Influence of End Wall Clearance on Guide Vane Self-Excited Vibrations at Small Openings during Pump Mode’s Starting Up Process of a Reversible Pump Turbine

Quanwei Liang 1, Wenzhe Kang 2,*, Lingjiu Zhou 2,3 and Zhengwei Wang 4

1 Research & Test Center, Dongfang Electric Machinery Co., Ltd., Deyang 618000, China; liangquanwei@dongfang.com
2 College of Water Resources and Civil Engineering, China Agricultural University, Beijing 100083, China; zlj@cau.edu.cn
3 Beijing Engineering Research Center of Safety and Energy Saving Technology for Water Supply Network System, Beijing 100083, China
4 Department of Thermal Engineering, Tsinghua University, Beijing 100084, China; wzw@mail.tsinghua.edu.cn
* Correspondence: kwz@cau.edu.cn

Abstract: Unstable guide vane torsional mode self-excited vibrations that occur at small guide vane openings during the transient operations with pump flow, such as the starting and closing of the pump mode, are considered to have potentially severe consequences, such as guide vane slippage or damage to the link and lever mechanism. Related site tests have indicated that the end wall clearance of a guide vane may have important influences on torsional mode self-excited vibrations. In this paper, numerical investigations, which were based on computational fluid dynamics (CFD) with a single degree of freedom (1DOF) mass-spring oscillator, were carried out on a prototype high-head reversible pump turbine. The results showed that the guide vane self-excited vibrations are unstable under steady-state conditions and during the pump mode’s starting up process for cases with small end wall clearances. In addition, the critical conditions of self-excitation instability under steady-state conditions have larger safety margins than those during the pump mode’s starting up process. After further discussion, it was concluded that increasing the end wall clearance to suppress unstable guide vane self-excited vibration is unreliable due to the complexity and randomness of the initial vibration excitations.

Keywords: reversible pump turbine; pump mode’s starting up; guide vane self-excited vibration; end wall clearance; coupling simulation

1. Introduction

Pumps are prevalent in many different applications in the agricultural, industrial, and other sectors, and there are several different types, including positive displacement pumps, centrifugal pumps, axial pumps, etc. [1–4]. In modern electrical power networks, pumped storage hydropower technology is used to improve grid stability and to balance load variation [5,6]. As key components in pumped storage power stations, reversible pump turbines are required to withstand many transient operations, such as starting up, shutting down, power failure, load rejection and emergency shutdown [7]. These transient operations occur frequently due to the demands of the power grid and may lead to hydraulic instability and unacceptable mechanical vibrations [8–10]. Guide vane vibration is one of the common type of mechanical vibration that occur during the operations of a prototype pump turbine, which have potentially severe consequences, such as guide vane slippage and the damage to the link and lever mechanism [9,11–13]. Rotor-stator interaction, flow stall and von Karman vortex shedding are often found to be the main sources of guide vane vibration in normal steady-state operating ranges.
One of the typical problems of guide vane vibration during the transient processes of a prototype pump turbine is the experienced vibration during the commissioning tests of pump-starting, pump-closing, and turbine mode over-speed tests in the No.1 unit of Yixing pumped storage power stations (Yixing, Jiangsu, China) [9,13]. In these tests, pressure fluctuations in the radial gap between the guide vanes and the runner, and the guide vane dynamic torque, which was approximated from servo pressures, were found to have large amplitudes around the torsional mode natural frequency [12]. These vibrations that occur during the different operating modes are considered to be related to two phenomena in the pump flow direction at small openings: bi-stable flow that is caused by the attaching flow on the trailing edge of the guide vane, and torsional mode self-excited vibrations [12]. For a prototype high-head reversible pump turbine, high pressure level is needed in the radial gap area between the guide vanes and runner before opening the guide vanes, in order to deliver an efficient starting up process. According to the study in [14], the attaching flow on the trailing edge of a guide vane is barely induced in these cases due to the relatively high flow rate. Instead, the torsional mode self-excited vibrations pose a higher risk, especially at small openings. Considering the frequent transient operations and small openings, the guide vane self-excited vibrations may also lead to the structural damage of hardware and even premature mechanical failures in extreme cases.

