Influence of prewhirl angle on a centrifugal pump with inlet guide vane running at turbine mode

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Abstract. The prewhirl regulation by inlet guide vanes (IGVs) has proved to be an effective method for optimal operation of the centrifugal pump. Pump as turbine (PAT) is a simple approach for small hydropower generation. The energy performance of centrifugal pump with IGVs running in turbine mode is investigated by numerical simulations under different prewhirl angles and flow rates. It is found that the best efficiency point (BEP) shifts towards large flow rate. The efficiency with GVs in turbine mode is slightly lower than that without GVs. The average efficiency decline is 0.46% with the appropriate prewhirl angle. With the installation of GVs, the shaft power is even somewhat improved, which indicates that PAT with GVs can recover the energy from fluid more effectively. The investigation of internal flow field shows an obvious vortex downstream GVs due to the strong rotation behind the impeller, and appropriate prewhirl angle of GVs is helpful to reduce the vortices strength.

1. Introduction

Hydropower is one of the most reliable renewable energy worldwide. Because of the complicity of large hydropower, including huge reservoir and human rehabilitation, the exploitation and utilization of small hydropower have attracted public attentions. However, conventional Francis and Pelton turbines are not appropriate for the small hydropower station because they are specially designed for large hydropower system. As an alternative method, the Pump as Turbine (PAT) has become an effective method, which runs the pump reversely. The pump is a relatively simple machine with low manufacturing cost and stable operation, and hence it is economically feasible in small hydropower stations.

In order to provide the guidance for selection of PATs, many investigation has been conducted to obtain the characteristic curves. Experimental results showed that the PAT operated at higher head and discharge than pump mode. Bozorgi [1] studied the axial pump as turbine for Pico hydropower plants experimentally and numerically. Barbarelli [2] established a statistical model for finding suitable pump based on sufficient experimental measurement. Many efforts have been made to relate the performance in turbine mode and in pump mode. Derakhshan [3] measured the performance curves of centrifugal pumps as turbines under various specific speed and derived a model to obtain the Best Efficiency Point (BEP) of PAT. The predicted results were in good coincidence with experimental data. The method was employed by Pugliese [4] in a centrifugal horizontal single-stage pump. It was found that the model of Derakhshan for head still worked outside the studied range while it failed for power. Yang [5] proposed a theoretical method to predict the performance of PATs based on theoretical analysis and empirical
correlation. Barbarelli [6] developed a one-dimensional model to estimate the performance of PAT, and the accuracy was validated by the experimental measurement of 6 centrifugal pumps with specific speed varied from 9 to 65. For performance enhancement of PAT, studies have been carried to investigate the effect of geometry and operation parameters, including impeller rounding [7, 8], impeller trimming [9], impeller diameter [10, 11], and impeller blade [12].

Generally, the pump is designed for rated discharge and pump performance will deteriorate under off-design conditions [13, 14], which affects the safe and stable operation [15-17]. The operation regulation of pump has been fully studied, and the prewhirl regulation by Inlet Guide Vanes (IGVs) proves to be an effective method for optimal operation of centrifugal pumps. The IGVs are located at the inlet of turbomachine and can alter the mass flow rate. The prewhirl regulation by IGVs is initially applied in compressors [18-20], and is introduced to centrifugal pumps by Tan [21]. Experimental and numerical results showed that the pump efficiency could be improved and the high efficiency scope could be extended. The cavitation performance [22] and unsteady characteristics [23] of centrifugal pumps with IGVs have been researched, and the effect of geometry structure including axial distance [24, 25] and clocking effects [26] has been estimated.

Despite various study on the PATs and prewhirl regulation by IGVs, the application of guide vanes on PATs is relatively rare. The inlet guide vanes (IGVs) in pump mode will become the outlet guide vanes (OGVs) in turbine mode when pump works reversely. In the current research, the centrifugal pump with inlet GVs as turbine will be investigated numerically. The energy performance will be estimated and the internal flow field will be analyzed according to numerical results.

2. Physical model and computational domain

2.1. Physical model
A single-stage centrifugal pump without or with IGVs is employed in the current study. The IGVs are installed upstream of the impeller, with hydrofoil located in the circumferential direction evenly. Detail information can be found in Ref. [27]. Table 1 shows the main parameter of the pump system. The geometry structure of GVs is determined by the hydraulic design method proposed by Tan [21], as illustrated in figure 1.

| Component       | Item                              | Value |
|-----------------|-----------------------------------|-------|
| Centrifugal pump| Volumetric rate $Q$ (m$^3$/h)     | 340   |
|                 | Head $H$ (m)                      | 30    |
|                 | Rotational speed $n$ (r/min)      | 1450  |
|                 | Numbers of blade $Z_i$            | 6     |
|                 | Diameter of suction pipe $D_0$ (mm)| 200  |
|                 | Diameter of impeller outlet $D_2$ (mm)| 329  |
|                 | Hub of IGVs (mm)                  | 40    |
|                 | Shroud of IGVs (mm)               | 200   |
|                 | Number of vanes $Z_g$             | 6     |
|                 | Blade angle at inlet $\beta_{gi}$ (°)| 90   |

