1. Introduction

Cavitation, a complicated physical phenomenon involving multi-scale and multi-phase changes, often occurs in the devices handling liquid, such as turbine blades, ship propellers and pumps. As a prevailing research topic, cavitation in vortical flows has been investigated for well over hundred years. Improvements in the design of hydraulic machines have led to much better control of cavitation effects. Consequently, various kinds of cavitation phenomenon which were not much considered are attracting more attention, such as tip vortex cavitation. Delaying or eliminating this type of cavitation (entirely or partially) is a major goal in design of turbomachines, because of the undesirable consequences such as mechanical damage, performance loss, noise and vibration.

An axial-flow pump in which the flow is parallel to the axis has been widely used for engineering applications. The presence of tip clearance between the rotor blade tip and fixed casing wall is indispensable for its operation. The leakage jets through the clearance is driven by the pressure difference between the pressure side (PS) and suction side (SS). When passes through the narrow tip clearance, the tip leakage flow interacts with the mainstream and rolls up into a tip leakage vortex (TLV), which is dominant near the tip region (Lakshminarayana, 1996). In the case of pumps operating close to atmosphere pressure, TLV is most prone to generating cavitation as a consequence of the inherent
low pressure in the vortex core (Arndt, 2002; Higashi et al., 2002; Murayama, 2006). Thus, cavitation inception is often coupled with vortex structures, such as tip leakage, trailing and hub vortices as well as turbulent vortex structures developing in regions of flow separation.

In the aforementioned research, much attention has been paid to the tip leakage vortex cavitation around the fully-wetted hydrofoils (Boulon et al., 1999; Park et al., 2006; Dreyer et al., 2014). Although many progress has been made in regard to the vortex structures and self-induced cavitation, some important parameters are still missing in actual hydraulic machines, for instance, the body force in rotating frame which can alter the cavitation phenomenon. As a result, for the practical applications, the numerical and experimental investigations have been carried out in many rotating machines, both for single and multiphase flows. Farrell and Billet (1994) measured the important variables for the correlation which predict the vortex minimum pressure in a high Reynolds number axial-flow test rig. Moreover, an optimum tip clearance has been theoretically identified as experiments have shown. Afterwards, a combined study of tip clearance and tip vortex cavitation in an axial flow pump was carried out by Laborde et al. (1997). A conclusion is drawn that cavitation inception is determined for various gap heights, clearance and blade geometries, and rotor operating conditions. Then, Kim et al. (2007, 2009) investigated the influence of tip clearance on the performance of the axial-flow-type waterjet propulsion based on a standard analysis method. Later on, Wu et al. (2011) focused on the flow structures in the tip region of a water-jet pump rotor, including the tip-clearance flow and rollup process of a tip leakage vortex (TLV). With the same facility, Wu et al. (2011) used Stereo particle image velocimetry (SPIV) to measure the flow structures and turbulence within the tip leakage vortex (TLV). Recently, concerning the cavitation breakdown in the aft part of the flow passage, Tan et al. (2015) found that a vortical cloud cavitation at the trailing edge of the sheet cavitation near the tip region is induced by TLV initially, and then oriented in a direction that is nearly perpendicular to the blade SS surface. Once takes over the flow passage, it would result in the severe degradation of the pump performance.

Computational fluid dynamics, predominantly Reynolds-averaged Navier-Stokes (RANS) computations but also large-eddy simulations (LES), provide insightful and detailed information about TLV dynamics, far beyond from analytical models. Using the shear stress transport (SST) $k$-$\omega$ along with a refined high-quality structured mesh, Zhang et al. (2015) studied the tip leakage vortex trajectories and dynamics in an axial flow pump under different operating conditions. With the increase of the flow rate, the TLV origin moves further downstream and the angle between the vortex core line and blade tip decreases. Though two-equation models can predict the general trend on TLV structures, over-prediction of the eddy viscosity has great limitations in the vortex shedding at the blade exit. Therefore, many advanced turbulence models with the modified eddy viscosity has sprung out (Wu et al., 2005; Ji et al., 2012; Liu et al., 2014). Zhang et al. (2015) simulated the formation of three-dimensional tip leakage vortex cavitation cloud and periodic collapse of TLV induced suction-side perpendicular cavitating vortices (SSPCV) using an improved SST $k$-$\omega$ turbulence model and the homogeneous cavitation model, which is consistent with the observation reported by Tan et al. (2015). Cavitation related flow phenomenon is one of the least-understood aspects of the tip-leakage flows due to the unsteady velocity and pressure which should be measured or computed spatially and temporally. Hence, the large-eddy simulation provides the most promising and feasible alternative to conventional numerical methods. You et al. (2006) investigated the effect of tip-gap size on the tip-leakage vortical structures and velocity and pressure fields using LES computation. Simultaneously, the mechanisms for viscous losses in a turbomachinery tip-leakage flow was revealed by You et al. (2007). He concludes that the velocity gradients are the major source for viscous losses in the cascade endwall region, and an approach which can alleviate the viscous losses through changing the direction of the tip-leakage flow was proposed. Besides, detailed steady and unsteady numerical simulations were performed to investigate tip clearance flows in an axial water jet pump by Hah (2012). The results show that the steady pressure field and resulting steady tip leakage vortex from a steady flow analysis do not account for the measured cavitation inception correctly because of the unsteady flow fields and highly transient cavitation inception. LES method which can calculate unsteady vortex motion is necessary to predict cavitation inception.