The problems of guide vane self-excited vibrations have been studied through theoretical analyses and model experiments [11]. Due to the complexity of self-excited vibrations, some simplified assumptions and setups have been carried out in those theoretical models and experimental tests, which has led to some limitations on the considerations of the geometric shape or circumferential settings of guide vanes. In contrast, numerical simulations of fluid-structure coupling have been extensively applied on many fluid-induced vibration problems [15,16]. As studied in [17,18], the fluid-structure interaction of an oscillating hydrofoil can be modeled by a single degree of freedom (1DOF) oscillator and verified by the agreement between the predicted and experimental hydrodynamic torque. In the site tests and numerical studies in [12–14], it was concluded that, the only involved guide vane vibration at the small openings in prototype high-head reversible pump turbines, is the torsional mode self-excited vibration. Thus, the angular displacement of a guide vane, which is caused by the elastic deformation of the guide vane stem, can be also approximately represented using a 1DOF mass-spring oscillator. Similar approaches, such as coupling unsteady computational fluid dynamics (CFD) with a 1DOF oscillator, were used for the guide vane vibration problems of the No.1 unit of the Yixing pump turbine in [12,13]. The torsional vibrations were predicted in the corresponding guide vane openings, with response frequencies around the torsional mode natural frequency in water, as observed at site tests [12].

Guide vane torsional vibrations and high-intensity noises were also experienced when the guide vane opening is lower than 1° in recent tests on the starting up process of a prototype high-head reversible pump turbine. The pressure fluctuations in the radial gap area between the guide vanes and runner outlet clearly demonstrated high amplitudes around the torsional mode natural frequency of the guide vanes. The site tests also showed that the torsional vibrations and pressure fluctuations in the radial gap were weakened for the same prototype pump turbine when larger guide vane end wall clearances were used. This indicates that the end wall clearance may have important influences on the torsional mode self-excited vibrations of guide vanes. However, less research has been carried out on this subject. Thus, in this study, numerical investigations were performed on a prototype high-head reversible pump turbine, aiming to help us to understand how the end wall clearance influences guide vane torsional mode self-excited vibrations. This study could also help us to understand the feasibility and reliability of suppressing torsional mode self-excited vibrations by changing the end wall clearances. Firstly, three-dimensional steady-state RANS calculations were carried out to study the influence of the leakage flow through guide vane end wall clearance. Then, coupling simulations that were based on the unsteady computational fluid dynamics method and a 1DOF mass-spring oscillator
were performed at different steady-state operating conditions with different small openings and end wall clearances. Considering the influence of transient flows and guide vane opening motions, coupling simulations were also carried out for the starting up process with different end wall clearances. In the end, analysis and further discussions were carried out to establish the feasibility and reliability of suppressing torsional mode self-excited vibrations by changing the end wall clearance during the pump mode’s starting up process.

2. Numerical Methods

2.1. Governing Equations

The turbulent flow calculations were carried out using the Reynolds-Averaged Navier-Stokes (RANS) solver in ANSYS CFX. The mass conservation equation and momentum equation were represented by Equations (1) and (2), respectively, where \( \frac{\partial}{\partial t} (\rho u_i) \) is the velocity, \( x_i (i = x, y, z) \) is the position, \( t \) is the time, \( p \) is the pressure, \( \mu \) is the dynamic viscosity and \( S_M \) is the external momentum source term. The Reynolds stresses \( \rho u_i u_j \) could be modified using a turbulence model. All averaged bars were removed except for the Reynolds stresses [19]. Due to the advantages of low costs and high accuracy in predicting flow separation, the turbulence model of \( k-\omega \) shear stress transport (SST) was utilized in this study.

\[
\frac{\partial}{\partial t} (\rho u_i) = 0 \quad (1)
\]

\[
\frac{\partial}{\partial t} (\rho u_i u_j) = -\frac{\partial}{\partial x_j} \left( \rho u_i \right) + \frac{\partial}{\partial x_i} \left( \mu \frac{\partial u_i}{\partial t} - \rho \delta_{ij} u_i u_j \right) + S_M \quad (2)
\]

The torsion could be represented by a 1DOF mass-spring oscillator [17,18]. The differential equation for this model was represented by Equation (3), where \( \theta \) is the rotation degree of freedom, \( M \) is the hydraulic torque, \( J \) is the mass moment inertia and \( \omega_1 \) is the natural circular frequency, which was calculated using Equation (4), where \( K \) is the torsion stiffness. The damping which is proportional to the vibration velocity of 1DOF oscillator was not considered in this study and neither was the additional external damping, e.g., from mechanical friction, because they would always be positive and stabilize the self-excited vibrations. Hydrodynamic damping was included in the hydraulic torque \( M \).

\[
\dot{\theta} + \omega_1^2 \theta = \frac{M}{J} \quad (3)
\]

\[
\omega_1 = \frac{K}{J} \quad (4)
\]

The rigid body solver in ANSYS CFX was applied to simulate the fluid-structure interaction, which avoided the structure mechanics calculation and the coupling between fluid and structure solver [20]. Based on the hydraulic torque and the displacement of the guide vane profile, the new angular position could be recalculated using the mesh deformation algorithm. In this way, the flow-induced torsional vibrations could be simulated for the guide vanes. A detailed description of the rigid body solver can be found in [20].

