Study on the characteristics of horn-like vortices in an axial flow pump impeller under off-design conditions

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ABSTRACT
The dominant vortex structure near the hub in rotor corner separation flow has an important influence on the internal flow of the axial flow rotating machinery. However, the current quantitative research on the key vortical characteristics of the dominant vortex structure in the corner separation flow is still insufficient. The analysis of the dominant vortex structure (horn-like vortex) in an axial flow pump was conducted and found that the average vortex intensity and turbulence eddy dissipation show a gradual weakening trend along the vortex core line from the corner region to the vortex tail under various working conditions. The evolution of the horn-like vortex in a life cycle includes five stages, inception-growth-development-attachment-decay in the typical working condition 0.62 \( Q_0 \). The statistical average frequency of the periodic evolution is about 2.76 times the shaft rotation frequency. The pressure fluctuation near the hub induced by the horn-like vortex increases rapidly and dominates in the flow channel. The average peak-peak value on the impeller outlet section close to the horn-like vortex is about 2.79 times that of the impeller inlet section and should be paid enough attention to.

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1. Introduction

Rotor corner separation refers to the backflow and accumulation of low-energy fluid on the suction surface of the blade-end wall corner region, which is an inherently unstable flow characteristic of axial-flow rotating machinery under small flow conditions. Zierke et al. analyzed the secondary flow features in the hub corner region of the axial flow pump through oil visualization experiments and found that there was significant flow separation, leading to the gradual deterioration of the flow (Zierke & Straka, 1971). Dring et al. analyzed the internal flow field of a compressor rotor with high blade loading and found that there is a high loss area related to corner separation near the hub. As the flow rate decreased, the region of corner separation expanded to the entire blade and the resulting high loss leads to a gradual deterioration of unit efficiency (Dring et al., 1982). Hah et al. gave the topological structure of the compressor corner separation, as shown in Figure 1, and found that there is a torsional complex flow in the corner region (Hah & Loellbach, 1999). Corner separation is one of the main reasons restricting the development of axial flow rotating machinery. At present, there are few studies on rotor corner separation in axial flow pumps, which will affect the safe, stable and efficient operation of large flow and low head pumping stations.

Based on a series of studies, Denton pointed out the basic mechanism of corner separation (Denton, 1993), refer to the adapted diagram of Figure 1, corner separation originates from the secondary flow in the flow channel. Driven by the lateral pressure gradient, the low-energy fluid in the flow channel moves to the hub corner region. The low-energy fluid at the corner region cannot resist the blocking effect of the reverse pressure gradient of the flow direction, so that a large amount of fluid accumulates at the corner region, forming a low-velocity reflux area surrounded by the end wall and the suction surface.

At present, scholars have discovered the dominant vortex structure near the hub in the corner separation flow of axial-flow rotating machinery. Ochi et al. (2009) studied the internal flow of the propeller and pointed out that as the flow rate decreases, the adverse effect of the vortex near the hub on the propeller thrust gradually increases (Ochi et al., 2009). Chaluvadi et al. pointed out that as the flow rate decreases, the additional loss caused by the vortex near the hub gradually increases (Chaluvadi et al., 2004). Ashton et al. studied the flow field of axial flow fans and found that as the flow changes, the unstable vortex near the hub aggravates the turbulence effect, reduces the thrust coefficient, and has an important impact on the evolution of the downstream wake and the fatigue load of the fan impeller (Ashton et al., 2016). Yan et al. analyzed the vortex near the hub in the corner separation flow of the axial flow compressor and found that as the flow rate decreases, the vortex blockage of the flow passage gradually increases (Yan et al., 2018). It can be seen that the change in flow rate further deteriorates the structural characteristics of the dominant vortex near the hub. Axial flow pumps often operate under variable conditions or small flow conditions. Therefore, the study of the changes in the structural characteristics of the dominant vortex near the hub when the flow changes is extremely important. Liu et al. and Dong et al. pointed out that the absolute swirling strength of the fluid and the turbulence eddy dissipation are important indicators for analyzing the vortex structure, reflecting the strength of the eddy current and the change of the eddy current loss (Dong et al., 2019; Liu et al., 2018). However, in the current research on axial flow pumps, the relationship between the change in flow rate and the above-mentioned indicators of the dominant vortex structure near the hub is still unclear, and further analysis is needed.

In addition, according to the research of Kan et al., Wu et al., Yang et al. on pressure fluctuation in axial flow pumps, at the optimal operating point, the pressure fluctuation from the hub to the tip shows a gradually increasing trend, but as the flow decreases, the closer the radial position is to the hub, the more sharply the pressure fluctuation changes (Kan et al., 2018; Wu et al., 2018; Yang & Liu, 2014). Wang et al. pointed out that high-amplitude pressure fluctuation is directly related to the generation of vortex structure (Wang, Wang, et al., 2021). Obviously, the rapid increase in pressure fluctuation near
the hub is closely related to the vortex structure near the hub generated by the corner separation flow. However, the specific relationship between the pressure fluctuation and the vortex structure near the hub is still unclear, and further research is needed to quantify the relationship between the pressure fluctuation and the vortex structure near the hub.

The following research questions are summarized through the above analysis.

Question 1: Under small flow conditions, with the change of flow rate, how does the key vortical characteristic of the dominant vortex structure near the hub of the axial flow pump change?

Question 2: The periodic evolution characteristics of the vortex structure and its relationship with pressure fluctuation?

The remainder of this research is organized according to the following structure. Part 2 introduces the numerical method and schemes. Part 3 provides details of experimental validation. In part 4, the key vortical characteristics of the dominant vortex structure in rotor corner separation flow are reported. In part 5, the periodic evolution of the vortex structure and its relationship with pressure fluctuation characteristics are analyzed. Finally, conclusions and prospects are put forward in part 6.