With special emphasis on the trajectory and dynamic of PCVs at the blade trailing edge near the tip region, several visualized experiments coupling with the numerical computations were carried out, to reveal the effects of the flow coefficient, blade number and blade tip geometry on this unstable PCVs, in terms of the spiral motion and how it interacts with the adjacent blade. By the correlated analysis of the three-dimensional tip cavitating vortices, an expectation is that more effective approaches can be proposed to control the large-scale structure in order to improve the overall performance of pumps.
2. Pump geometry

The axial flow pumps with high efficiency and low head are extensively applied in many large-scale projects. The pump model in this study has two impellers with different blade numbers, but the same geometry. The numerical details about the geometrical parameters of the pump model was shown in Table 1 ($M_e$ is the torque (N. m), $\nu$ is the kinematic viscosity of water (m$^2$/s) and $d_{\text{shaft}}$ represents the diameter of rotating shaft (m)). The rotor diameter is 198.0mm, i.e. the nominal tip-clearance width for a perfectly centered rotor is 1.0mm. However, direct measurements, showing that the non-uniform tip-clearance width varies from blade to blade, with an average value of 1.0mm. The sketch of the model pump design was presented in Fig.1a while Fig.1b is the geometry of blade. The relevant parts mainly consist of the impeller with three/four blades, the guide vane with seven blades, the ribs supporting the stator and the discharge elbow pipe. Fig.2 shows the computational parameters of the blade rotor geometry. Plains relative to the blade are defined by $sc^{-1}$, where $s$ is a coordinate aligned with the blade chord and $c$ is the chord length. Moreover, the blade thickness fraction represents the distance from the entrance to the exit of tip clearance, defined by $\zeta L^{-1}$, where $L$ is the tip thickness and $\zeta$ is a changeable location from the pressure side. $\gamma$ represents the axial coefficient from the inlet to the outlet of blade while $r^*$ is a radial coefficient from the hub to the casing wall.

<table>
<thead>
<tr>
<th>Number of rotor blades</th>
<th>3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of stator blades</td>
<td>7</td>
</tr>
<tr>
<td>Chord length [$c$]</td>
<td>113.7mm</td>
</tr>
<tr>
<td>Rotor diameter [$D_2$]</td>
<td>198mm</td>
</tr>
<tr>
<td>Casing diameter [$D$]</td>
<td>200mm</td>
</tr>
<tr>
<td>Tip clearance [$h$]</td>
<td>1.0mm</td>
</tr>
<tr>
<td>Clearance ratio [2$hD$]</td>
<td>0.01</td>
</tr>
<tr>
<td>Rotor angular velocity [$\Omega$]</td>
<td>151.84 rad s$^{-1}$ (1450rpm)</td>
</tr>
<tr>
<td>Tip velocity [$U_{\text{tip}}$]</td>
<td>15.03m s$^{-1}$</td>
</tr>
<tr>
<td>Flow rate [$Q$]</td>
<td>0.101m$^3$ s$^{-1}$</td>
</tr>
<tr>
<td>Head rise [$H$]</td>
<td>3.02m</td>
</tr>
<tr>
<td>Efficiency [$\eta=\rho g HQ(M_e\Omega)^{-1}$]</td>
<td>0.712</td>
</tr>
<tr>
<td>Flow coefficient [$\phi=2\pi Q\Omega^{-1}D^{-3}$]</td>
<td>0.524</td>
</tr>
<tr>
<td>Head coefficient [$\psi=(2\pi)^2gH(\Omega D)^{-2}$]</td>
<td>1.27</td>
</tr>
<tr>
<td>Blade chord Reynolds number [$Re_c=U_{\text{tip}}c\nu^{-1}$]</td>
<td>$1.71\times10^6$</td>
</tr>
<tr>
<td>Tip clearance Reynolds number [$Re_h=U_{\text{tip}}h\nu^{-1}$]</td>
<td>$1.5\times10^4$</td>
</tr>
<tr>
<td>Inlet flow Reynolds number [$Re_{IN}=4Q(\pi(D^2-d_{\text{shaft}}^2))^{-1}D\nu^{-1}$]</td>
<td>$6.63\times10^5$</td>
</tr>
</tbody>
</table>
In order to clarify the influence of blade geometry on tip vortex cavitation, especially on the PCVs, the pressure edge in one of blades is rounded with a small radius. Although some flow patterns differ from that case with all rounded blades, the gross feature can be captured by the observations. A first involved clearances with shape edges. The clearances of the second blade have a rounded edge on the pressure side and a sharp edge on the suction side. The schematic drawing of clearance geometries is shown in Fig.3.

