from 0.8Q to 1.2Q. These two findings may account for the largest efficiency drop at 1.2Q. The intensity of secondary TLV increases when the flow rate raises from 0.8Q to 1.0Q. Meanwhile, obvious interaction process between primary TLV and secondary TLV can be observed at 1.0Q. The secondary TLV moves towards trailing edge at 1.2Q, and a tip vortex starts from leading edge and soon terminates.

A characteristic parameter: the separation angle $\alpha$ between primary TLV and blade tip is introduced. As shown in Figure 4, the separation angles at 0.8Q and 1.0Q are in the same value of 25º, while the separation angle is 15º at 1.2Q. The difference of separation angle also results in that the vortexes concentrate in regions near blade leading edge at 0.8Q and 1.0Q, but there are more vortexes expanding into the middle of flow passage at 1.2Q. Consequently, the decrease of separation angle may also account for the serious efficiency drop at large flow rate.

![Figure 4. Distribution of velocity swirling strength on circumferential section of 98% blade height.](image)

4.4. Unsteady flow instability.

The transient calculation is performed for 16 periods of impeller rotation. Using the data in last 8 revolutions, the pressure fluctuation intensity $\bar{p}'$ is defined by the RMS (root mean square) method as follows:

$$\bar{p} = \frac{1}{N} \sum_{i=1}^{N} p_i$$

$$\bar{p}' = \sqrt{\frac{1}{N} \sum_{i=1}^{N} (p_i - \bar{p})^2}$$
where $\bar{p}$ is the arithmetic average of pressure; $N$ is the sample number; $p_i$ is the pressure at each time step, and. The $p'$ is nondimensionalized by the following method:

$$I_{PF} = \frac{1}{2} \rho U_{Tip}^2$$

(4)

where $I_{PF}$ is the nondimensionalized pressure fluctuation intensity; $\rho$ is the density of fluid; $U_{Tip}$ is the tip velocity at leading edge.

From the transient simulation result, it can be seen that the pressure fluctuation is strong in tip clearance and near leading edge. So six axial sections located at LE (leading edge), 20%$\lambda$, 40%$\lambda$, 60%$\lambda$, 80%$\lambda$ and TE (trailing edge) are selected to analyse pressure fluctuation in tip clearance. In Figure 5, $\lambda$ denotes blade chord; $\delta$ denotes tip clearance height; $\varepsilon$ denotes blade tip thickness. Figure 5(a) shows the variation of $I_{PF}$ versus $\delta$ from blade tip to shroud wall at these sections under design flow rate. For all sections, the highest pressure fluctuation intensity along $\delta$ occurs on blade tip. As can be seen, the average $I_{PF}$ is highest at 20%$\lambda$, followed by $I_{PF}$ at LE and TE. The strong pressure fluctuation at 20%$\lambda$ may result from the inception of secondary TLV near 20%$\lambda$, shown in Figure 4(b). Figure 5(b) shows the variation of $I_{PF}$ on blade tip versus $\varepsilon$ from blade PS to SS at six axial sections. The pressure fluctuation intensity at LE is extremely high on blade tip near SS, due to the inception of primary TLV near leading edge. The $I_{PF}$ rarely changes along $\varepsilon$ at 80$\lambda$ and TE, demonstrating that the pressure fluctuation intensity is independent of tip thickness when getting close to trailing edge.

Figure 5. Pressure fluctuation intensity in tip clearance at six sections

(a) From blade tip to shroud (b) From pressure side to suction side.

Figure 6 shows the distribution of $I_{PF}$ on blade near leading edge at three flow rates. At design flow rate, there are strong pressure fluctuations on both blade tip and leading edge. On blade tip, pressure fluctuations close to leading edge are violent in regions near SS, corresponding to Figure 5(b), there are also violent pressure fluctuations on regions about 2% to 5% $\lambda$ near PS, where strong flow separation may occur. At low flow rate, pressure fluctuations on leading edge are greatly intensified, with nearly five times $I_{PF}$ value. This may be related to flow impact at leading edge and the blocking effect of secondary flow on the TLV development in mainstream channel at low flow rate. Therefore, for the pump with tip clearance, failure probably occurs at blade leading edge near blade tip at low flow rate. At large flow rate, the pressure fluctuations are significantly weakened. This may partly attribute to that the decrease of separation angle makes fewer TLV arriving at adjacent blade surface, thus decreasing pressure fluctuations on blade. In general, the increase of flow rate will restrain the flow instability induced by tip clearance in pump.
5. Conclusion.

The present research conducts work on energy performance and unsteady flow characteristics for a mixed-flow pump with tip clearance. Based on the numerical results and theoretical analysis, the main conclusions can be obtained as follows:

1. The tip clearance significantly influences the energy performance of pump. The 1mm increase of tip clearance size results in that the pump head reduces by 1.79m, and pump efficiency decreases by 5.49% at design flow rate. At large flow rate, tip clearance generates stronger negative impact on the energy performance.

2. At low and design flow rates, the start points of primary TLV are near leading edge, and the separation angle is always 25º. The vortexes concentrate in regions near leading edge. At large flow rate, the start point of primary TLV shifts towards trailing edge, and the separation angle decreases. Consequently, there are more vortexes expanding into the middle of flow passage at 1.2Q, leading to larger performance drop.

3. Strongest pressure fluctuations appear at leading edge and nearby blade tip. The pressure fluctuation on leading edge is significantly intensified under low flow rate. With the increase of flow rate, the flow instability induced by tip clearance gets restrained.

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