2. Numerical method and schemes

2.1. Turbulence model and governing equations

In this paper, the VLES model is used for subsequent flow numerical simulation. VLES model as a hybrid model is currently widely used in separation flow simulations (Han & Krajnović, 2013; Jin et al., 2019). On the basis of the RANS model, the resolution control function $F_r$ damped Reynolds stress is introduced to simulate the sub-grid-scale turbulence stress tensor (Han & Krajnović, 2013; Han & Krajnović, 2015). This method uses RANS to solve the flow region near the wall, and the mainstream region realizes the conversion from DNS to RANS by damping turbulent viscosity. Compared with the LES model, this method can achieve high-precision analysis of the flow field with fewer grids. Equation (1) shows the transport of $k$ (turbulence kinetic energy) and $\omega$ (turbulence eddy frequency) in the modified VLES model. This method has been used in different separation flow calculations and accurate prediction results have been obtained (Jin et al., 2019; Xia et al., 2019).

$$
\begin{align*}
\frac{\partial k}{\partial t} + U_j \frac{\partial k}{\partial x_j} &= \frac{\partial}{\partial x_j} \left[ \left( v + \frac{\mu_{\text{VLES}}}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + P_k - \beta' k \omega \\
\frac{\partial \omega}{\partial t} + U_j \frac{\partial \omega}{\partial x_j} &= \frac{\partial}{\partial x_j} \left[ \left( v + \frac{\mu_{\text{VLES}}}{\sigma_\omega} \right) \frac{\partial \omega}{\partial x_j} \right] + \alpha_2 P_k - \beta_1 \omega^2 \\
&\quad + 2(1 - F_1) \frac{\partial}{\partial x_j} \left( \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} \right)
\end{align*}
$$

In particular, the turbulent viscosity $\mu_t$ is given by

$$
\mu_t^{\text{VLES}} = F_r \cdot \mu_t^{\text{RANS}} = F_r \cdot \rho k/\omega
$$

$$
F_r = \min \left( 1.0, \left[ 1.0 - \exp \left( - \beta L_i/\Delta k \right) \right]^{\nu} \right)
$$

$$
L_c = C_k (\Delta x \Delta y \Delta z)^{1/3}, L_i = k^2/(\beta_1^* k \omega),
$$

$$
L_k = v^2/(\beta_1^* k \omega)^{1/2}
$$

$$
C_x = \sqrt{0.3} C_s / \beta_1^*, C_s = \sqrt{\left[ \left( C_{2,0} \Delta^2 |S|^2 + v^2 \right)^{1/2} - v \right]/\Delta^2 |S|},
$$

$$
\Delta = (\Delta_x \Delta_y \Delta_z)^{1/3}
$$

where $\Delta$ is the mesh scale, $L_k$, $L_i$, and $L_c$ are the Kolmogorov scale, integral length scale, and turbulence length scale, respectively. $n$, $\beta$, and $\beta_1^*$ are model constants which take values of 2, 0.09, and 0.002 respectively (Han & Krajnović, 2013; Han & Krajnović, 2015).

2.2. Computation domain and boundary conditions

The computation domain is shown in Figure 2, including 6 parts: the inlet extension part, elbow inlet channel, impeller, guide vane, S-shaped outlet channel, and outlet channel extension part. Total pressure inlet (1atm) and flow outlet (value depends on the operating conditions) as the boundary conditions of numerical simulation are adopted. In this paper, the transient numerical calculation is used, and the transient simulation results are based on steady-state calculation results and through the verification of three groups of time steps (the time steps are $2^\circ$, $1^\circ$ and 0.5$^\circ$ respectively), the time step is finally selected as $2^\circ$ of impeller rotation on the basis of comprehensive calculation accuracy and efficiency. At this time, the average courant number in the flow field is less than 5, which meets the calculation requirements pointed out by Shen et al. (2021). The interface between rotor and stator adopts the transient rotor-stator method. For the treatment of the wall surface, the non-slip wall surface condition is used and the influence of the wall surface roughness is considered. The conversion relationship between the surface roughness $R_s$ in engineering calculations and the equivalent roughness in the simulation refers to Adams et al. (2012). In the discrete format, the central difference format of the diffusion term, the second-order backward Euler format the transient term, and the second-order upwind format of the convection term are adopted. The transient calculation adopts the fully implicit coupling algorithm, and the residual convergence criteria is $1.0 \times 10^{-5}$. All calculations herein were undertaken using ANSYS CFX software on Intel 56 core workstations for subsequent simulation.
2.3. Grid generation and independence validation

The discretization of the grid is an important part of the numerical calculation (Abadi et al., 2020; Ghalandari et al., 2019). In order to ensure the quality of grid, hexahedral grid is used to discretize all parts in the computational domain. To ensure the accuracy and precision of the flow simulation in the impeller domain, the blade boundary layer grid was added, and the complex geometric positions in each computational domain were refined. In order to ensure the accuracy and efficiency of numerical simulation, the grid convergence index method is adopted to analyze the convergence of grids (Celik et al., 2008). According to the requirements of this method, three groups of meshing schemes were determined. The grid numbers were 9,328,140 (Case 1), 3,468,970 (Case 2), 1,474,511 (Case 3), and the grid refinement factors were approximately 1.39 and 1.33, respectively. The mean values of pressure coefficient $C_p$, radial velocity coefficient $V_r$, and the rigid vorticity ratio $R_v$ (the core parameter in the Omega vortex identification method) (Liu et al., 2019) of the middle section plane in the impeller domain and pump head $H$ are used as error evaluation parameters. The specific expressions of the main parameter are shown in Equations 6–8.

$$C_p = \frac{P - \bar{P}}{0.5\rho V^2}$$  \hspace{1cm} (6)

$$V_{rp} = \frac{V_u \cos \theta + V_v \sin \theta}{V}$$  \hspace{1cm} (7)

$$R_v = \frac{||\Omega||_F^2}{||D||_F^2 + ||\Omega||_F^2 + \varepsilon}$$  \hspace{1cm} (8)

where $P$, $\bar{P}$, $\rho$ and $V$ are pressure, average pressure, density of water and circumferential velocity, respectively. $V_u$, $V_v$ are the component of the velocity in the $x$ and $y$ direction in rectangular coordinates, and $\theta$ is the angle between the radial velocity and the $x$ direction. $\Omega^2_F$ and $D^2_F$ are the Frobenius norm of rotation rate tensor and strain rate tensor, and $\varepsilon$ is a positive small quantity to ensure that the denominator is not zero.

