Influence of guide vane setting in pump mode on performance characteristics of a pump-turbine

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Abstract. Performance characteristics in pump mode of pump-turbines are vital for the safe and effective operation of pumped storage power plants. They are resultant of Euler head (power input) and hydraulic losses (power dissipation). In this paper, 3-D steady simulations were performed under 13mm, 19mm and 25mm guide vane openings (GVOs). Three groups of operating points under the three GVOs were chosen based on experimental validation to investigate the influence of guide vane setting on flow patterns upstream and downstream. Analysed results show that, the guide vane setting will obviously change the flow pattern downstream, which in turn influences the flow upstream. It shows a strong effect on hydraulic losses in guide and stay vanes. In addition, at the large part load conditions, the change of GVO will increase the relative flow angle at the runner outlet. As a consequence, it decreases the Euler head. However, at other operating conditions, it only has a little influence on Euler head. Flow patterns in pump mode are very dependent on the GVO and discharge.

1. Introduction
As the requirements of the electric grid, pump-turbines are supposed to higher head, larger capacity and higher specific speed. This leads to much more problems for pump-turbines, such as higher pressure fluctuation, more instabilities in the transient processes. A pump-turbine is a compromise between a pump and a turbine. Generally, pump-turbines are designed on the performances of the pump for the priority considering with turbine characteristics. In such case, there are two unstable phenomena in the pump-turbines. One occurs in pump mode as a head at small discharge and high head conditions in discharge-head curve (hump characteristic), and another one appears in turbine mode at low load off-design conditions (S-shape characteristic)[1].

The hump instability will limit the operation head, and influence the start-up. Hence it is significantly important to investigate the performance characteristics in pump mode of pump-turbines.

Recently, more and more researchers put emphasis on the pump characteristics of pump-turbines. Lots of achievements have been obtained through the experimental and numerical methods. Yin et al. [2] carried out performance prediction and flow analysis in the vaned distributor of a low specific speed pump-turbine. It was found that special discharge-head profile comes from the special hydraulic losses in the stay vanes and guide vanes. Li et al. [3] found that the drooping behavior comes from both the incidence/deviation effect and hydraulic losses.
losses, through the three-dimensional (3-D) steady simulations based on Euler theory. Unsteady simulations for the same pump-turbine were also performed to correlate the relation between vortex in stay and guide vanes and hump characteristics [4], and also the distribution of the frequencies in the hump region was revealed [5]. Furthermore, many experiments in pump mode of pump-turbine were carried out. Ran et al. [6] analyzed the pressure fluctuations at large partial flow conditions, and found rotating stall in the flow passages of the wicket gates. Yang et al. [7] characterized two unsteady structures when the pump-turbine operated at part load conditions. Additionally, some researchers [8-11], found the cavitation is related to hump characteristics and carried out lots of studies.

Nevertheless, the studies about influence of the GVO on the pump characteristics of pump-turbines, especially for hump characteristics, are rather limited. More recently, Li et al. [12] analyzed the flow characteristics in pump mode under different GVOs through simulations based on the experimental validation, but only in the stay and guide vanes parts. In addition, they [13] obtained unstable discharge-head characteristics under different GVOs and revealed the variation of the hump characteristics. Although these research is related to performance characteristics under different GVOs, the influence of GVO on the performance was not reported.

In this paper, firstly, 3-D steady simulations were carried out to predict the performance characteristics under 13mm, 19mm and 25mm GVOs based on the experimental validation. Then, three groups of operating points under the three GVOs were chosen. Each group has almost the same discharge. In such case, based on Euler theory and hydraulic loss analysis, we could find the effect of downstream geometric conditions on the performance characteristics.

The net head is resultant of Euler momentum (input) and hydraulic losses (dissipation). Euler momentum \((\Delta c_u\cdot u)\) is the function of incidence angle \((c_{u1})\) at inlet and deviation angle at the outlet \((c_{u2})\). \((\Delta c_u\cdot u)\) can be determined by (1) and (2). For experimental methods, (1) gives an approximate value of Euler torque. However, both (1) and (2) can be used while working with the numerical results [3].

\[
g(H_1 - H_2) = \Delta c_u \cdot u = c_{u2} \cdot u_2 - c_{u1} \cdot u_1 = T \cdot \omega / Q \tag{1}
\]

\[
\Delta c_u \cdot u / g = H_{\text{net}} + H_{\text{losses}} = H_{\text{input}} \tag{2}
\]

2. Numerical methods
A pump-turbine model (shown in figure 1) including the runner with 9 blades, 20 guide vanes, 20 stay vanes, the draft tube and the spiral casing was investigated in this research. The sketch of the pump-turbine is shown in figure 2. The runner inlet diameter \(D_1\), outlet diameter \(D_2\) and guide vane height \(B_0\) are respectively 274mm, 524mm and 45.77mm. The specific speed \(n_q\) is 30.7, which belongs to a low one.

