

PAPER • OPEN ACCESS

Study on hump characteristics of pump turbine with different guide vane exit angles

To cite this article: J Jia *et al* 2019 *IOP Conf. Ser.: Earth Environ. Sci.* **240** 072038

View the [article online](#) for updates and enhancements.

Study on hump characteristics of pump turbine with different guide vane exit angles

J Jia¹, JF Zhang¹, YF Qu¹, HK Cai¹, SB Chen¹

¹Research Center of Fluid Machinery Engineering and Technology, Jiangsu University, Zhenjiang 212013, China

Email: 510084537@qq.com

Abstract. In order to study the influence of different guide vane outlet angles on the external characteristic hump phenomenon under the pump mode of pump turbine with splitter blades, the pump work condition of a pump turbine model pump of a pumped storage generator unit in China was studied. Three different guide vane outlet angle models of 9.6, 17.5 and 25.6 were selected to discuss the variation law of hump characteristics with the guide vane outlet angle under the pump working condition, and the internal flow pattern of the draft tube, impeller and guide vane region was studied. The results show that the calculated external characteristic curves of the three different guide vane exit angles are in good agreement with the test results under the corresponding working conditions. With the increase of the guide vane exit angle, the hump phenomenon becomes more and more obvious and appears earlier, and the smaller the guide vane exit angle is, the higher the head is. There is no wide high efficiency region under the small guide vane exit angle, and the high efficiency region widens as the guide vane exit angle increases. There are secondary flow and backflow in the vane rotating passage, and a large number of whirlpools exist in the guide vane basin. Due to the existence of these bad flow state, the flow loss is greatly increased and the head is reduced, which eventually leads to the hump phenomenon of the external characteristic curve.

1. Introduction

With the development of pumped storage technology, reversible pump-turbine units have been widely used in power systems[1]. Under the pump condition, the H-Q curve of the pump turbine has hump characteristics, which not only affects the start-up and operation of the pump turbine under the pump condition with high lift, but also generates strong vibration and noise, thus leading to unstable operation of the energy storage unit. Therefore, it is of great significance to study the hump characteristics of the pump turbine under the pump condition [2-4].

From the point of view of the literature reviewed, there are still relatively few studies on the hump area of the pump turbine in foreign countries[5]. The existing researches mainly focus on the static interaction, pressure pulsation, and flow loss of the pump-turbine[6]. YANG[7] studied the hump characteristics of multi-stage pump turbines through experiments, and observed the transient flow distribution of diffuser cascades in the hump region. Wang leqin et al.[8] simulated and analyzed the characteristics of pump turbine operating conditions, and pointed out that delaying or reducing the occurrence of backflow can improve the performance of the unit hump. Ran Hongjuan et al.[9] used numerical simulation to study the hump characteristics of the pump turbine operating conditions and found that the complex separated flow within the flow channel is closely related to the formation of



the hump. LI et al.[10] used a very definite numerical simulation method to obtain hump characteristics related to the vortex structure of a double row cascade. Shu lingfeng[11-12] studied the hump area with numerical simulation method and analyzed the hydraulic loss and flow characteristics of different parts with different openings in detail. Braun et al. simulated and tested the hump characteristics of pump turbine and found that the hump characteristics are closely related to the secondary flow in the diffusion channel.

This paper conducts simulation and experimental research on the working conditions of a reversible pump-turbine pump with splitter blades in a high-head pumped storage power station in China. It analyzes the variation laws of the external characteristics and hump characteristics of the pump-turbine under the three working conditions of small guide vane outlet angle of 9.6° optimal guide vane outlet angle of 17.5° and large guide vane outlet angle of 25.6° , analyzes its unstable operation mechanism, and reveals the relationship between the internal and external unstable characteristics, with a view to provide theoretical basis and basis for designing a stable, low-noise high-head and low-specific speed pump-turbine.