Table 1. Parameters for the centrifugal pump with IGVs.
2.2. Computational mesh

The numerical simulations are employed in the current study and the computational domain is made up of three parts, including suction pipe without and with GVs, impeller, and volute. Figure 2 depicts the schematic diagram of the computational domain. In the test pump system, the GVs are located inside the suction pipe along the circumferential direction, and their central line is 380 mm away from the impeller outlet in turbine mode. In order to investigate the effect of GVs under different prewhirl angle γ, totally six computational models are simulated in the following study, including PAT without GVs and PAT with GVs at prewhirl angles of γ = 0°, ±12°, and ±24°. The prewhirl angle is defined as positive when the prewhirl direction is the same as the rotation direction of the impeller in pump mode.
Figure 3. Mesh distribution on GVs and impellers.

The commercial mesh generating code ANSYS TurboGrid and ICEM 14.5 are employed to provide the structured meshes for the impeller and suction pipe with GVs, and the unstructured mesh for volute, respectively. By means of O-block topology and local refinement around the blades of GVs and impeller, the flow inside the boundary layer can be captured and the simulation accuracy can be guaranteed.

Figure 3 displays the mesh distribution on GVs and impellers.

3. Numerical settings

3.1. Numerical method

The commercial Computational Fluid Dynamics (CFD) codes ANSYS CFX 14.5 is applied in the current study and the three-dimensional Reynolds-Average Navier-Stokes equations are solved to obtain the internal flow field inside the centrifugal PATs. The Renormalization Group (RNG) $k$-$\varepsilon$ turbulence model is selected to simulate the turbulent flow. The boundary conditions in turbine mode consist of an imposed static pressure at PAT inlet, a given mass flow rate at PAT outlet and no slip conditions at other wall surfaces. The technique of frozen rotor is used to connect the rotational domain (impeller) and stationary domain (suction pipe and volute). When the root-mean-square residual is less than $10^{-4}$, the steady simulation can be stated to be converged.

3.2. Independence test of mesh number

The independence test of mesh number is carried out with four sets of computational mesh. During the independence test, the mesh of impeller gradually refined while the mesh of suction pipe and volute remains, and the total mesh number varies from 2.80 million to 4.11 million. It can be found that when the mesh number exceeds 2.80 million, there appears tiny variation among the predicted values of PAT performance ($\eta$, $H$, $P_{\text{shaft}}$ are efficiency, head, and shaft power, respectively), with a maximum change of 0.55%. Therefore, the independence of mesh number on numerical simulation can be demonstrated, and the Mesh 2 of 2.86 million elements is finally employed in the following study in the account of simulation accuracy and resource consumption.

Table 2. Independence test of mesh number.

| Item          | Mesh 1  | Mesh 2  | Mesh 3  | Mesh 4  |
|---------------|---------|---------|---------|---------|
| Volute        | 674234  | 674234  | 674234  | 674234  |
| Impeller      | 1117260 | 1180440 | 1793880 | 2431800 |
| Suction pipe  | 1008420 | 1008420 | 1008420 | 1008420 |
| Total meshes  | 2799914 | 2863094 | 3476534 | 4114454 |
| $\eta$ (%)    | 75.06   | 75.18   | 75.06   | 75.04   |
| $H$ (m)       | 29.27   | 29.32   | 29.40   | 29.43   |
| $P_{\text{shaft}}$ (kW) | 22.00   | 22.07   | 22.09   | 22.11   |
| $|\eta-\eta_1|/\eta_1$ | 0.00%   | 0.16%   | 0.00%   | 0.03%   |
| $|H-H_1|/H_1$    | 0.00%   | 0.17%   | 0.44%   | 0.55%   |
| $|P_{\text{shaft}}-P_{\text{shaft},1}|/P_{\text{shaft},1}$ | 0.00%   | 0.32%   | 0.41%   | 0.50%   |

3.3. Simulation accuracy validation

The performance of centrifugal pump without and with IGVs has been measured experimentally in our previous research [21], with maximum comprehensive error of measurement within $\pm 0.358\%$. These experimental results are employed in the current research to validate the accuracy of numerical simulations. The comparison between experimental and numerical results are illustrated in figure 4. It can be found that the numerical results are in good coincidence with the experimental measurements.
4. Results and discussions