As shown in Table 1, $\emptyset_1 \sim \emptyset_3$ are the simulation values corresponding to the evaluation parameter, $|\varepsilon_{32}/\varepsilon_{21}|$ is the error ratio, $\emptyset_{21}^{\text{ext}}$ is the extrapolated value, $e_{\text{ext}}^{21}$ is the relative error of the extrapolated value, and GC$_{\text{fine}}^{21}$ is the grid convergence index. The error ratio of the above three variables ranged from 1.4167 to 1.8571. The relative errors of the extrapolated values ranged from 0.4236% to 2.4699%. The grid convergence index is ranged from 0.5273% to 3.0129%. Obviously, the meshing scheme meets the requirements of grid convergence, which can provide accurate numerical results. Therefore, the grid scheme of Case 1 is selected to carry out the subsequent flow calculation. The overall and local grids of Case 1 and the near-wall grids and $y+$ distribution are shown in Figure 3. 18 nodes are arranged in the boundary layer of the impeller domain and pump head $H$ are used as error evaluation parameters. The number of grids and nodes of each part is shown in Table 2.

2.4. Arrangement of pressure fluctuation monitoring points

To comprehensively study the amplitude change of pressure fluctuation in the corner separation flow, 216
Figure 3. Grids and $y^+$ distribution: (a) global grids (b) local grids (c) schematic diagram of the near-wall grid (d) $y^+$ distribution of impeller blade.

Table 1. Evaluation results of discretization error.

| Domain     | $C_p$ | $V_r$ | $R_s$ | $H/m$ |
|------------|-------|-------|-------|-------|
| $\phi_1$  | 0.0712| 0.1291| 0.5779| 7.1550|
| $\phi_2$  | 0.0724| 0.1298| 0.5821| 7.1930|
| $\phi_3$  | 0.0741| 0.1311| 0.5983| 7.2610|
| $|\sqrt{f_{32}/f_{21}}|$ | 1.4167| 1.8571| 1.4762| 1.7894|
| $\phi_{21}^{21}$ | 0.0695| 0.1286| 0.5725| 7.1231|
| $\varepsilon_{\text{rat}}$ (%) | 2.4699| 0.4236| 0.9412| 0.4484|
| GCI $\varepsilon_{\text{rat}}$ (%) | 3.0129| 0.5273| 1.1656| 0.5580|

Table 2. Final number of grids and nodes for each domain.

| Domain            | Number of grids | Number of nodes |
|-------------------|-----------------|-----------------|
| Inlet Extension Part | 253,820        | 266,400         |
| Elbow Inlet Passage | 1,273,728      | 1,346,322       |
| Impeller          | 3,822,720      | 3,701,360       |
| Guide Vane        | 2,581,712      | 2,472,855       |
| S-shaped Outlet Passage | 1,177,064 | 1,171,288       |
| Outlet Extension Part | 219,096       | 229,632         |
| Total             | 9,328,140      | 9,193,857       |

pressure fluctuation monitoring points are arranged in the flow channel as shown in Figure 4. The specific arrangement is as follows: six sections are arranged along the flow direction, the flow direction coordinate of the inlet of the impeller is 0, and the flow direction coordinate of the outlet is 1. The relative coordinates of the six flow direction sections are $St_w$ (Streamwise) = 0.01, $St_w$ = 0.2, $St_w$ = 0.4, $St_w$ = 0.6, $St_w$ = 0.8, $St_w$ = 0.99. Six sections are arranged in the span direction, the spanwise coordinate at the hub is 0, and the spanwise coordinate at the shroud is 1. The relative coordinates of the spanwise sections are $Sp_w$ (Spanwise) = 0.05, $Sp_w$ = 0.1, $Sp_w$ = 0.3, $Sp_w$ = 0.5, $Sp_w$ = 0.7, $Sp_w$ = 0.9. Along the direction of rotation of the impeller, six the circumferential direction (pressure surface to suction surface). In the same flow channel, the coordinate of the front of the blade is 0, the back of the blade is 1, and the relative coordinates of the six sections are $Cd$ (circumferential direction) = 0.05, $Cd$ = 0.2, $Cd$ = 0.4, $Cd$ = 0.6, $Cd$ = 0.8, $Cd$ = 0.95. The intersection of these sections is the positions of the pressure fluctuation monitoring points. The monitoring points are named according to the flow direction position $St_w$, the span direction position $Sp_w$, and the circumferential direction $Cd$, as shown in Figure 7(b). For example, P661 indicates that the monitoring point is located at the sixth flow direction section ($St_w = 0.99$)
Figure 4. Schematic diagram of monitoring points layout: (a) streamwise and spanwise section position (b) circumferential direction and arrangement of monitoring points.

Figure 5. Experimental configuration: (a) experimental axial flow pump system (b) location of pressure fluctuation monitoring points.

and the sixth span direction section ($Sp_{w} = 0.9$), and the first $Cd$ section ($Cd = 0.05$); whilst P656 indicates that the monitoring point is located at the sixth flow direction section ($St_{w} = 0.99$), the fifth span direction section ($Sp_{w} = 0.7$), and the sixth circumferential direction section ($Cd = 0.95$).