2.2. Overview of the Studied Object and Numerical Settings

A structure that is similar to a water turbine is commonly used for a reversible pump turbine, which generally consists of a draft tube, a runner, adjustable guide vanes, stay vanes and a spiral casing, as shown in Figures 1 and 2. In the most common regulations, the reversible pump turbine rotates as a pump for water storage when power becomes affluent. During the pump mode’s starting up process, the small guide vane opening situations with pump flow commonly occur and may be accompanied by unstable torsional mode self-excited vibrations in some prototype high-head pump turbines.

As shown in previous studies, the torsional mode self-excited vibrations are closely related to a high-speed flow through a narrow gap between two guide vane blades. In
this paper, the flow rates through the narrow gap between two guide vane blades were first calculated using 3D steady-state RANS simulations. Then, the influence of end wall clearance on the self-excited vibrations was analyzed using 2D coupling simulations. This indirect simulation approach could be not only used to analyze the influence of end wall clearance, but also to avoid the high simulation costs and the difficulty of mesh deformation near the clearance.

Figure 1. The main flow passage components of a pump turbine (pump mode).

Figure 2. Flow passage model of a pump turbine (pump mode).

Considering the leakage flow through the end wall clearance of the guide vane, three-dimensional steady-state RANS calculations were carried out under different guide vane opening conditions and end wall clearances. In this study, a half-height periodic flow passage was selected as the computational domain, which contained a unilateral end wall clearance \( (c_e) \) and two semi-guide vane blades, as shown in Figure 3. The inlet boundary was between the runner and the guide vane, while the outlet boundary was between the guide vane and the stay vane. The radial and circumferential velocities were set as the
inlet boundary conditions, which were converted by the characteristic curves of the pump turbine. The static pressure was set as the outlet boundary condition. These simplifications were considered tolerable in order to obtain the flow rates through the narrow gap between two guide vane blades.

Figure 3. The computational domain of the guide vane for the 3D steady-state RANS calculations (Left), where $c_e$ is the unilateral end wall clearance width (Right).

As shown in previous studies, the only involved vibrations for the guide vane are torsional mode self-excited vibrations [12–14], which are only related to the angular displacement that is caused by the elastic deformation of the guide vane stem. Thus, the 2D approach, which is based on CFD and a 1DOF mass-spring oscillator, was considered to be a valid approximation for simulating the torsional mode self-excited vibrations. In addition, comparative studies have shown that the influence of neighboring guide vanes on self-excited vibrations can be neglected. Thus, the computational domain was a partial flow passage that contained one guide vane (shown in Figure 4 with red lines) to produce a relatively fine mesh resolution and a high simulating accuracy. The downstream of the computational domain was extended outward for a better boundary condition. The pressure difference between the inlet and outlet was the cause of the high-speed flow through the narrow gap between the two guide vane blades. The very short vibration duration of the self-excited vibrations left little time to change the average flow rate. Additionally, the outlet pressure could be considered as constant during the self-excited vibration process due to the low frequency of the flow stall in the stay vanes. Thus, the total pressure and static pressure were set as boundary conditions in this study.

Figure 4. The computational domain of the guide vane for the 2D URANS-1DOF coupling simulations.
The near-wall flow was solved by the automatic near-wall treatment in ANSYS CFX. It automatically switched from wall functions to a low-Reynolds near-wall formulation as the mesh was refined. The guide vane profile was calculated using the rigid body solver, in which the 1DOF mass-spring oscillator could be established with the rotational degree of freedom, mass moment of inertia and torsional stiffness. The time-step was applied using \( \Delta t = 1.875 \times 10^{-4} \) s, considering the natural frequency that was imposed on the mass-spring oscillator. Then, the coupling calculating solution was set up for the guide vane self-excited vibrations, based on the unsteady Reynolds-averaged Navier-Stokes solver and the 1DOF oscillator model (URANS-1DOF). The main numerical settings are listed in Table 1. A detailed description of the numerical methodology can be found in [20].

Table 1. The basic numerical settings for the URANS-1DOF coupling simulation.

| Item                      | Setting                                      |
|---------------------------|----------------------------------------------|
| Mesh motion of the guide vane | Rigid body solution                          |
| External torque           | 1DOF torsional spring solution               |
| Wall function             | Automatic near-wall treatment                |
| Advection scheme          | High resolution                              |
| Turbulence numeric        | High resolution                              |
| Transient scheme          | Second-order backward Euler                  |
| Convergence criterion     | \( 10^{-5} \)                                |
| Maximum number of inner loops | 20                                           |