3. Facility and experimental setup

The axial-flow pump is located in a closed test loop facility shown in Fig.4. The operation of the whole facility is driven by a precision-controlled motor, which is located outside of the loop. It is connected to a shaft consists of an encoder and a torquemeter. The shaft penetrates into the pipe with the flexible coupling, passes through a fixed stator and is connected to the axial-flow pump rotor. The test loop mainly includes a gate valve, a butterfly valve, a turbine flow meter, a boosting pump and a water tank. In the process of the visualized experiments, a transparent outer casing made up of Perspex is horizontally installed in the test section of the system. The water tank with high storage capability is situated in the upstream of the inlet and connected to a vacuum pump, which functions to keep the temperature of the water and flow steady. A reduction in the static pressure in the test section leads to the vaporization of the vortex core by the vacuum pump under different cavitation conditions. A honeycomb wall is installed at the middle section of the water tank to improve the flow uniformity and reduce the scales of turbulence of the incoming flows. Owing to the low head of the model pump, the boosting pump is necessary to extend the range of the flow rate which was measured by a turbine flow meter. The inlet pipe diameter of the loop is $\phi 200$mm, while the outlet is $\phi 250$mm. Two pressure transducers are installed on test section respectively, for collecting the pressure difference in the entrance and exit of the model pump. Finally, the overall measurements were transmitted to a signal acquisition system to analyze the results.
In this section, the external performance measurements and visualized experiments were illustrated thoroughly. Under no-cavitating condition, before the performance test, the motor should be unloaded to adjust the torquemeter to zero. Then through the flux regulation from the high flow rate to the unsteady operations, the correlated parameters were obtained by the acquisition system. Due to the lower head of the model pump compared with the centrifugal pump, repeatable experiments were necessary to reduce the discrepancy. During the experiments, when reaches the hump region, the model pump generates the high vibration and noise, which should be avoided in the industrial applications.

As shown in Fig.5, the transparent outer chamber has a shape of roundness inside but squareness outside, and was used to visualize the cavitation patterns and unsteady characteristics of the flow fields in the tip region. The high speed imaging system includes a high-speed digital video camera, a macro lens, fill lights and an imaging acquisition computer. The high-speed digital video camera applied in this experiment is IDE Y-series 4L with the sampling frequency of 5000Hz and exposure time of 107μs. One 750W continuous halogen light was used for illumination at high frame rates. The imaging calibration was performed using a picture posted on the shooting area of the casing. Afterwards, the focus and exposure time of the camera were adjusted slightly to obtain obvious cavitation patterns near the blade tip region. Cavitation can be a convenient way to visualize the trajectory of vortices as reported by Chang et al. (2007). A reduction in the static pressure in the test section leads to the vaporization of the vortex core, which makes it visible. However, as the vortex core is filled with vapor completely, the vorticity would redistribute, resulting in alternations of the flow dynamics possibly.

**Fig.4 Schematic and overall dimensions of the axial-flow pump loop**

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**Fig.5 Visualized experiments**

4. Results and discussion

4.1. Effect of flow coefficient

On the basis of previous work, the effects of flow coefficient on TLV trajectory and strength were investigated extensively. Yoon et al. (2006) demonstrated that tip vortex becomes strengthened and migrates upstream as the flow coefficient decreases, which was also validated by Zhang et al. (2015) via numerical simulations. In addition, under
cavitation condition, Tan et al. (2014) found that the evolution of TLV becomes more abrupt, in terms of the initial location and breakdown process near the pre-stall condition. Furthermore, using high-speed imaging, Tan et al. (2012) also discovered that the sheet cavitation is thin at BEP condition, while it becomes thickened as a result of an increase in the incidence angle of the flow entering the passage relative to the blade.

Fig.6a shows hydraulic performance of the pump model with 3 blades (BEP is under rated condition). The general trend of the head and efficiency increases initially and then drops abruptly. Near the hump region, the head has a transition obviously. When operates under this unstable condition, the pump model generates severe vibration and noise, which should be avoided in the engineering applications. The cavitation performance for various flow rates were presented in Fig.6b. A typical cavitation number is defined as \( \sigma = \frac{P_{in} - P_v}{0.5 \rho U_{tip}^2} \), where \( P_{in} \) is the pressure in the inlet of pump and \( P_v \) is the saturated pressure. Under the condition with lower cavitation number, an abrupt drop of the head coefficient occurs and serious noise and vibrations are also generated in the visualized experiments. For a specific flow condition, there is a wide range of inlet pressure where the performance has a little change. However, when reaches a certain cavitation number, the performance of the pump drops apparently, induced by the PCVs shedding and breakdown. By the comparative analysis of the cavitation performance curves, it is evident that the cavitation breakdown occurs earlier for the small flow rate condition. Fig.7 shows the cavitation inception for different flow rate conditions in this pump model. Cavitation inception always occurs when local pressure reaches to a level and it is captured by using visual observation in this research. According to the observations, the cavitation inception emerges earlier for the small flow rate condition (Label A), and it primarily consists of TLV cavitation due to the low pressure in the vortex core, the shear layer cavitation that connects with TLV consisting of a series of vortex filaments shedding from the tip clearance and tip clearance cavitation in the narrow clearance. Additionally, the corner vortex cavitation induced by the flow separation near the pressure side was also shown. The flow structure associated with these cavitation patterns in the tip region was presented in Fig.8. Under the small rate condition, as a consequence of increased blade loading, the leakage jet has high momentum and the interactive position with the mainstream shifts away from the blade suction side. Moreover, the pressure in the primary vortex core is very low that is most prone to generating cavitation.