3. Experimental validation

Figure 5 shows the experimental system of the axial flow pump. Under the condition of no cavitation, the energy characteristics and pressure fluctuation experiments were carried out on the test bench. The uncertainty of pressure transmitter (V15712-HD1A1D72D) is less than ±0.10%, the uncertainty of magnetic flowrate meter (LDG-500S) is less than ±0.20%, and the uncertainty of tacho-torquemeter (JCZL2-500) is less than ±0.10%, the uncertainty of pressure fluctuation signal sensor (HDP503) is less than ±0.25%. The diameter of the impeller $D$ is 300 mm, the diameter of the hub $D_h$ is 140 mm, the impeller blades number is 4, the guide vanes number is 7, the rated speed $n_r$ is 1450 rpm, the rated head $H_0$ is 7.12 m, the rated flow $Q_0$ is 335.38L/s, and the specific speed $n_s (n_r Q_0^{0.5}/H_0^{0.75})$ is about $193m^{0.75}/s^{1.5}$. The experiments were carried out on a test bench that complies with the IEC-60193 standard.

Figure 6 shows a comparison of the energy performance between the experiment and the simulation, and the locations of the pressure measurement selected by the simulation are the same as the experiment. All selected working conditions were calculated for at least 50 revolutions of the impeller. Data covering 10 revolutions of the impeller were selected for time-average analysis and energy performance results were obtained. From the highest point in the hump region ($0.62Q_0$) to the large flow rate, the maximum value of the relative error between the numerical simulation and experiment of the head and efficiency is less than 3.5%. As shown in Figure 7, in addition to the comparison of energy performance, the pressure fluctuation results of some key monitoring points P1 (right side of the middle part of the impeller) and P2 (bottom of the impeller outlet)
Figure 6. Energy performance curve.

Figure 7. Comparison of the PPV.

(Figure 5(b)) in the experiment were selected for quantitative comparison with the numerical simulation results. The peak-peak values (PPV) of pressure fluctuation of simulation is less different from the experiment, and the trend is the same. Comprehensive above, the results of the numerical simulation can reflect the result of the experiment relatively accurately.

4. Horn-like vortices in rotor corner separation flow

To illustrate the characteristics of corner separation under off-design conditions, this paper selects the $0.62Q_0$ (the highest point in the hump region), $0.80Q_0$, $0.90Q_0$, and the optimal condition ($1.0Q_0$) for subsequent analysis.

4.1. Dominant vortex structure in the impeller

Figure 8 shows the average streamline distribution of the impeller rotating 30 times under these four working conditions. In the $1.0Q_0$ condition, the streamlines flow out almost uniformly along the blade profile. In the $0.9Q_0$ condition, helical streamlines begin to appear near the hub, and interwoven flows become more intense as the flow rate decreases. This interwoven spiral flow starts from the exit hub corner area on the suction surface and extends to the pressure surface, with the mixing and shear between the fluids generated by this special flow often inducing large-scale vortex structures (Jarrin et al., 2006).

The results of vortex identification based on absolute swirling strength (Liutex) are shown in Figure 9. Liutex is an important parameter index for evaluating the structural characteristics of the vortex (Dong et al., 2019; Liu et al., 2018; Liu et al., 2019; Zhan et al., 2020), which is widely used in the identification of internal vortices of rotating machinery (Wang et al., 2020; Wang, Zeng, et al., 2021). Liutex exactly represents the rigid rotation part of local fluid motions and an explicit expression of the
Figure 9. Absolute swirling strength of the horn-like vortex under different working conditions ($Liutex = 1000 s^{-1}$): (a) $1.0Q_0$ (b) $0.9Q_0$ (c) $0.8Q_0$ (d) $0.62Q_0$.

The vector is given by Wang et al. (2019).

$$R = \left[ \omega \cdot r - \sqrt{(\omega \cdot r)^2 - 4\lambda_{ci}^2} \right] \cdot r \quad (9)$$

where $r$ represents the real eigenvector of velocity gradient tensor ($\nabla \mathbf{V}$) and $\lambda_{ci}$ is the imaginary part of the complex conjugate eigenvalues of velocity gradient tensor ($\nabla \mathbf{V}$). $Liutex$ represents the angular velocity of rigid rotation (absolute swirling strength).

At the same identification threshold ($Liutex = 1000 s^{-1}$), there is no obvious vortex structure in the impeller at the $1.0Q_0$ working condition, and as the flow rate decreases, large-scale vortex structures extending from the corner region of the suction surface to

Figure 10. Average turbulence eddy dissipation and absolute swirling strength along the vortex core line: (a) average turbulence eddy dissipation (b) average absolute swirling strength.
pressure surface at $0.9Q_0$, $0.8Q_0$, and $0.62Q_0$ working conditions. With the decrease of flow rate, the unstable flowfield in the corner region is obviously intensified, the separation characteristic is gradually strengthened, and the absolute swirling strength of the horn-like vortex continues to increase. The locations where these large-scale vortex structures appear accord with the helical streamlines shown in Figure 8. Since the shape of the large-scale vortex is similar to a horn, to better describe this vortex structure, we define it as a horn-like vortex. From the distribution of the vortex structure in the impeller, it can be seen that the horn-like vortices are the dominant vortex structure in rotor corner separation flow in axial flow pumps under off-design conditions.