![Figure 1. Computational domain](image1)
![Figure 2. Sketch of the pump-turbine](image2)

ANSYS ICEM 14.0 was used to create the mesh. Except tongue of the spiral casing, for all the rest parts, the meshes are generated by structured grids. The \(y^+\) of blades surfaces (runner blades, stay and guide vanes) are less than 2 in average. The total number of nodes is 7.92 million. All the information about grid, as well as grid independency validation were described in the paper published by the authors in the before [14].
ANSYS CFX 14.0 was applied to carry out the steady incompressible turbulent flow numerical simulations. The finite volume method with High resolution scheme used for both convection and diffusion terms was chosen to solve Reynolds Navier-Stokes equations. In this research, SST k-ω turbulence model was chosen to perform the simulations. The validation of the turbulence model in pump mode of the pump-turbine was shown in the paper [12]. Root mean square (RMS) is set less than 10^{-6}.

Static pressure (0 Pa) was set at the draft tube inlet (pump mode), while the discharge was specified at the spiral casing outlet according to the experimental data. No-slip conditions were used for all the rest walls.

3. Experimental validation

Comparison of performance characteristics between simulations and experiments under 13mm, 19mm and 25mm GVOs is shown in figure 3. Corresponding error variation is also depicted in figure 3. The error is calculated according to following equation.

$$\text{Error} = \frac{\text{Experiment} - \text{Simulation}}{\text{Experiment}} \times 100\% \quad (3)$$

As for 13mm GVO, from figure 3-a, the head is underestimated at over-load and large part-load operating conditions and overestimated at the hump instability operating conditions. The error decreases from the maximum 9.96% at the highest discharge condition until best efficiency point (BEP), and then shows an increase tendency as the discharge is reduced. The torque is underestimated in the whole range through numerical methods, and the error increases with the reduction of the discharge with maximum 7.29% at the lowest discharge condition. The variation of error for efficiency is almost the same with the head. The maximum error (9.42%) occurs at the highest discharge operating condition.

As for 19mm GVO, from figure 3-b, the heads at the over-load and large part-load operating conditions are overestimated, while they are underestimated around the BEP and in the hump region. The maximum error (6.68%) occurs at the highest discharge operating point. The torque predicted at the over-load operating conditions is much larger than experimental torque, while it is lower at the large part-load operating conditions. The error increases as the discharge increases or decreases from the BEP, and the maximum error (9.92%) appears at lowest discharge operating point. The efficiency at the off-design operating points is overestimated, and the error increases when the discharge is far away with BEP.

As for 25mm GVO, from figure 3-c, the head in the whole range is overestimated, and the error in the whole range is less than 3%. The variation of error for torque is almost the same with the ones of 13mm and 19mm GVOs. The maximum error (5.45%) is located at the lowest discharge operating condition. The efficiency in the whole range is overestimated. The largest error (7.13%) occurs at the lowest discharge operating condition.

In a sum, the errors for the head, torque and efficiency decrease with the increase of the GVO. The head at the off-design operating conditions of 13mm GVO is underestimated, while it is overestimated for most operating points for 19mm and 25mm GVOs. At small GVO operating conditions, flow separation, back flow and vortex motion are serious, which could be not predicted properly. This might lead to the numerical head lower than experiments. However, for large GVO operating conditions, unstable flow pattern is relatively little, and the some losses are ignored in the simulations. Hence, the head predicted is much larger than experiments. For the torque, it is found that, at any GVO, below the discharge 240l/s, it is underestimated, and the maximum error occurs at the lowest discharge operating condition. The variation of error is almost the same with the head. Although there exists large error at off-design operating points in the steady simulations. The tendency of performance characteristics predicted is almost the same with experimental data. In addition, the drooping phenomena for head could be captured although the positions have a bit shift. The numerical analysis could be carried out based on the simulations.
4. Analysis of performance characteristics

4.1 Variation of Euler head

The net head and Euler head are shown in figure 4, and three groups of analysed points are shown in figure 5. The detailed information of analysed points is listed in Table 1. From figure 4, it could be observed that there is only a little difference for Euler head. Euler head is dependent on the relative flow angle at the inlet and outlet of the runner blades. The net head is resultant of the Euler head and hydraulic losses.