2.3. Mesh Generation and Independence

Hexahedral meshes were generated for both the 3D and 2D computational domains. The mesh resolutions were evaluated using the grid convergence index (GCI) method, which was based on the Richardson extrapolation [21]. The mesh resolution for the operating conditions of \( \alpha_0 = 0.2^\circ \) and \( c^* = 0.13\% \) was assumed reliable enough to merit extension to other conditions, where \( \alpha_0 \) is the average guide vane opening and \( c^* \) is the relative width of the unilateral end wall clearance. The leakage flow rate coefficient of the unilateral end wall clearance (\( C_{\phi c} \)) and the damping ratio of the guide vane opening (\( \zeta_{\alpha} \)) were used as the key parameters (\( \chi \)) of the 3D steady-state RANS results and the 2D URANS-1DOF coupling simulations, respectively. The damping ratio was defined by Equation (5), where \( \alpha \) is the guide vane opening, subscript P and R represent two peaks with an interval of \( NT (N = 1, 2, \ldots) \) and \( T \) represents the vibration period, as shown in Figure 5. The positive damping ratio implied that the guide vane self-excited vibration was unstable.

\[
\zeta_{\alpha} = \frac{\ln(\alpha_P/\alpha_R)}{2\pi N}
\]  

(5)

Figure 5. The definition of the damping ratio of the self-excited vibrations.
The calculations of the discretization error of the mesh are shown in Table 2. The convergence index of \( GCI_{fine}^{21} \) was within 5%. Thus, the mesh numbers 513,821 and 81,201 were considered to be tolerable for the 3D and 2D calculations in this work, as shown in Figure 6.

| Item       | \( C_{\phi c} \) (3D Steady-State RANS) | \( \xi_{\alpha} \) (2D URANS-1DOF) |
|------------|--------------------------------------|----------------------------------|
| \( N_1, N_2, N_3 \) | 513,821, 210,822, 80,923             | 81,201, 47,025, 26,532            |
| \( r_{21} \) | 1.346                                | 1.314                            |
| \( r_{32} \) | 1.376                                | 1.331                            |
| \( \chi_1 \) | 0.1705                               | -0.0290                          |
| \( \chi_2 \) | 0.1670                               | -0.0304                          |
| \( \chi_3 \) | 0.1795                               | -0.0339                          |
| \( \chi_{21}^{ext} \) | 0.1720                               | -0.0279                          |
| \( e_{21}^{ext} \) | 0.87%                                | 4.04%                            |
| \( GCI_{fine}^{21} \) | 1.1%                                 | 4.86%                            |

Figure 6. The partial cells of the computational domains: (a) the 3D steady-state RANS calculation; (b) the 2D URANS-1DOF coupling simulation.

3. Results and Discussion

3.1. Influence of End Wall Clearance on the Boundary Conditions of URANS-1DOF Coupling Simulations

Figure 7 shows the flow rate coefficients at different guide vane openings during the pump mode’s starting up process (converted by the characteristic curves of the pump turbine), which were applied as velocity inlet boundary conditions. The three-dimensional steady-state RANS calculations were carried out under different guide vane opening conditions (\( \alpha_0 = 0.05^\circ, 0.2^\circ, 0.4^\circ, 0.6^\circ \) and \( 1^\circ \)) and end wall clearance widths (\( c^* = 0.06\%, 0.13\%, 0.21\%, 0.3\%, 0.39\% \) and 0.48\%) in order to obtain the flow rates through the narrow gap between two guide vane blades. Figure 8 shows the pressure and velocity at the throat of the narrow gap as a function of height, in which \( z^* = 0 \) represents the horizontal central plane of the guide vane flow passage. Due to the leakage flow of the end wall clearance, there were small differences in the velocity fields for different heights, as shown in Figures 8 and 9. However, these were identified as axial disturbances, which were not considered in this study. The flow through the narrow gap is attached to the surface of the neighboring guide vane at different heights, which is similar to the 2D simulation results in Section 3.2.
Figure 10 shows the flow rate coefficients of the flow through the narrow gap between two guide vane blades at different operating conditions and demonstrates that the influence of end wall clearance on the flow rate cannot be ignored, especially at small openings with large end wall clearance widths.
The flow rate coefficients of the flow through the narrow gap between two guide vane blades vs. the guide vane opening and end wall clearance width.

The two-dimensional steady-state RANS calculations were also carried out for the computational domain shown in Figure 4 using the flow rate coefficients from Figure 7. The curves of the pressure difference between the inlet and outlet vs. the guide vane opening are drawn in Figure 11, which were employed as the boundary conditions of the URAN-1DOF coupling simulations. For the conditions of $c_e^* \leq 0.13\%$, the pressure difference reached its maximum value at $\alpha_0 = 0.3^\circ$ and then gradually decreased as the opening angle increased. For the others, the pressure difference gradually increased as the opening value increased. In addition, the pressure difference at $\alpha_0 = 1^\circ$ was less affected by the end wall clearance compared to other opening conditions.

The URANS-1DOF coupling simulations were carried out under the above operating conditions. Considering that severe hard excitation with a large initial amplitude would not likely occur in real pump turbine starting up processes, only the soft excitation was simulated in this study by subtracting the static value away from the total hydraulic torque in Equation (3).