Figure 10 shows the distribution of average turbulence eddy dissipation and the absolute swirling strength distribution along the vortex core line of the horn-like vortex. According to the magnitude of turbulent eddy dissipation and the absolute swirling strength, the horn-like vortex can be divided into three regions: the hub corner separation region, the main trunk region, and the vortex tail region. On the whole, there is a certain correlation between the average turbulence eddy dissipation and the absolute swirling strength, which gradually decreases along the vortex core line. Due to the strong

![Figure 11](image)

**Figure 11.** Vortical characteristics of the horn-like vortices and radial gradient distribution of the axial velocity under different working conditions: (a) $0.9Q_0$-Vortical characteristics (b) $0.9Q_0$-radial gradient of $V_m$ (c) $0.8Q_0$-Vortical characteristics (d) $0.8Q_0$-radial gradient of $V_m$ (e) $0.62Q_0$-Vortical characteristics (f) $0.62Q_0$-radial gradient of $V_m$. 
effect of unstable reflux in the corner region, the vortex structure in the corner region has high turbulence eddy dissipation and strong rotation effects. As the flow develops, the turbulence eddy dissipation and the absolute swirling strength of the horn-like vortex in the main trunk region gradually weaken and reach the minimum after reaching the vortex tail. In fact, the attenuation of the turbulence dissipation and absolute swirling strength along the vortex core line indirectly reflect the gradual recovery of fluid energy.

4.2. Volume change of the horn-like vortex tube

Figure 11 shows the vortical features of the horn-like vortex (the vortex core lines and the helical streamlines) and the contours of radial gradient of the axial velocity near the impeller outlet. As can be seen from Figure 11, with the decreases of flow rate, the radial gradient of the axial velocity is particularly strong near the position of the horn-like vortex, which changes greatly, while the tangential component of the fluid is quite constant (the rotor works at a constant shaft speed), leading to significant changes in the attack angle of the flow near the corner region. Therefore, in order to analyze the impact of flow field deterioration at the corner region on downstream flows, the variation of vortex tube volume of the horn-like vortex was used to make a judgment under the same identification threshold with the decreased of flowrate, shown in Figure 12. Figure 12 shows the change in the vortex tube volume ratio (the ratio of the volume of horn-like vortex tubes to the impeller volume) under the recognition threshold \( \text{Lius}_{\text{x}} = 1000 \text{s}^{-1} \). The vortex tube volume ratio satisfies the law of \( \zeta_v = -1.95\left(\frac{Q}{Q_0}\right) + 1.94 \) after fitting and the goodness of fit \( r^2 \) of this curve is 0.9542, which indicates that the regression line fits the observations well. The vortex tube volume ratio \( \zeta_v \) increases linearly as the flow rate decreases and reaches 0.8% under the 0.62\( Q_0 \) condition (approximately 3.82 times the volume ratio of the vortex tube under 0.9\( Q_0 \)), which may lead to a gradual increase in flow passage blockage. That is to say, with the decrease of the flow rate, a non-optimal span-wise distribution of the axial velocity generates excessive incidence in the corner region, the flow pattern then deteriorates rapidly. The deteriorating unstable circulation enhances the formation of horn-like vortices, resulting in a rapid increase in the vortex tube volume.

5. Periodic evolution and pressure fluctuation characteristics of the horn-like vortex

To ensure the safe and stable operation of the axial flow pumping station, its working condition is often not less than the highest point in the hump region (0.62\( Q_0 \)). To analyze the periodic evolution characteristics (the stretching and deformation process of the horn-like vortex tube) of the horn-like vortex and its influence on pressure fluctuation, considering the similarity of the evolution process of the horn-like vortex, the most unfavorable condition 0.62\( Q_0 \) is selected for subsequent analysis.

5.1. Periodic evolution characteristics of the horn-like vortex

As the impeller rotates, the unsteady flow in the corner region begins to develop downstream. The periodic evolution process of the horn-like vortex under the 0.62\( Q_0 \) working condition is shown in Figure 13. The basic evolution process of the horn-like vortex includes five stages: inception in the hub corner, gradual growth, developing from the blade suction surface to the pressure surface, attaching to the blade, decay accompanied by the inception of the next evolution in the hub corner (inception-growth-development-attachment-decay). The relative time of each stage of the evolution process of the horn-like vortex is given, where \( t \) is the time for the impeller to rotate about 52.3 revolutions (2.163 s), and \( t_0 \) is the time for the impeller to rotate one circle. The periodic evolution of the horn-like vortex is the stretching and deformation process of the vortex tube and the statistical average frequency of the periodic evolution is about 2.76 times the impeller rotation frequency.

5.2. Influence of the horn-like vortices on pressure fluctuation

Appendix 2 shows the four-period evolution of the horn-like vortex and the corresponding periodic changes of the
Figure 13. Periodic evolution process of the horn-like vortex (Liutex = 1000 s⁻¹): (a) inception (2.163 s + 0.35τ₀) (b) growth (2.163 s + 0.57τ₀) (c) development (2.163 s + 0.87τ₀) (d) attachment (2.163 s + 0.94τ₀) (e) growth and decay (2.163 s + 1.22τ₀) (f) development (2.163 s + 1.24τ₀).