![Fig.6 Performance curves](image-url)
The influence of flow coefficient on the PCVs trajectory and dynamics were not analyzed clearly before, with a special emphasis on the interaction between the PCVs and adjacent blade. The following figures, depicting an instantaneous snapshot, portrays the three-dimensional cavitation patterns and the shedding of cloud cavity, which is believed to reflect the interaction between the TLV and the vortices emanating from the suction side of the rotor. The shedding of vortices in the wake of triangle cloud cavitation are oriented largely perpendicular to the blade suction side, which is well known as “perpendicular cavitating vortices”. When the cavitation number is reduced to a certain level, the TLV cavitation, tip clearance cavitation and shear layer cavitation emerges into a triangle cloud cavitation, which consists of two instinctive parts: a relative stable cavity near the leading edge and fluctuating cloud cavitation at the aft part of the blade tip. At this region connecting with the blade sheet cavitation trailing edge, PCVs generates caused by the TLV as well as the pushing of tip leakage flows, and its orientation gradually becomes perpendicular to the blade suction side. When it migrates further downstream, the breakdown phenomenon occurs, bringing about a large number of small-scale cavities, which occupy a large area of the flow passage.

In Fig.9a, under small flow rate condition, the incidence angle of flow entering the passage relative to the blade increases, and with it, the blade loading. As a result, the backward leakage flow also increases, further increasing the incidence angle relative to the blade in the tip region. Influenced by the TLV trajectory, the relative inclination angle of the PCVs (Label A) is large, nearly perpendicular to the blade suction side. Then, the TLV interacts with the primary PCVs and emerges as a highly unstable structure, shedding from the blade trailing edge. The initial nearly continuous TLV cavitation terminates in a trail of bubbles and then vanishes, which condenses to the entrainment by the PCVs and a rise in pressure in the vortex core. Actually, the TLV in this region never rolls up into a distinctive single structure and contains multiphase interlacing vortex filaments (Wu et al., 2011a; Wu et al., 2011b; Wu et al., 2012; Miorini et al., 2012). When migrates away from the blade suction side, two cavity structures generate as the PCVs is cut off by the neighboring blade (Label B). One is the residual cloud cavity (Label C), directly impacts on the pressure side of adjacent blade and breaks down abruptly. The other structure (Label D) as the nuclei of the tip vortex cavitation near the
blade leading edge expands rapidly due to the relative low pressure on the blade suction side. These two highly destructive processes are periodic that should be controlled for triggering flow instabilities on the neighboring blade.

With the increase of flow rate, the relative angle between the PCVs and blade suction side decreases on account of TLV trajectory. Consequently, the phenomenon of cut-off by the incoming blade would not happen, and the breakdown of PCVs just imposed on the pressure surface directly. It is also obvious that a relative loose structure emerges compared with that under the small flow condition. Moreover, further increase of the flow coefficient, the orientation of PCVs is almost parallel to the blade suction side, and bursts into a large fraction of the small-scale vortex filaments immediately at the exit of blade. Therefore, the trajectory and structure of PCVs and its interactive process with the adjacent blade is ascribed to the TLV trajectory, which depends on the operating conditions significantly.

Fig.9 Instantaneous motion of the PCVs for different flow rates. (The dash line represents the TLV trajectory while part of triangle cloud cavitation is shown by the solid line; the time interval between two consecutive images is 2ms)
4.2. Mechanism for radial re-entrant jets

In order to describe the detailed flow structures in the cavitating vortical flows near the tip region, the numerical simulations are necessary as complements to the experiments. The numerical computations presented in this study were same with that in reference (Zhang et al., 2015), in aspects of the block topology and mesh arrangement in the tip region as well as the numerical setup. Moreover, an improved Filter-based model (Density Correction of Filter Based Model, DCMFBM) (Zhang et al., 2015; Huang et al., 2014) and a homogenous cavitation model were used to accomplish this work.