The time histories of the guide vane openings and total hydraulic torque are drawn in Figure 12 to demonstrate the unstable case of $\alpha_0 = 0.2^\circ$ and $c_e^* = 0.06\%$. The positive value of the hydraulic torque signified the direction in which the guide vane was closing. The static pressure and velocity distributions near the narrow gap between two guide vane blades are also shown in Figure 13 at the moments corresponding to the red dots. The flow through the narrow gap was attached to the surface of the neighboring guide vane and led to large gradients of pressure and velocity. This flow pattern was aggravated by
smaller openings, which induced larger positive values of hydraulic torque. As the opening increased, the hydraulic torque became negative, which was in the opening direction of the guide vane. The high sensitivity of the pressure and velocity fields to the guide vane torsional vibrations led to the high-amplitude oscillation of the hydraulic torque. On the other hand, the fluctuation of hydraulic torque could, in turn, strengthen the torsional vibrations due to the small phase delay of the guide vane opening to hydraulic torque. This positive feedback caused small disturbances to eventually develop into unstable vibrations.

The damping ratios of the self-excited vibrations under different guide vane openings are drawn in Figure 14 as a function of end wall clearance. The self-excited frequencies were close to the torsional mode natural frequency of the guide vane, which was also consistent with the main vibration frequencies in the site tests. The damping ratios were negative under most operating conditions, as shown in Figure 14, which suggests that the self-excited vibrations in these cases were unstable and posed a high risk of causing structural collisions between the guide vane blades. The instability of the self-excited vibrations was gradually weakened with the increase in end wall clearance, especially under the condition of $\alpha_0 = 0.05^\circ$. This means that the influence of the end wall clearance on self-excited vibrations was significant at small openings.

![Figure 12](image_url)

**Figure 12.** The time histories of the guide vane opening and total hydraulic torque in an unstable case ($\alpha_0 = 0.2^\circ$ and $c^*_{e} = 0.06\%$).

![Figure 13](image_url)

**Figure 13.** Cont.
Figure 13. The static pressure and velocity magnitude distributions near the narrow gap between two guide vane blades during one vibration cycle: (a) static pressure; (b) velocity magnitude.

Figure 14. The damping ratios of self-excited vibrations under different guide vane openings as a function of end wall clearance.

3.3. Guide Vane Self-Excited Vibrations During the Pump Mode’s Starting Up Process

As experienced in site tests, guide vane self-excited vibrations occurred during the transient processes of the pump mode’s starting up with small guide vane openings. Therefore, in this section, the unsteady coupling calculations of URANS-1DOF were carried out in small opening ranges of the starting up process. According to the site tests, the opening speed of the guide vane was about $1^\circ/s$. The initial average guide vane opening of $0.05^\circ$ was chosen in this study, considering the limit of grid topology. The pressure boundary conditions, which were related to the average opening, were set according to Figure 11 with different end wall clearances. The static values were also subtracted from the total hydraulic torque to set the initial soft excitation.

Figure 15 shows two typical time histories of the guide vane opening during the pump mode’s starting up process. A small initial disturbance rapidly developed into high-amplitude vibration under the condition of $c^*_e = 0.06\%$. The minimum value of guide vane opening $\alpha_{1/2}$ was almost $0^\circ$, which implied that the neighboring guide vane blades could collide when at $t_{pp-max}$. The self-excited vibrations in these cases were unstable and threatening for the guide vane system. By contrast, the increase in the vibration amplitude was slow and limited under
the condition of $c_e^* = 0.39\%$. These self-excited vibrations were considered as tolerable due to the small maximum peak-to-peak value of the guide vane opening ($A_{pp\text{-}max}$).

Figure 15. The time histories of the guide vane opening during the pump mode’s starting up process: (a) under the condition of $c_e^* = 0.06\%$; (b) under the condition of $c_e^* = 0.39\%$.

The fluctuating cycle of guide vane opening with the maximum peak-to-peak value ($T_{pp\text{-}max}$) could be considered as an effective foundation to estimate the threat of self-excited vibrations for the guide vane system during the starting up process. In Figure 16, half of the peak-to-peak value ($0.5A_{pp\text{-}max}$) and the average opening ($\alpha_{pp\text{-}max}$) in this cycle are drawn as a function of end wall clearance. These two values are equal for the conditions of $c_e^* \leq 0.21\%$, which implied that the too large vibration amplitudes lead to the collisions between neighboring guide vane blades. For the conditions of $c_e^* \geq 0.21\%$, $0.5A_{pp\text{-}max}$ is smaller than the average opening. In these conditions, the self-excited vibrations have limited amplitudes and are considered to be tolerable.

Figure 16. Half of the peak-to-peak value and average opening of the fluctuating cycle $T_{pp\text{-}max}$ as a function of end wall clearance.