pressure fluctuation. This section gives a more detailed analysis of one of the cycles. Figure 14 shows the distribution of the pressure fluctuation coefficient ($C_p$, Equation 6) in the corresponding flow channel during the evolution of the horn-like vortex. Static pressure $p$ usually tends to have a local minimum value in the vortex core because of ‘cyclostrophic balance’ (Jeong & Hussain, 1995). To analyze the change of pressure fluctuation induced by the periodic evolution of the horn-like vortex, some key monitoring points were selected. Figure 14 shows the distribution of the pressure fluctuation coefficient during the periodic evolution of the horn-like vortex. As can be seen in Figure 14(b), in the inception stage of the horn-like vortex, the $C_p$ of monitoring point P525 near the hub corner region begins to decrease. As the main vortex structure has not yet developed to
the remaining monitoring points, the $C_p$ at other monitoring points gradually increases. During the growth stage of the horn-like vortex (Figure 14(c)), as the vortex core passes through, the $C_p$ at each monitoring point (except the monitoring point near the corner) is gradually reduced, and there is a significant pressure drop at each monitoring point at this time (Figure 14(d)). With the development of the horn-like vortex (Figure 14(e)), monitoring points P522 and P632 near the tail of the horn-like vortex reach the lowest point of the pressure coefficient. As the vortex core reaches monitoring point P641, the $C_p$ of P641 begins to decrease at this time (Figure 14(f)). When the horn-like vortex is attached to the blade (Figure 14(g)), monitoring point P641 in the tail of the horn-like vortex reaches the lowest value of the local pressure coefficient at this time (Figure 14(h)). In the next cycle, the horn-like vortex begins to grow (Figure 14(i)), and the pressure coefficient $C_p$ shown in Figure 14(j) begins to decrease, which is the same trend as shown in Figure 14(b). To further clarify the characteristics of pressure fluctuation, the following statistical analysis was conducted for the 216 monitoring points.

Figure 15(a) shows the peak-to-peak value (PPV) distribution at all monitoring points. The abscissa indicates the position of the monitoring point in the circumferential direction. Taking the first flow direction section ($St_w = 0.01$) as an example, 1–6 means the first spanwise section ($Sp_w = 0.05$) monitoring points P111 $\sim$ P116, 7–12 means the second spanwise section ($Sp_w = 0.1$) monitoring points P121 $\sim$ P126, and others are based on this analogy. Figure 15(b) shows the average PPV of all monitoring points on each section, and the abscissa 1–6 corresponds to the position of each section.

As shown in the area enclosed by the red line in Figure 15(a), the maximum peak-to-peak value (PPV) occurs at the P626 ($St_w = 0.99, Sp_w = 0.1, C_d = 0.95$) monitoring point near the corner area, and its maximum value is about 2.10 times the head of this working condition. The larger PPV appears near the sixth streamwise section and the second spanwise section ($St_w = 0.99, Sp_w = 0.1$), which corresponds to the area where the horn-like vortex grows. In addition, the average PPV of each section given in Figure 15(b) shows that along the streamwise section, the average PPV generally shows an increasing trend, and the maximum average PPV appears at the impeller outlet section ($St_w = 0.99$). The average PPV on the impeller outlet section close to the horn-like vortex is about 2.79 times that of the impeller inlet section ($St_w = 0.01$). From the perspective of the spanwise section and the circumferential direction, the closer the corner area is, the higher the PPV.

Figure 14. Pressure fluctuation coefficient changes during the periodic evolution of the horn-like vortex: (a) inception (b) pressure fluctuations in inception stage; (c) growth (d) pressure fluctuations in growth stage (e) development (f) pressure fluctuations in development stage (g) attachment (h) pressure fluctuations in attachment stage (i) growth (j) pressure fluctuations in growth stage.

Obviously, the above results reflect the rapid increase in pressure fluctuation caused by the horn-like vortex in the rotor corner separation flow. This kind of unstable pressure fluctuation may aggravate the instability of the axial flow pump and endanger the normal operation of the system.
6. Conclusions and prospect

In this paper, the key vortical characteristics of the dominant vortex structure (horn-like vortex) near the hub in an axial flow pump and the relationship between its periodic evolution and pressure fluctuation have been carried out. The main conclusions are as follows:

With the decrease of the flow rate, the average vortex intensity and turbulence eddy dissipation show a gradual weakening trend along the vortex core line from the corner area to the vortex tail under various working conditions. The volume ratio of the horn-like vortex tube gradually increases with the decrease of the flow rate ($\zeta_v = -1.95(Q/Q_0) + 1.94$) and the volume ratio of the vortex tube under the 0.62$Q_0$ working condition is about 3.82 times that of the 0.9$Q_0$ working condition.

The analysis of the horn-like vortex in the typical working condition 0.62$Q_0$ found that it has obvious periodicity and its evolution in a life cycle includes five stages, inception-growth-development-attachment-decay. The statistical average frequency of the periodic evolution is about 2.76 times the shaft rotation frequency.

The pressure fluctuation near the hub induced by the horn-like vortex increases rapidly and dominates in the flow channel. The average peak-peak value of pressure fluctuation on the impeller outlet section close to the horn-like vortex is about 2.79 times that of the impeller inlet section. The horn-like vortex has an important influence on the flow near the hub of the axial flow pump under the small flow condition, and should be paid enough attention to.

This paper still has some limitations. It only analyzes the evolution of the horn-like vortex and the corresponding pressure fluctuation in a single flow channel under the 0.62$Q_0$ working condition. However, due to the existence of circumferential disturbance, the flow capacity of each flow channel is not the same. This also leads to differences in the flow development of each channel and the corner separation in adjacent channels is not synchronized, and the vortex structure induced by the corner separation does not occur and die simultaneously. Due to the non-synchronization of the development of vortex structure in different channels, the vibration of the unit may be intensified. Therefore, more in-depth research is needed on the issue of 'vortex evolution in different flow channels-pressure fluctuation-unit vibration'.