2. Computational model and grid division

Pump turbine model

The research object of this paper is a reversible pump turbine, when it is running under the pump condition, water flows through the draft tube, runner, guide vane, stay vane and volute in turn from the inlet, and flows out from the volute, the optimal outlet angle of guide vane is 17.5° . The model ratio is listed as $s=7.4733$, runner inlet diameter $D_1=0.3\text{m}$, outlet diameter $D_2=0.584\text{m}$, outlet width $B_2=0.055\text{m}$, speed $n=1100\text{r/min}$, number of blades $z=5+5$ (5 long and 5 short), number of guide vane blades and stay vane blades $Z_D=16$. The three-dimensional model of the full flow passage of the pump turbine is completed in Proe, its full-flow three-dimensional modeling is shown in Figure 1.

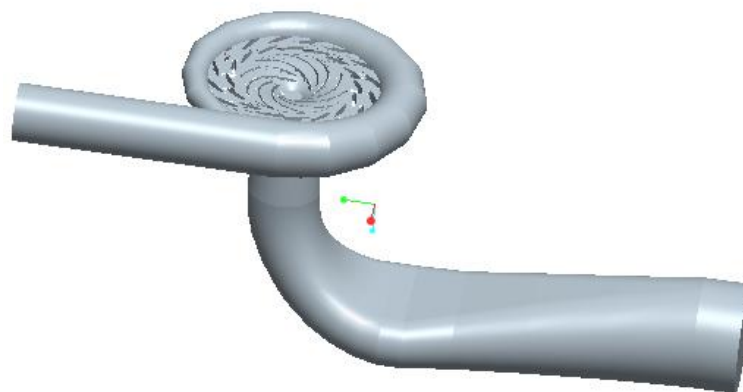


Figure 1. Water body model of pump turbine.

Grid division

Using ANSYS ICEM software to mesh the model, taking into account the model is large and the complexity of the structure, taking into account the calculation time, using a strong adaptability of the unstructured grid for each computational domain mesh[13]. Due to the large influence of grid quality on the flow of the tongue of the volute, the part of the volute tongue is partially encrypted. At the same time, in order to ensure the quality of the grid and the accuracy of the calculation, the surface of the blade, the wall of the impeller and the interface between the impeller and the volute are encrypted, as shown in fig. 2, the quality of the grid is above 0.3. Finally, the number of grids in each calculation domain is 550,000 for the draft tube, 1280000 for the impeller, 940,000 for the double-row cascade and 1950000 for the volute, as shown in fig. 2.