4.1. Characteristic curves

Figure 5 depicts the characteristic curve of a centrifugal pump with IGVs running in turbine mode at different prewhirl angles. Compared with the operation without GVs in pump mode, the BEP is shifted towards large flow rate, from 340 m$^3$/s (pump mode) to 460 m$^3$/s (turbine mode). It can be found that the prewhirl angle has a significant influence on the BEP. For zero prewhirl angle $\gamma = 0^\circ$, the location of BEP is almost the same as that without GVs. By contrast, it moves towards the left side (partial flow rate) under positive prewhirl angle $\gamma = 12^\circ$ and $24^\circ$, and it shifts towards the right side (large flow rate) under negative prewhirl angle $\gamma = -12^\circ$ and $-24^\circ$. With the increasing of the absolute value of prewhirl angle, the characteristic curves of efficiency become steep gradually. The characteristic curves of efficiency indicates that appropriate prewhirl angle is necessary to maintain the performance of PAT. Positive and negative prewhirl angles are required for the partial flow rate and large flow rate, respectively.
It can be found that the PAT efficiency decreases slightly when GVs are installed. Despite somewhat decline of PAT efficiency after the installation of GVs, the average reduction is only 0.46% when prewhirl angle is settled reasonably. The maximum efficiency drop appears in the condition of $Q = 500$ m$^3$/s with $\gamma = 24^\circ$ and $Q = 340$ m$^3$/s with $\gamma = -24^\circ$, namely, which are 2.47% and 2.74%. This phenomenon shows that large prewhirl angle should be avoided because it may lead to remarkable performance degradation when the operation condition deviates from the BEP. Though the PAT efficiency is decreased, the PAT head is enhanced slightly in centrifugal PAT with GVs, with an average increase of 1.15%. It can be found that the highest head rise also occurs in the condition of $Q = 500$ m$^3$/s with $\gamma = 24^\circ$ and $Q = 340$ m$^3$/s with $\gamma = -24^\circ$, which corresponds with the maximum efficiency drop. For PAT system, the shaft power is determined by $P_{\text{shaft}} = q_m g H \eta$, where $q_m$ is mass flow rate and $g$ is gravitational acceleration. Due to both effect of efficiency decline and head enhancement, the shaft power nearly remains unchanged between the PATs without and with GVs, with a tiny increase of 0.01%. In the account of the requirement of shaft power as high as possible, it can be stated that the equipment of GVs is beneficial for the performance enhancement of centrifugal PAT, which can recover the energy from fluid more effectively.

**4.2. Internal flow field**

In order to investigate the influence of GVs and prewhirl angle on the performance of centrifugal PATs, the internal flow field at BEP point ($Q = 460$ m$^3$/s) is illustrated and analyzed. Figure 6 depicts the three-dimensional streamlines impeller and suction pipe. The installation of GVs indicates a significant role in the flow field downstream the impeller. In the condition of PAT without GVs, the flow downstream the impeller appears with remarkable rotation. When the GVs is equipped, the rotational motion is reduced under an appropriate prewhirl angle ($\gamma = 0^\circ$ and $\pm 12^\circ$). The flow direction tends to be axial at the outlet of the PAT system, which is beneficial for the utilization of flow energy in the following stage. In contrast, the large prewhirl angle ($\gamma = \pm 24^\circ$) may accelerate the rotational motion, and leads to efficiency decline.
Figure 6. Three-dimensional streamline inside centrifugal PAT.

Figure 7. The distribution of pressure coefficient and streamline on the plane downstream GVs. The pressure coefficient is defined by $c_p = \frac{(p_{\text{inlet}} - p)}{0.5\rho u_2^2}$, where $p_{\text{inlet}}$ is the pressure at PAT inlet, $\rho$ is the density, and $u_2$ the circumferential velocity of impeller. An obvious vortex exists in the centrifugal PAT without GVs, which is induced by the strong rotation of impeller. The vortex is divided
into several smaller parts in the PAT with GV$s at $γ = 0°$ due to the guide effect of GV$s passage. At prewhirl angle of $12°$, the vortex in the plane has been dismissed totally, which indicates that appropriate prewhirl angle is necessary for performance enhancement of PAT with GV$s$. The vortices occur again at prewhirl angle of -12°, and the location of vortices has changed. For PAT without GV$s$ and with GV$s$ at $γ =0°$, the vortex appears in the region near hub whereas it appears in the region near shroud in the condition of PAT with GV$s$ at $γ =12°$. This shows that inappropriate prewhirl angle may result in extra vortex and deteriorate the PAT performance.

5. Conclusion
The numerical studies are carried out on the centrifugal pump with inlet GV$s$ running in turbine mode. The accuracy of numerical simulations has been demonstrated by the comparison with experimental results. The characteristic curves of centrifugal PAT without GV$s$ and with GV$s$ at five prewhirl angles ($γ = 0°$, ±12°, and ±24°) are illustrated and the internal flow fields are analyzed. According to the numerical analysis, the following conclusions can be drawn:

1) After the equipment of GV$s$, the PAT efficiency is decreased slightly. The average efficiency decline is 0.46% when appropriate prewhirl angle has been set. The positive values are required at the left side of BEP, and the negative values should be selected at the right side of BEP.

2) The head of PAT with GV$s$ is increased by an average of 1.15% compared with that without GV$s$, and hence there appears tiny enhancement of shaft power, which indicates that PAT with GV$s$ can recover the energy from fluid more effectively. Larger prewhirl angle can lead to the higher head but not higher shaft power necessarily.

3) The GV$s$ can reduce the strong rotation of fluid downstream from the impeller, and the vortex can be dismissed completely at BEP with $γ = 12°$.

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