Disclosure statement

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References

Abadi, A., Sadi, M., Farzaneh-Gord, M., Ahmadi, M. H., & Chau, K. W. (2020). A numerical and experimental study on the energy efficiency of a regenerative heat and mass exchanger utilizing the counter-flow maisotsenko cycle. Engineering Applications of Computational Fluid Mechanics, 14(1), 1–12. https://doi.org/10.1080/19942060.2019.1617193

Adams, T., Grant, C., & Watson, H. (2012). A simple algorithm to relate measured surface roughness to equivalent sand-grain roughness. International Journal of Mechanical Engineering and Mechatronics, 1(2), 66–71. https://doi.org/10.11159/ijmem.2012.008
Ashton, R., Viola, E., Camarri, S., Galler, E., & Lungo, G. V. (2016). Hub vortex instability within wind turbine wakes: Effects of wind turbulence, loading conditions, and blade aerodynamics. Physical Review Fluids, 1(7). https://doi.org/10.1103/PhysRevFluids.1.073603

Celik, I. B., Ghia, U., Roache, P. J., & Ferzlas, C. J. (2008). Procedure for estimation and reporting of uncertainty due to discretization in CFD applications. Journal of Fluids Engineering, 130(7), 078001. https://doi.org/10.1115/1.2960953

Chaluvadi, V. S. P., Kalfas, A. I., & Hodson, H. P. (2004). Vortex transport and blade interactions in high pressure turbines. Journal of Turbomachinery, 126(3), 395–405. https://doi.org/10.1115/1.1773849

Denton, J. D. (1993). Loss mechanisms in turbomachines. Journal of Turbomachinery, 115(4), 621–656. https://doi.org/10.1115/93-GT-435

Dong, X. R., Gao, Y. S., & Liu, C. Y. (2019). New normalized vortex/vortex identification method. Physics of Fluids, 31(1), 011701. https://doi.org/10.1063/1.5066016

Dring, R. P., Joslyn, H. D., & Hardin, L. W. (1982). An investigation of axial compressor rotor aerodynamics. Journal for Engineering for Power, 104(1), 84–96. https://doi.org/10.1115/1.3227270

Ghalandari, M., Koohshahi, E. M., Mohamadian, F., Shamshirband, S., & Chau, K. W. (2019). Numerical simulation of nanofluid flow inside a root canal. Engineering Applications of Computational Fluid Mechanics, 13(1), 254–264. https://doi.org/10.1080/19942060.2019.1578696

Hah, C., & Loellbach, J. (1999). Development of hub corner stall and its influence on the performance of axial compressor blade rows. Journal of Turbomachinery, 121(1), 67–77. https://doi.org/10.1115/1.2841235

Han, X. S., & Krajnović, S. (2013). An efficient very large eddy simulation model for simulation of turbulent flow. International Journal for Numerical Methods in Fluids, 71(11), 1341–1360. https://doi.org/10.1002/fld.3714

Han, X. S., & Krajnović, S. (2015). Very-large-eddy simulation based on k-ω model. AIAA Journal, 53(4), 1103–1108. https://doi.org/10.2514/1.J053341

Jarrin, N., Benhamadouche, S., Laurence, D., & Prosser, R. (2006). A synthetic-eddy-method for generating inflow conditions for large-eddy simulations. International Journal of Heat & Fluid Flow, 27(4), 585–593. https://doi.org/10.1016/j.ijheatfluidflow.2006.02.006

Jeong, J., & Hussain, F. (1995). On the identification of a vortex. Journal of Fluid Mechanism, 285(1), 69–94.

Jin, Y., Cheng, Z. Y., Han, X. S., Mao, J. K., & Jin, F. (2019). VLES of drag reduction for high Reynolds number flow past a square cylinder based on OpenFOAM. Ocean Engineering, 190, Article 106450. https://doi.org/10.1016/j.oceaneng.2019.106450

Kan, K., Zheng, Y., Chen, Y. J., Xie, Z. S., Yang, G., & Yang, C. X. (2018). Numerical study on the internal flow characteristics of an axial-flow pump under stall conditions. Journal of Mechanical Science & Technology, 32(10), 4683–4695. https://doi.org/10.1007/s12206-018-0916-z

Liu, C. Q., Gao, Y. S., Dong, X. R., Wang, Y. Q., Liu, J. M., Cai, X. S., & Gui, N. (2019). Third generation of vortex identification methods: Omega and Liutex/rortex based systems. Journal of Hydrodynamics, 31(2), 205–223. https://doi.org/10.1007/s42241-019-0022-4

Liu, C. Q., Gao, Y. S., Tian, S. L., & Dong, X. R. (2018). Rortex – a new vortex vector definition and vorticity tensor and vector decompositions. Physics of Fluids, 30(3), 035103. https://doi.org/10.1063/1.5023001

Ochi, F., Fujisawa, T., Ohmori, T., & Kawamura, T. (2009). Simulation of propeller hub vortex flow. First International Symposium on Marine Propulsors.

Shen, X., Zhang, D. S., Xu, B., Shi, W. D., & Bart, V. E. (2021). Experimental and numerical investigation on the effect of tip leakage vortex induced cavitation flow on pressure fluctuation in an axial flow pump. Renewable Energy, 163, 1195–1209. https://doi.org/10.1016/j.renene.2020.09.004

Wang, C. Y., Wang, F. J., Tang, Y., Wang, B. H., Yao, Z. F., & Xiao, R. F. (2020). Investigation on the horn-like vortices in stator corner separation flow in an axial flow pump. Journal of Fluids Engineering, 142(7), 071208. https://doi.org/10.1115/1.4046376

Wang, C. Y., Wang, F. J., Xie, L. H., Wang, B. H., Yao, Z. F., & Xiao, R. F. (2021). On the vortical characteristics of horn-like vortices in stator corner separation flow in an axial flow pump. Journal of Fluids Engineering, 143(6), 061201. https://doi.org/10.1115/1.4049687

Wang, C. Y., Zeng, Y. S., Yao, Z. F., & Wang, F. J. (2021). Rigid vorticity transport equation and its application to vortical structure evolution analysis in hydro-energy machinery. Engineering Applications of Computational Fluid Mechanics, 15(1), 1016–1033. https://doi.org/10.1080/19942060.2021.1938685