4.2.1. Governing equations

In the uniformity assumption of the mixture of water and vapor in the cavitating flows, the multiphase fluid components are assumed to share the same velocity and pressure. The continuity and momentum equations for the mixture flow are given by

\[
\frac{\partial \rho_m}{\partial t} + \frac{\partial}{\partial x_j}(\rho_m u_j) = 0
\]  

(1)

\[
\frac{\partial}{\partial t}(\rho_m u_i) + \frac{\partial}{\partial x_j}(\rho_m u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[(\mu + \mu_t)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \frac{\partial u_k}{\partial x_k}\right)\right] - \frac{2}{3} \frac{\partial u_k}{\partial x_k} \frac{\partial \rho}{\partial x_j}
\]  

(2)

where \( u_i \) is the velocity component in the \( i \) direction, \( p \) is the mixture pressure, \( \mu \) is the laminar viscosity and \( \mu_t \) is the turbulent viscosity. The mixture density \( \rho_m \) is defined as \( \rho_m = \alpha_l \rho_l + \alpha_v \rho_v \), where the subscripts, \( l \) and \( v \), represent liquid and vapor phase respectively, \( \alpha \) is the volume fraction of one component.

4.2.2. Improved filter-based model

The standard RNG \( k-\varepsilon \) model is not able to predict the cavitation shedding dynamics accurately as it was developed for the fully incompressible single-phase flow, and the cavitating flow is actually compressible two-phase flows. As shown by Reboud et al. (1998), the standard RNG \( k-\varepsilon \) model fails in simulating the unsteady cavitation due to the over-prediction of eddy viscosity at the rear party of the cavity. To improve the simulation accuracy by considering the compressibility of cavitating two-phase flow, some modifications of the eddy viscosity are introduced as

\[
\mu_{t,\text{Mod}} = C_\mu \frac{\rho_m k^2}{\varepsilon} F, \quad F = \min \left[f(n), C_3 \frac{\Delta}{l_{\text{RANS}}} \right]
\]  

(3)

\[
f(n) = \frac{\alpha_l \rho_l + \alpha_v \rho_v}{\alpha_l \rho_l + \alpha_v \rho_v} \left(\rho_l - \rho_v\right)
\]  

(4)

where \( C_\mu = 0.085 \), \( C_3 = 1.0 \) and \( n=10 \) (Coutier-Delgosha et al., 2003). The turbulence size and filter size are the most important factors of the modified turbulence model. The turbulence size can be written as \( l_{\text{RANS}} = k^{2/3}/\varepsilon \), and the filter size depends on the grid size. The grid size \( \Delta = (\Delta x \Delta y \Delta z)^{1/3} \) in the tip region was used. According to some references (Shi et al., 2011, Yu et al., 2010) and a lot of validations, the filter size should be a little large than the grid size, \( \Delta = 1.01 \Delta \) was selected in the following simulations.
4.2.3. Cavitation model

Cavitation model used in this paper is proposed by Zwart-Gerber-Belamri (2004). It was based on the hypothesis that all the bubbles in a system have the same size. The cavitation process is governed by the mass transfer equation for the conservation of the vapor volume fraction, which can be expressed as

\[
\frac{\partial (\alpha \rho)}{\partial t} + \frac{\partial (\alpha \rho \mathbf{u})}{\partial x_j} = \dot{m}^+ - \dot{m}^-
\]  

(5)

The source terms for the specific mass transfer rate corresponding to the vaporization \( \dot{m}^+ \) and condensation \( \dot{m}^- \) are given by:

\[
\dot{m}^+ = C_v \frac{3 \rho (1 - \alpha) \alpha_{nuc}}{R_b} \sqrt{\frac{2 p - p}{3 \rho}} \quad (p \leq p_c)
\]  

(6)

\[
\dot{m}^- = C_v \frac{3 \rho \alpha}{R_b} \sqrt{\frac{2 p - p}{3 \rho}} \quad (p \geq p_c)
\]  

(7)

where \( C_v \) and \( C_e \) are empirical coefficients for the different phase change processes, \( \alpha_{nuc} \) is nucleation site volume fraction, and \( R_b \) is the typical bubble size in water. These empirical constants were set to \( C_v = 50 \), \( C_e = 0.01 \), \( \alpha_{nuc} = 5 \times 10^{-4} \) and \( R_b = 1 \times 10^{-6} \) m based on the work of validations using 3D cases, such as the cavitating flow around the three-dimensional hydrofoil and pumps (Ji et al., 2014, Guo et al., 2015).

4.2.4. Unsteady cavitating flows and in-plain re-entrant jets

Fig. 10 shows the comparison of time-averaged performance of the pump model, including the hydraulic variations and cavitation performance. A large discrepancy is evident under the small flow rate condition because of the flow unsteadiness. While far away from the unstable region, the error between the simulations and experiments is in a proper range. When the pump operates under the cavitation condition, the performance changes significantly due to the flow structures affected by the cavity shedding and breakdown. The head coefficient has a little change initially, and a slight rise before the breakdown. The phenomenon of head rise can be ascribed to the blade loading modification induced by cavitation probably (Brennen, 2001). After that, the PCVs sheds from the blade trailing edge and breaks down in the flow passage, resulting in the severe degradation of the pump performance. In general, the numerical computations agree well with the experimental data, and the acceptable error can be induced by the limitations of turbulence model and homogeneous cavitation model, etc.
Fig.11 presents the transient process of unsteady vortex cavitation in the tip region of the pump model. The gross feature of the cavity shedding and breakdown agrees well with the experiments. However, the entrainment process is ambiguous in the simulations. In the course of the PCVs development, at the tail of TLV strip, the vortical cloud cavitation is entrained and re-oriented by the TLV in the direction that is nearly perpendicular to the blade SS surface. When the shedding occurs, the TLV would also terminates in a trail of bubbles and realigns with the PCVs, then migrates further downstream finally.