3.4. Analysis of the Influence of End Wall Clearance

Generally, for the pump mode’s starting up process of a pump turbine, the rotating speed of the runner begins to increase from 0 to 100% (with around 20% power load). During this time period, the runner chamber is kept pressurized with air to avoid instant overload. Then, air is released out of the runner chamber and the ball valve starts to open. The guide vane starts to open after staying closed for a while. Finally, the load increases to full load when the ball valve and guide vane are completely open. In the site tests of guide vane self-excited vibrations during the starting up process, three experiments were carried out with different starting up processes and end wall clearances, as shown in Table 3.
Table 3. The main settings for the site tests.

| Case       | Starting Up Process | End Wall Clearance, $c_e$ |
|------------|---------------------|---------------------------|
| Site test 1| A                   | 0.039%                    |
| Site test 2| A                   | 0.090%                    |
| Site test 3| B                   | 0.039%                    |

The time histories of the guide vane opening, after the ball valve started to open, are drawn in Figure 17. The guide vanes opened linearly from 0 to 100% throughout the starting up process A. For starting up process B, the guide vanes maintained a small opening angle (around 5%) for a while and then continually opened to 100%. The time difference between the moment $t_{a0}$ and $t_{b0}$ in Figure 17 was about 10 s. This led to a lower pressure level in the radial gap (between the runner and guide vanes) before the guide vanes started to open in site test 3.

In the site tests, the guide vane with the largest vibration amplitude was chosen to evaluate the vibrations of the guide vanes in the pump turbine. The horizontal vibration velocities of the guide vane were monitored when the guide vanes started to open ($\alpha_{gv}$ changed from 0 to 0.8°). The time histories and amplitudes of the horizontal vibration velocities are shown in Figure 18, where $C_{vv}$ and $A_{vv}$ are the values nondimensionalized by the peak-to-peak value of the vibration velocities in site test 3. In addition to the frequencies that were related to the runner blade passing by, the high amplitude with a special frequency ($15.73f_n$, $15.47f_n$, or $15.13f_n$) corresponded to the torsional mode natural frequency of the guide vane. Based on the site test results, a qualitative conclusion was reached that the guide vane vibration amplitude was relatively small for the starting up process with a lower pressure level or larger end wall clearances.

![Figure 17](image-url)  
*Figure 17. The time histories of the guide vane opening after the ball valve started to open.*

As for the URANS-1DOF coupling simulations in this study, they were carried out under simplified conditions due to the assumptions of the computational domains, boundary conditions and initial disturbances. However, they could help us to qualitatively understand the influence of end clearance on self-excited vibrations. The results under steady-state and starting up conditions showed that the instability of guide vane self-excited vibrations was weakened in cases with larger end clearances. This was consistent with the conclusions from site tests.
The guide vane self-excited vibrations were unstable under most steady-state conditions, as shown in Figure 14, apart from the operating condition of $\alpha_0 = 1^\circ$ and $c^\ast_e = 0.39\%$. By contrast, the self-excited vibrations in the starting up processes were tolerable for cases with large end wall clearances ($c^\ast_e = 0.3\%$ and 0.39%). This suggests that the critical condition for self-excited vibration instability was harsher when the transient process of the pump mode’s starting up was not considered. At the beginning of the pump mode’s starting up process, the guide vanes remain closed for a while to ensure there is sufficient pressure at the runner outlet, which is known as the pressure-making process. Due to the machining precision of a guide vane blade and the wear that is experienced during the operating process, there may be a small vertical clearance between two guide vane blades when they are theoretically closed. Regardless of whether there are small vertical clearances, the initial torsional vibration of a guide vane is most probably stimulated during the pressure-making process due to the high pressure level and complex high-speed flow at the runner outlet. This means that there may be enough time for small disturbances to develop into high-amplitude vibrations in the unstable cases. Therefore, the critical conditions of self-excited vibration instability that were obtained from the coupling simulations of the pump mode’s starting up processes have smaller safety margin due to the inaccurate estimations of the initial openings and disturbances. By contrast, there was enough time for the evolution of initial disturbances under steady opening conditions. The critical conditions from these cases have relatively large safety margins.

The above results show that large end clearances for guide vanes are beneficial in weakening the instability of self-excited vibrations, but they are unfavorable for hydraulic efficiency due to the leakage flow from the end wall clearance. More importantly, the guide vane vibration velocity was still large in site test 2, as shown in Figure 18, due to the complexity of the initial vibration excitations and the relatively small end wall clearances. Thus, increasing the end wall clearances is considered unreliable for suppressing guide vane self-excited vibrations and the collision of neighboring guide vane blades. On the other hand, the pressure difference between the inlet (runner side) and outlet (stay vane side) were lower for the cases with large end wall clearances. This suggests that reducing the pressure level in the radial gap between the guide vanes and the runner is effective for avoiding high-amplitude guide vane vibrations and strong pressure fluctuations at small openings by changing the starting up operating process. These were in good agreement with the conclusions from the site validation tests, in which the guide vane vibration velocity significantly decreased and the strong noises disappeared with the decreasing pressure level in the radial gap area during the starting up process.