Wang, Y. Q., Gao, Y. S., Liu, J. M., & Liu, C. Q. (2019). Explicit formula for the Liutex vector and physical meaning of vorticity based on the Liutex-shear decomposition. Journal of Hydrodynamics, 31(3), 464–474. https://doi.org/10.1007/s42241-019-0032-2

Wu, X. F., Lu, Y. D., Tan, M. G., & Liu, H. L. (2018). Effect of vane angle on axial flow pump running characteristics in saddle zone. Transactions of the Chinese Society of Agricultural Engineering, 34(17), 46–53. https://doi.org/10.11975/j.issn.1002-6819.2018.17.007

Xia, Z. Y., Han, X. S., & Mao, J. K. (2019). Assessment and validation of very-large-eddy simulation turbulence modeling for strongly swirling turbulent flow. AIAA Journal, 1(1), 1–16. https://doi.org/10.2514/1.J058302

Yan, H., Liu, Y. W., Li, Q. S., & Lu, L. P. (2018). Turbulence characteristics in corner separation in a highly loaded linear compressor cascade. Aerospace Science & Technology, 75, 139–154. https://doi.org/10.1016/j.ast.2018.01.015

Yang, F., & Liu, C. (2014). Pressure pulsations of the blade region in s-shaped shaft-extension tubular pumping system. Mathematical Problems in Engineering, Article 820135. https://doi.org/10.1155/2014/820135

Zhan, J. M., Chen, Z. Y., Li, C. W., Hu, W. Q., & Li, Y. T. (2020). Vortex identification and evolution of a jet in cross flow based on vortex. Engineering Applications of Computational Fluid Mechanics, 14(1), 1237–1250. https://doi.org/10.1080/19942060.2020.1816496

Zierke, W. C., & Straka, W. A. (1971). Flow visualization and the three-dimensional flow in an axial-flow pump. Journal of Propulsion and Power, 12(2), 250–259. https://doi.org/10.1115/1.3227270
Appendices

Appendix 1. Selection of Liutex threshold and its influence on the results

This article makes a simple comparison on the selection of Liutex threshold. A larger threshold will result in the inability to fully display the shape of the horn-like vortex, and a smaller threshold will cause many complex vortices to appear in the flow channel, making it impossible to visually display the horn-like vortex. The vortex structure distribution in the impeller with threshold Liutex = 1200, 1000, 800, 600 s\(^{-1}\) are given, as shown in Figure A1. Through simple comparison, Liutex = 1200 s\(^{-1}\) cannot fully display the shape of the horn-like vortex under 0.9\(Q_0\) working condition. In the 0.62\(Q_0\) working condition, the vortices in the flow channel identified by Liutex = 600 s\(^{-1}\) is too complicated to visually display the horn-like vortex. Taking into account the unification of the identification of various working conditions and subsequent analysis, the threshold value of Liutex = 1000s\(^{-1}\) was selected after comprehensive comparison. Figure A2 shows the change of the volume ratio of the vortex tube under the above four thresholds with working conditions. Generally speaking, the change of the threshold will affect the volume ratio of the vortex tube, but the change trend is the same.

Figure A1. Horn-like vortex structure under different Liutex thresholds under different working conditions: (a) 0.9\(Q_0\)-Liutex = 600s\(^{-1}\) (b) 0.9\(Q_0\)-Liutex = 800s\(^{-1}\) (c) 0.9\(Q_0\)-Liutex = 1000s\(^{-1}\) (d) 0.9\(Q_0\)-Liutex = 1200s\(^{-1}\) (e) 0.8\(Q_0\)-Liutex = 600s\(^{-1}\) (f) 0.8\(Q_0\)-Liutex = 800s\(^{-1}\) (g) 0.8\(Q_0\)-Liutex = 1000s\(^{-1}\) (h) 0.8\(Q_0\)-Liutex = 1200s\(^{-1}\) (i) 0.62\(Q_0\)-Liutex = 600s\(^{-1}\) (j) 0.62\(Q_0\)-Liutex = 800s\(^{-1}\) (k) 0.62\(Q_0\)-Liutex = 1000s\(^{-1}\) (l) 0.62\(Q_0\)-Liutex = 1200s\(^{-1}\).

Figure A2. Vortex tube volume ratio under different Liutex thresholds.
Appendix 2. Multiple evolution periods of the horn-like vortex and the change of pressure fluctuation

This part supplements the change in pressure fluctuation of the horn-like vortex during the evolution of multiple cycles, as shown in Figure A3. Corresponding analysis is carried out on the two key monitoring points P524 and P523. As shown in Figure A4, in the inception stage, the pressure fluctuation coefficient $C_p$ at the two monitoring points began to rise gradually. As the flow evolves, from growth to development to attachment stage, the $C_p$ at the monitoring point P524 shows a downward trend and then an upward trend. The $C_p$ at the monitoring point P523 shows an overall downward trend. The pressure changes in the four cycles are generally the same, showing obvious periodicity.

Figure A3. The multi-period evolution process of the horn-like vortex and the change of pressure fluctuation: (a) 1T-Inception (2.163s) (b) 1T-Growth (2.163 s + 0.11$t_a$) (c) 1T-Development (2.163 s + 0.13$t_a$) (d) 1T-Attachment (2.163 s + 0.21$t_a$) (e) 2T-Growth (2.163 s + 0.57$t_a$) (f) 2T-Development (2.163 s + 0.87$t_a$) (g) 2T-Attachment (2.163 s + 0.94$t_a$) (h) 3T-Growth (2.163 s + 1.22$t_a$) (i) 3T-Development (2.163 s + 1.24$t_a$) (j) 3T-Attachment (2.163 s + 1.29$t_a$) (k) 4T-Growth (2.163 s + 1.57$t_a$).

Figure A4. Periodic changes of pressure fluctuation in the evolution of the horn-like vortex.