Are there any contributors that affect the shedding of the large-scale cavity? In reality, as reported by Tan et al. (2012, 2015), the aft part of the sheet becomes frothy in the vicinity of the blade tip, and periodically sheds cloud cavitation. The inherent instability of the sheet cavitation trailing edge on two dimensional lifting surface is well studied (Gallenaere et al., 2001, Franc and Michel, 1985; Le et al., 1993; Reisman and Brennen, 1996; Arndt et al., 2001). It includes formation of a periodic, backward-flowing, near-wall re-entrant jet, which flows under the cavity and pinches part it off to produce a large scale cavitating vortical cloud. In the case of the 3D lifting surface, e.g. a swept back stationary hydrofoil bounded on both sides. Laberteaux and Ceccio (2001) observed that the laterally directed re-entrant jet eventually impinged on the cavity surface near the wall, leading to the formation of an open cavity with shed bubbly clouds. Recently, Ji et al. (2013) used a Partially-Averaged Navier-Stokes method and a homogeneous cavitation model to simulate the cavitating turbulent flow around a twisted hydrofoil. The results show that shedding horse-shoe vortex includes a primary U-shape vapor cloud and two secondary shedding. The former is induced by a radially-divergence re-entrant jet while the collision of side-entrant jets and the radially-diverging re-entrant jet results in the latter.

In this test case, with the aid of the numerical simulations, the detailed information concerning the formation and development of the re-entrant jets would be presented in Fig.12. The three-dimensional cavity patterns near the tip region is apparent, including the TLV cavitation, PCVs and small-scale shedding cavities at the blade trailing edge. To clarify the mechanism for the re-entrant jets generation and evolution clearly, three cross sections at different chord fractions \( \lambda = 0.9, 1.1 \) and 1.3 respectively, were established with the vapor fraction and velocity vectors. An assumption with respect to the radial Re-entrant jets formation can be concluded as follows: 1) the obvious hub vortices characterizing wing-body junctions plays an important role possibly; 2) the mainstream impinges on the sheet cavitation, leading to the formation of the component velocity in direction that from the hub to the blade tip. The cavity fluctuates and separates from the trailing edge of the sheet cavitation near the tip region as the movement of the velocity component. Simultaneously, the re-entrant jets will be entrained into the leakage flows. Three plains located in various positions show the re-entrant jets direction and strength. Evidently, the re-entrant jet in plain S2, which is suited at the region where a large cluster of different scales cavity gather together, is more intensive compared with that in other two plains. Further far away from the blade tip, as shown in Fig.12d, the re-entrant jets almost vanish. Therefore, it deduces that at a specific position of the blade exit, a strong radial re-entrant jet leads to the cloud cavitation shedding, then entrained by tip leakage flows. After that, the shedding cloud cavitation would be entrained and re-oriented by TLV. During the process, the hub vortices becomes a critical contributor, which is in accordance with the observations reported by Tan.

![Fig.10 Performance curves.](image-url)
Fig. 11. Instantaneous evolution of PCVs. $Q/Q_{\text{BEP}}=1.0$, $\sigma=0.217$. Up: Isosurface of vapor fraction, $\alpha_v=0.1$; Down: Experimental results. (TLV trajectory is shown by the dash line)

(a) Tip vortex cavitation structure
4.3. Effect of blade number

In the design of an axial-flow pump with high efficiency, blade number is a significant factor both for the hydraulic and cavitation performance. With the increase of the blade number, the trajectory and dynamics of PCVs vary considerably because of the narrow flow passage. Fig.13 presents the large-scale cloud cavitation patterns for different flow rates and cavitation numbers in the pump model with four blades. Compared with that in the three blades, the trajectory and dynamics of the PCVs in the narrow channel has several features shown as follows: 1) under the small...
flow rate condition, when reaches to the neighboring blade, the PCVs would not be cut off by the blade, and it transports the cavitation nucleus into the tip clearance of the incoming blade (Label B), causing a rapid development of the cavitation in the blade leading edge (Label C). However, in the impeller with three blades, the PCVs will be cut off by the adjacent blade and divided into two parts: one is above the pressure surface while the other is under the suction side. The latter develops rapidly due to the low pressure on the suction side. Simultaneously, the cavitation cloud above the blade pressure surface breaks down and then produces high pressure, promoting the cavitation in the leading edge. 2) when the flow coefficient increases, the PCVs gradually generates two cavitation cloud structures, moving along two different paths. The branch would migrates toward the blade trailing edge, and eventually interacts with the shedding cloud cavity, affecting the formation of the PCVs in the next blade (Label D). 3) as the flow rate increases further, the leading edge of the pressure side near the blade tip also emerges cavitation (Label E), possibly due to the large decrease of pressure rise. Moreover, the weak PCVs moves in the direction that is nearly parallel to the pressure side of the next blade. Once gets to the blade exit, the PCVs would mix up with the next one and then becomes a new structure. 4) in Figure.13d, under a severe cavitation condition, a strong PCVs approaches the blade trailing edge, and influences the flow structures near the next flow passage. Additionally, the region of leading edge cavitation on the pressure side extends toward the middle chord. As shown by Label F, the cavitation also appears on the guide vane inlet. During this operating condition, the noise and vibration is generated clearly, as well as the performance degradation because of the flow passage blockage.