According to equations (4) and (5) [15], one can obtain the relation between Euler head and other parameters (circumferential velocity, discharge, area and relative flow angle at the runner inlet and outlet).

\[
gH_{\text{input}} = c_{u2} - c_{u1} - t\left(u_2 - c_2 + \tan(\beta_2) - u_1 - c_1 - \tan(\beta_1)\right) \quad (4)
\]

\[
= u_2 (Q/A_2 \tan(\beta_2) - u_1 Q/A_1 tan(\beta_1)) \quad (5)
\]

For every group, the discharge under any GVO is almost the same, and the circumferential velocity is also the same at the runner inlet and outlet. Hence, the difference of the Euler head only comes from the relative flow angle at the runner inlet and outlet. Relative flow angle is angle between the circumferential velocity and relative velocity. Relative flow angle averaged from runner inlet to runner outlet is extracted as shown in figure 6.

For group A, the Euler head of 19mm GVO shows much larger than other two GVOs. From the figure 6, it can be seen that averaged flow angle from streamwise 0 (runner inlet) till streamwise 0.9 (runner out is streamwise 1.0) is the same for these three operating points, while
it shows different from streamwise 0.9 to streamwise 1.0. It can be concluded that the difference of Euler head for three GVOs is only dependent on relative flow angle at the runner outlet. Due to the change of the GVO, the relative flow angle at the runner outlet is influenced. For point $A_{19}$, relative flow angle at the runner outlet is largest. The second is the one of point $A_{25}$, and the lowest one is point $A_{13}$. Hence, the Euler head of point $A_{19}$ is the highest, and the one of point $A_{13}$ is the lowest, which is same as shown in figure 4.

For group B and group C, the difference of Euler head for three GVOs is rather little. In addition, the discharge for every groups has a little difference. Hence, for groups B and C, the relative flow angle is not influenced seriously due to the change of the GVO.

4.2 Variation of hydraulic losses

From the figure 4, it can be seen that hydraulic loss is the most important part to cause the change of the net head. For every group, hydraulic losses of every parts are shown in figure 7. For group A, the total losses of point $A_{25}$ are the highest, and then point $A_{19}$, and the lowest is point $A_{13}$. Hydraulic losses in the runner, and guide vanes decrease with the decrease of the GVO, while ones in stay vanes increases sharply. Hydraulic losses mainly occur in the runner, guide vanes and stay vanes. Although the Euler head of 19mm is highest, hydraulic losses is much high. Hence, the net head of point $A_{19}$ is not highest. The distribution of streamline in the draft tube, stay and guide vanes, and runner (midsapn 0.5) is shown in figure 8. Frome figure 8, it is found that the vortices in stay vanes increase obviously as the GVO is reduced, while the flow separation in the guide vanes decreases. Serious flow separation in guide vanes would lead to backflow in the vaneless space. From figure 8, under 19mm and 25mm GVOs, some reversal vectors in the vaneless space could be observed. This may lead to the increase of
hydraulic losses in the runner and guide vanes. In the draft tube, when the discharge is the same, the hydraulic losses in the draft tube is nearly the same., and the distribution of streamline in the draft tube is almost the same. Furthermore, for these operating points, obvious backflow and vortices in the cone of the draft tube could be found. It confirms that there exists pre-rotation in the runner inlet. Combining with figure 7-a, it can be concluded that the hydraulic losses in stay vanes come from the vortices, and ones in guide vanes result from flow separation, and ones in the runner come from the backflow in the vaneless space.

For group B and C, the change of net head only depends on the hydraulic losses. From figures 7-b and c, the variation of total hydraulic losses is almost the same. With the reduction of the GVO, the total hydraulic losses increase sharply. Most of hydraulic losses distribute in the runner, guide vanes and stay vanes.

For group B, operating points of the 19mm and 25mm GVOs are in the hump region. With respect to these two points, more than 40% of hydraulic losses occur in the runner, while less than 30% of hydraulic losses is from the runner for the point B13. From figure 9, no obvious flow separation in the guide vanes at the point B19 and B25 could be found. However, six stable vortices distributing equally at the circumference block most of stay vane passages at B13. This may be rotating stall, which need be confirmed through unsteady simulations. Hydraulic losses in the stay vanes, guide vanes and spiral casing have a sharp increase, which maybe come from the blockage in the stay vane. The vortices close to the wall in the cone disappear, and the two symmetrical vortices appear in the inside of the elbow as the discharge is increased.