![Figure 18](image_url)
4. Conclusions

The current study aimed to investigate the influence of end wall clearance on guide vane torsional mode self-excited vibration. Numerical simulations that were based on steady-state RANS methods and URANS-1DOF coupling modeling were carried out under different operating conditions.

The three-dimensional steady-state RANS calculations show that the influence of end wall clearance on the narrow gap flow rate cannot be ignored, especially at small guide vane openings and large end wall clearance widths. This leads to different pressure differences between the inlet (runner side) and outlet (stay vane side) at the same opening.

The URANS-1DOF coupling simulations under steady-state conditions and during the pump mode’s starting up process show that the guide vane self-excited vibrations are unstable under the conditions of small openings and end wall clearances. The high sensitivity of the pressure and velocity fields to torsional vibration, and the positive feedback between the hydraulic torque and vibrating amplitude, cause small disturbances to eventually develop into high-amplitude vibrations.

The critical conditions of self-excited vibration instability that were obtained from the coupling simulations under steady-state conditions have larger safety margins than those under the pump mode’s starting up processes. However, increasing the end wall clearances to suppress unstable self-excited vibrations is considered unreliable due to the complexity and randomness of the initial excitations. By contrast, reducing the pressure level in the radial gap between the guide vanes and the runner during the starting up operating process is effective for avoiding the high-amplitude vibrations and strong pressure fluctuations at small openings.

Author Contributions: Conceptualization, Q.L.; methodology, W.K. and L.Z.; software, W.K.; validation, Q.L.; investigation, Q.L., W.K. and L.Z.; writing—original draft preparation, Q.L.; writing—review and editing, W.K.; visualization, Q.L. and W.K.; supervision, Z.W.; project administration, L.Z. and W.K. All authors have read and agreed to the published version of the manuscript.

Funding: This work was supported by the National Natural Science Foundation of China (No.: 52079141).

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: Not applicable.

Conflicts of Interest: The authors declare no conflict of interest.

Abbreviations

The following abbreviations are used in this manuscript:

- $c_e$: Width of unilateral end wall clearance (m)
- $c^{*}_e$: Normalized width of unilateral end wall clearance, $c^{*}_e = \frac{c_e}{b}$
- $b$: Guide vane height (m)
- $C_\phi$: Flow rate coefficient, $C_\phi = \frac{\varphi}{U^2}$
- $U_1$: Circumferential velocity at runner inlet (m s$^{-1}$), $U_1 = \frac{2\pi n R_1}{60}$
- $n$: Rotating speed of runner (r min$^{-1}$)
- $R_1$: Runner inlet radius (m)
- $C_{\phi e}$: Flow rate coefficient of end wall clearance
- $C_{\phi o}$: Flow rate coefficient of blade-to-blade narrow gap
- $\varphi_{rin}$: Radial velocity at inlet (m s$^{-1}$)
- $C_p$: Static pressure coefficient, $C_p = \frac{p}{0.5 \rho U^2}$
- $p_s$: Static pressure (Pa)
- $C_v$: Velocity coefficient, $C_v = \frac{v}{0.5 \rho U^2}$
- $v$: Velocity (m s$^{-1}$)
\[ c^*_s \] Normalized axial coordinate value, \( z^* = \frac{z}{\delta} \)
\[ \delta \psi \] Guide vane inlet-outlet pressure difference, \( \delta \psi = \frac{p_{11} - p_{12}}{0.5 \rho U^2} \)
\[ p_{11} \] Total pressure at guide vane inlet (Pa)
\[ p_{12} \] Static pressure at guide vane inlet (Pa)
\[ \rho \] Density of fluid (kg m\(^{-3}\))
\[ C_T \] Hydraulic torque coefficient on guide vane, \( C_T = \frac{\tau}{\pi R_p^2} \)
\[ \tau \] Hydraulic torque on guide vane (N·m)
\[ M \] Mass of a guide vane (kg)
\[ g \] Gravitational acceleration (m s\(^{-2}\))