In general, the trajectory and intensity of PCVs are determined for various operating conditions strongly in present configuration. When given the blade number into consideration, the runner width and blade loading also have a great impact on the PCVs evolution.

Fig.13 Cavitation patterns for various flow parameters and cavitation numbers.
4.4. Effect of blade tip geometry

As a critical parameter, blade tip shape also has great impact on the TLV trajectory and strength. A number of investigations, where a reduction of the tip-leakage flow related detrimental effects was attempted through the change of blade tip shape, were carried out by experiments and numerical simulations (Laborde et al., 1997; Heyes et al., 1992; Nho et al., 2012). In this work, for the sake of illustrating the effects of blade tip shape on cavitation inception and PCVs dynamics for different flow conditions, the clearance edge was rounded on the pressure side of one blade in the rotor. As was demonstrated by many previous research, the rounded pressure side can effectively eliminate clearance cavitation. However, the increased amount of leakage flux would lead to more severe tip leakage vortex cavitation. Moreover, the TLV trajectory also affected by the interactive location between the leakage flows and the mainstream. Hence, the question whether the PCVs dynamics is also under the influence of the blade tip shape is worthwhile.

Fig.14 shows the cavitation inception for two test cases under the same experimental conditions. Visual observations shows that cavitation inception greatly depends on clearance geometry. In the condition of the pump model operating far higher than barometric pressure, cavitation inception occurs near the tip leading edge. Cavitation inception in the tip region of the tested axial-flow pump with sharp clearance edge seems more slight compared with that in the rotor with rounded pressure side edge, and mainly consists of TLV cavitation, corner vortex cavitation, tip clearance cavitation and the shear layer cavitation. The mechanisms for these different types of the cavitation were revealed by Zhang et al. (2015). When the clearance was rounded on the pressure side, the corner vortex cavitation and tip clearance cavitation were eliminated, as was shown by the dash line in Fig.14b. Additionally, the more intense TLV cavitation also appears, induced by the lower pressure in the vortex core. Thereby, if the TLV is mainly responsible for the pump performance, this method should not be accepted.

![Cavitation inception patterns](image)

Fig.14 Cavitation inception patterns. ($Q/Q_{REF}=1.0$, $\sigma=0.982$)

Some snapshots depicting the specific characteristics of the PCVs evolution were selected in Fig.15, involving the trajectory and dynamics in the process of wandering motion. The initial formation of PCVs is due to the existence of the radial re-entrant jets and induced by TLV. Clearly, there is a substantial reduction in the flow through the tip region of the rotor passage once the PCVs emerge. As was indicated by the casing pressure measurement, Tan et al. (2015) found the blade loading near the tip diminishes with a low relative flow. In this study, it is also obvious that the attached cavitation in the tip clearance disappears with little relative flow, as the area highlighted by the dash line in Fig.15a. During the early stage of PCVs formation, it reduces the pressure on the SS of the blade at mid-chord, increasing the blade loading there. However, with the development of the PCVs, the blade loading near the aft part of the blade chord decreases, as a result of the pressure rise on the blade suction side. These phenomenon indicate that once the PCVs take over, the blade work in the tip region diminishes. Compared with that in the clearance edge rounded on the pressure side, the PCVs changes significantly in the tip region with shape edges. Severe tip clearance cavitation is observed when the clearance has sharp edges while no cavitation is presented in the clearance with a rounded edge on the pressure side. Moreover, it seems that the PCVs has a loose structure in the tip region of the blade with sharp edges and its strength is weaker, as shown in instant 3. Consequently, more small-scale cavitating vortices connecting with the primary would shed, and leads to the initial interaction with the adjacent blade leading edge (Label A). When the main cloud cavitation reaches to the incoming blade, the second interaction would happen, which has been discussed in the above section. On the contrary, a compact and more intensive structure was created at the trailing edge of the attached cavitation near the tip region with rounded clearance edge. During the migration, it is relative stable and then cut off by the next blade. It concludes that the PCVs formation is more violent and its strength is also more intensive when the
clearance edge is rounded on the pressure side, which should be ascribed to the effects of TLV strength presumably. Additionally, due to the TLV trajectory and its strength, the PCVs also has different movement paths for two test cases.