For group C, operating points are close to BEP. The runner blades are designed at the BEP. There is no two much flow separation in the runner. Hence, the proportion of hydraulic losses in the runner decreases obviously, and the runner losses are almost the same for the three GVOs. However, the losses in the guide vanes, stay vanes and spiral casing have a dramatic increase with reduction of the GVO. From the figure 10, for 13mm GVO, serious separation in stay vanes could be found, while only several obvious separation vortices exist unlike group A and B. For point C19, four groups of stable vortices distribute equally at the circumference. This also maybe rotating stall. The hydraulic losses in the draft tube for group C are also the same. From figure 10, the vortices in the draft tube are almost the same with group B.

![Figure 7. Variation of hydraulic losses for every parts](image)
Figure 8. Distribution of streamlines for group A

(a) 13mm 128.37l/s
(b) 19mm 126.95l/s
(c) 25mm 127.88l/s

Figure 9. Distribution of streamlines for group B

(a) 13mm 179.61l/s
(b) 19mm 178.25l/s
(c) 25mm 180.24l/s

Figure 10. Distribution of streamlines for group C

(a) 13mm 201.69l/s
(b) 19mm 206.76l/s
(c) 25mm 206.66l/s
In order to obtain the change of direction and amplitude of velocity in stay and guide vanes, vectors in each group are shown in figures 11 to 13.

For group A, from figure 11, the velocity increases in guide vane passages as the GVO is reduced. Reversal vectors could be observed at guide vane inlet for point A25. Combining figure 8, flow separation areas (red box) shift from the guide vane inlet to stay vane outlet. Hence, hydraulic losses in guide vanes decrease, while ones in stay vanes increase with the decrease of the GVO. Also, it can be found that the direction of inlet flow at the guide vane changes. It can be concluded that hydraulic losses mainly come from the flow separation at the suction surfaces of guide vanes, and flow pattern in the vaneless space (between runner blade outlet and guide vane inlet) is influenced due to the decrease of the GVO, which could lead to change of the Euler head. It is same with the result shown in figure 6.

For group B, from figure 12, the amplitudes of velocity only have a little change due to the increase of the GVO unlike the group A. Combining figure 9, it can be seen that, flow separation at pressure surfaces of guide vanes at point B25 shifts to the trailing edges at point B19 and becomes weak, and then disappears in the guide vanes at point B13. Hence, hydraulic losses in guide vanes decrease from points B25 to B19. However, compared with points B25 and B19, hydraulic losses of point B13 in guide vanes increase. From figure 9-a, serious blockage in stay vane passages could be found, and also, at the stay vane outlet of each blockage passage, there generates a secondary vortices due to complete blockage of the first separation vortices appearing at the pressure surfaces of stay vanes. From figure 13-a, it shows that complete blockage leads to the change of flow direction upstream and decrease of amplitudes shown in red box (figure 12-a). Consequently, this increases the amplitudes of velocity in adjacent passage. This phenomenon usually comes from a typical rotating stall. Hence, the increase of hydraulic losses in guide vane for B13 should be from complete blockage downstream.

For group C, from figure 13, the decrease of the GVO also changes the direction and the amplitude of velocity. However, at point C13, the amplitudes of velocity in guide vanes decrease. Although stable vortices in stay vanes are much less than point B13, flow separation is much more serious. Hence, it blocks the stay vanes channels and slows down the velocity upstream (in guide vanes). Consequently, hydraulic losses in stay and guide vanes increase compared with points C19, C25 and B13.
5. Conclusions
The paper analyzed the influence of the change of the GVO on performance characteristics in pump mode of a pump-turbine through 3-D steady simulations based on experimental validation. Three groups of operating points at three GVOs (13mm, 19mm and 25mm) were chosen to investigate the variation of hydraulic losses and flow characteristics. Generally speaking, with the increase of the discharge at one GVO, the velocity in guide vane increases, flow separation shifts from the suction surfaces of guide vanes to stay vanes, and separation vortices and backflow in guide vanes decrease and become unclear. For the same discharge at different GVOs, due to the decrease of GVO, the velocity in the guide vane increases. The flow separation in the guide vanes and vaneeless space is improved as the increase of the velocity, while it in stay vanes is promoted. However, at some operating conditions (group A), the flow patterns downstream (stay vanes) in turn improve the flow upstream (guide vane and runner), performance characteristics show extremely different with other operating conditions. It can be concluded that the flow patterns downstream, caused due to the change of the GVO, will have an obvious influence on the flow pattern upstream. Namely, Euler head and hydraulic losses are influenced by the change of the GVO and the change of the discharge. As a result, performance characteristics which are resultant of Euler head and hydraulic losses are changed. At some operating conditions, they have a good condition to produce stable vortices, while in other conditions, they generate unstable patterns, including serious flow separation, backflow and secondary vortices. Furthermore, the positions of flow separation, the number of vortices and the blockage channels are very dependent on the change of the GVO and discharge.