References

1. Stepanoff, A.J. Centrifugal and Axial Flow Pumps: Theory, Design, and Application; John Wiley & Sons, Inc.: Hoboken, NJ, USA, 1948.
2. Fraenkel, P.L. Water Lifting Devices; FAO Irrigation and Drainage paper 43; FAO: Rome, Italy, 1986.
3. Cacucciolo, V.; Shintake, J.; Kuwajima, Y.; Maeda, S.; Floreano, D.; Shea, H. Stretchable pumps for soft machines. Nature 2019, 572, 516–519. [CrossRef] [PubMed]
4. Mao, Z.; Iizuka, T.; Maeda, S. Bidirectional electrohydrodynamic pump with high symmetrical performance and its application to a tube actuator. Sens. Actuators Phys. 2021, 332, 113168. [CrossRef]
5. Zuo, Z.; Fan, H.; Liu, S.; Wu, Y. S-shaped characteristics on the performance curves of pump-turbines in turbine mode—A review. Renew. Sustain. Energy Rev. 2016, 60, 836–851. [CrossRef]
6. Li, D.; Wang, H.; Qin, Y.; Han, L.; Wei, X.; Qin, D. Entropy production analysis of hysteresis characteristic of a pump-turbine model. Energy Convers. Manag. 2017, 149, 175–191. [CrossRef]
7. Li, D.; Gong, R.; Wang, H.; Wei, X.; Liu, Z.; Qin, D. Numerical investigation on transient flow of a high head low specific speed pump-turbine in Pump Mode's. J. Renew. Sustain. Energy 2015, 7, 063111. [CrossRef]
8. Zhang, Y.; Zhang, Y.; Wu, Y. Pressure fluctuations in the vaneless space of High-head pump-turbines during normal turbine shutdown. J. Hydraul. Res. 2017, 55, 520–537. [CrossRef]
9. Zuo, Z.; Liu, S.; Sun, Y.; Wu, Y. Pressure fluctuations in the vaneless space of High-head pump-turbines—A review. Renew. Sustain. Energy Rev. 2015, 41, 965–974. [CrossRef]
10. Li, Z.; Bi, H.; Karney, B.; Wang, Z.; Yao, Z. Three-dimensional transient simulation of a prototype pump-turbine during normal turbine shutdown. J. Hydraul. Res. 2017, 55, 520–537. [CrossRef]
11. Pulpitel, L. Self-excited vibration of pump turbine guide vanes. In Proceedings of the IAHR 11th Symposium on Hydraulics Machinery, Equipment and Cavitation, Amsterdam, The Netherlands, 13–17 September 1982.
12. Nennemann, B.; Parkinson, E. YiXing pump turbine guide vane vibrations: Problem resolution with advanced CFD analysis. IOP Conf. Ser. Earth Environ. Sci. 2010, 12, 012057. [CrossRef]
13. Nennemann, B.; Sallabarger, M.; Henggeler, U.; Gentner, C.; Parkinson, E. Assessment of guide vane self-excitation stability at small openings in pump flow. IOP Conf. Ser. Earth Environ. Sci. 2012, 15, 062032. [CrossRef]
14. Tao, R.; Zhou, X.; Xu, B.; Wang, Z. Numerical investigation of the flow regime and cavitation in the vanes of reversible pump-turbine during Pump Mode’s starting up. Renew. Energy 2019, 141, 9–19. [CrossRef]
15. Liaghat, T.; Guifault, F.; Allenbach, L.; Nennemann, B. Two-way fluid-structure coupling in vibration and damping analysis of an oscillating hydrofoil. In Proceedings of the ASME International Mechanical Engineering Congress and Exposition, Montreal, QC, Canada, 14–20 November 2014; American Society of Mechanical Engineers: New York, NY, USA, 2014; Volume 46476, p. V04AT04A073.
16. Dörfler, P.; Sick, M.; Coutu, A. Flow-Induced Pulsation and Vibration in Hydroelectric Machinery: Engineer’s Guidebook for Planning, Design and Troubleshooting; Springer Science & Business Media: Berlin/Heidelberg, Germany, 2012.
17. Münch, C.; Ausoni, P.; Braun, O.; Farhat, M.; Avellan, F. Hydro Elastic Behavior of Vibrating Blades. In Proceedings of the 24th Symposium on Hydraulic Machinery and Systems, Foz do Iguassu, Brazil, 27–31 October 2008; Volume 1, p. 10.
18. Münch, C.; Ausoni, P.; Braun, O.; Farhat, M.; Avellan, F. Fluid–structure coupling for an oscillating hydrofoil. J. Fluids Struct. 2010, 26, 1018–1033. [CrossRef]
19. Guo, Q.; Zhou, L.; Wang, Z.; Liu, M.; Cheng, H. Numerical simulation for the tip leakage vortex cavitation. Ocean Eng. 2018, 151, 71–81. [CrossRef]
20. ANSYS ANSYS CFX-Solver Modeling Guide: Release 18.0; Ansys Inc.: Canonsburg, PA, USA, 2017.
21. Celik, I.B.; Ghia, U.; Roache, P.J.; Freitas, C.J. Procedure for estimation and reporting of uncertainty due to discretization in CFD applications. J. Fluids Eng. 2008, 130, 078001. [CrossRef]