(a) clearance with sharp edges

(b) clearance with rounded edge on the pressure side

Fig. 15 Evolution of PCVs for two blade tip shapes. ((Q/Q_{BEP}=0.8, \sigma=0.340)

5. Conclusions

This paper deals with the effects of operating condition, blade number and blade tip shape on the trajectory and dynamics of PCVs in the tip region of an axial-flow pump using the high-speed photography. The visualized observations show that the PCVs evolution, in terms of the trajectory, strength and interaction with the adjacent blade, vary considerably for various parameters. The most significant results can be summarized as follows:

1. Cavitation inception for small flow rate occurs more early, and it mainly consists of TLV cavitation, corner vortex cavitation, tip clearance cavitation and shear layer cavitation. A combined effects of TLV and radial re-entrant jet from the hub to the blade tip on the formation of PCVs at the trailing edge of the attached cavitation near the tip region were clarified. Under the small rate condition, the PCVs trajectory is almost perpendicular to the blade suction side. When moves downstream, it would interacts with the incoming blade and be divided into two parts: an important proportion becomes the nucleus for the tip vortex cavitation on the next blade, while the remained directly impacts on the pressure side, triggering the flow instabilities on the pressure side. With the increase of the flow coefficient, the PCVs migrates in the direction that is nearly parallel to the blade suction side, and collapses near the blade exit immediately.
In this study, an improved turbulence mode coupled with a homogenous cavitation model were used to identify the existence of re-entrant jet. An assumption is that its formation can be ascribed to the role of hub vortices and the velocity component of the mainstream.

Compared with that in the rotor with three blades, the PCVs evolution is quite different due to the narrow flow passage and blade loading distributions. In the process of moving towards the pressure side of the next blade, both the main cloud cavitation and relative small-scale cavity shedding from the primary would provide nucleus for the cavitation on the next blade. Under the large flow rate condition, the PCVs of one blade even interacts with another at the trailing edge of next blade.

The blade with a rounded edge on the pressure side can effectively eliminate the corner vortex cavitation and tip clearance cavitation. Moreover, it shows that once the PCVs takes over, the cavitation at the aft part of the blade chord disappears, as a consequence of the substantial reduction in the flow. A compact and more intensive PCVs is generated when the clearance edge is rounded on the pressure side. Conversely, in the original one, owing to the presence of some relative small-scale cavity separating from the main cloud, the phenomenon of multiple interaction with the adjacent blade would happen, affecting the flow fields in the tip region of next blade strongly.

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References

Huang, B., Wang, G. and Zhao, Y., Numerical simulation unsteady cloud cavitating flow with a filter-based density
Heyes, F. J. G., Hodson, H. P., Dailey, G. M., The effect of blade tip geometry on the tip leakage flow in axial turbine
cavitating flow around a marine propeller in a non-uniform wake, International Journal of Heat and Mass
Ji, B., Luo, X., Arndt, R. E. A. and Wu, Y., Numerical simulation of three dimensional cavitation shedding dynamics
Kim, M. C. and Chun, H. H., Experimental investigation into the performance of the axial-flow-type waterjet according
Laborde, R., Chantrel, P. and Mory, M., Tip clearance and tip vortex cavitation in an axial flow pump. Journal of Fluids
Le, Q., Franc, J. P. and Michel, J. M., Partial cavities: pressure pulse distribution around cavity closure, Journal of
Murayama, M., Unsteady tip leakage vortex cavitation originating from the tip clearance of an oscillating
Miorini, R. L., Wu, H. and Katz, J., The internal structure of the tip leakage vortex within the rotor of an axial waterjet
Nho, Y. C., Park, J. S., Yong, J. L., Kwak, J. S., Effects of turbine blade tip shape on total pressure loss and secondary
Park, K., Seol, H., Lee, S., Park, K., Seol, H. and Lee, S., Numerical Analysis of Tip Vortex Cavitation Behavior and
Noise on Hydrofoil, Proceedings of ASME 2006 2nd Joint U.S.–European Fluids Engineering Summer Meeting
Collocated With the 14th International Conference on Nuclear Engineering. American Society of Mechanical
Reboud, J. L., Stutz, B. and Coutier, O., Two-phase flow structure of cavitation: experiment and modeling of unsteady
Reisman, G. E. and Brennen, C. E., Pressure Pulses Generated by Cloud Cavitation, Proceedings of Fluids Engineering
three-dimensional hydrofoil by numerical and experimental methods, Engineering Mechanics, (2012), DOI:
Tan, D., Li, Y., Wilkes, I., Vagnoni, E., Miorini, R. L. and Katz, J., Experimental Investigation of the Role of Large
Scale Cavitating Vortical Structures in Performance Breakdown of an Axial Waterjet Pump, Journal of Fluids
Tan, D., Li, Y., Wilkes, I., Miorini, R. L. and Katz, J., Visualization and time-resolved particle image velocimetry
measurements of the flow in the tip region of a subsonic compressor rotor, Journal of Turbomachinery, Vol.137,
No.4 (2014).
Waterjet Pump Rotor, Proceedings of ASME 2012 Fluids Engineering Division Summer Meeting Collocated with
the ASME 2012 Heat Transfer Summer Conference and the ASME 2012 10th International Conference on