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References
[1]. Gentner C, Smiljcic Z and Sallaberger M 2010 Analysis of Unstable Pump Characteristics of Pump Turbines. In: 16th International Seminar on Hydropower Plants-Reliable Hydropower for a Safe and Sustainable Power Production, Vienna, Austria
[2]. Yin J L, Liu J T, Wang L Q, Jiao L, Wu D Z and Qin D Q 2010 Performance Prediction and Flow Analysis in the Vaned Distributor of a Pump Turbine Under Low Flow Rate in Pump Mode SCIENCE CHINA Technological Sciences 53 3302-9
[3]. Li D Y, Gong R Z, Wang H J, Fu W W, Wei X Z and Liu Z S 2015 Fluid Flow Analysis of Drooping Phenomena in Pump Mode for a Given Guide Vane Setting of a Pump-Turbine Model Journal of Zhejiang University-SCIENCE a (Applied Physics & Engineering) 16 851-63
[4]. Li D Y, Wang H J, Xiang G M, Gong R Z, Wei X Z and Liu Z S 2015 Unsteady Simulation and Analysis for Hump Characteristics of a Pump Turbine Model RENEW ENERG 77 32-42
[5]. Li D Y, Wang H J, Xiang G M, Gong R Z and Qin D Q 2014 Frequencies Spectrum Analysis for Hump Characteristic in a Pump Turbine. In: 6th International Symposium on Fluid Machinery and Fluid Engineering, Wuhan, China
[6]. Ran H J, Luo X W, Zhu L, Zhang Y, Wang X and Xu H Y 2012 Experimental Study of the Pressure Fluctuations in a Pump Turbine at Large Partial Flow Conditions CHIN J MECH ENG-EN 25 1205-9
[7]. Yang J, Pravesi G, Yuan S, Cavazzini G and Ardizzon G 2015 Experimental Characterization of a Pump-Turbine in Pump Mode at Hump Instability Region Journal of Fluids Engineering 137 1-11
[8]. Liu J T, Liu S H, Wu Y L, Jiao L, Wang L Q and Sun Y K 2012 Numerical Investigation of the Hump Characteristic of a Pump–Turbine Based On an Improved Cavitation Model Computer & Fluids 68 105-11
[9]. Liu J T, Wu Y L and Liu S H 2013 Study of Unsteady Cavitation Flow of a Pump-Turbine at Pump Mode. In: 6th International Conference on Pumps and Fans with Compressors
[10]. Li D Y, Wang H J, Xiang G M, Gong R Z and Wei X Z 2015 Investigation On Cavitation for Hump Characteristics of a Pump Turbine in Pump Mode. IOP Publishing) p 42034

[11]. Liu D M, Zhao Y Z, Liu X B, Ma Y and Wang W F 2015 Pump Hump Characteristic Research Based On Mass Transfer Equation IOP Conf. Series: Materials Science and Engineering 72 32016

[12]. Li D Y, Gong R Z, Wang H J, Zhang J, Wei X Z and Shu L F 2016 Numerical Investigation in the Vaned Distributor under Different Guide Vanes Openings of a Pump Turbine in Pump Mode J APPL FLUID MECH 9 253-66

[13]. Li D Y, Gong R Z, Wang H J, Wei X Z, Liu Z S and Qin D Q 2016 Unstable Head-flow Characteristics of pump-turbine under different guide vane openings in pump mode Journal of Drainage and Irrigation Machinery Engineering 1 1-8

[14]. Li D Y, Gong R Z, Wang H J, Wei X Z and Qin D Q 2015 Numerical Investigation On Transient Flow of a High Head Low Specific Speed Pump-Turbine in Pump Mode J RENEW SUSTAIN ENER 7 63111

[15]. Nielsen T K 2015 Simulation model for Francis and Reversible Pump Turbines International Journal of Fluid Machinery and Systems 8 169-82