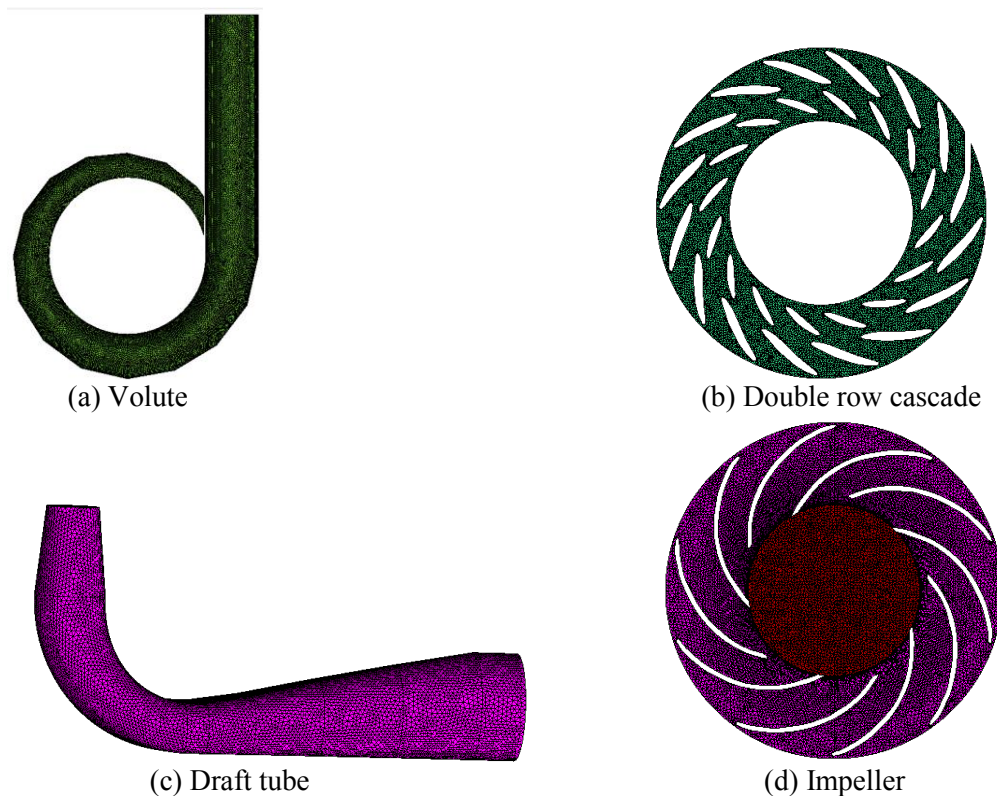


Figure 2. Grid of each computing domain.

3. Numerical computation method

3.1 Turbulence model selection

Since the RNG $k-\epsilon$ model corrects the turbulent viscosity, the rotation and swirling flow conditions in the average flow are considered, and the time-averaged equivalence rate E_{ij} reflecting the main flow is increased in the ϵ equation, which is higher than the standard $k-\epsilon$ model. To better deal with complex flow with large streamline bending and high strain rate and low requirements on the boundary layer of the wall, considering the calculation ability, the turbulent model is selected to use the RNG $k-\epsilon$ model with better adaptability[14-15], and use the finite volume method to spatially discretize the second-order accuracy of the control equations. The time-discrete is in the fully implicit form. The boundary conditions are pressure inlet and mass flow outlet. The solid wall surface adopts the no-slip boundary condition, and the near wall area adopts the Scalable wall function[16-17]. The interface between impeller and draft tube and moving vane is set as the frozen rotor model.

3.2 Calculation method

CFX software is used to calculate the three-dimensional full flow field of the model pump turbine, changing the outlet angles of the movable guide vanes to 9.6° , 17.5° , (optimal outlet angle of the guide vanes) and 25.6° , respectively to carry out multi-working condition steady simulation calculation, selecting a total of 14 flow points at 69L/s, 138.1 L/s, 207.1 L/s, 241.6 L/s, 276.2 L/s, 310.7 L/s, 345.2 L/s, 379.7 L/s, 414.2 L/s, 448.8 L/s, 483.3 L/s, 552.32 L/s, 621.36 L/s, 690.4 L/s to carry out calculation to obtain the relationship between the head, efficiency and power and the flow change, and comparing and analyzing the internal flow characteristics of the highest point in the hump area under the three different outlet angles of the guide vanes.

4. Analysis of calculation results

4.1 external characteristic curve

Figure.3 and figure.4 show the head characteristic curve and efficiency characteristic curve of the simulation and test under the outlet angles of the three guide vanes respectively. It can be seen from the figure that there is a certain deviation between the simulation value and the test value, but the change trend of the external head curve is relatively consistent, the simulation value is higher than the test value, and the trend of the head under the condition of large flow rate is good, the maximum head error of the two is 4.9%, and the maximum efficiency error is 3.3%. this is mainly due to the simplification of the three-dimensional model of water body for flow field analysis, ignoring the friction loss between the bearing and the impeller. Generally speaking, the change trend of simulation and test results is basically the same, which shows that the accuracy of numerical calculation results of flow field is relatively high, it is feasible to use this simulation to qualitatively analyze the flow pattern in the full flow passage of pump turbine.

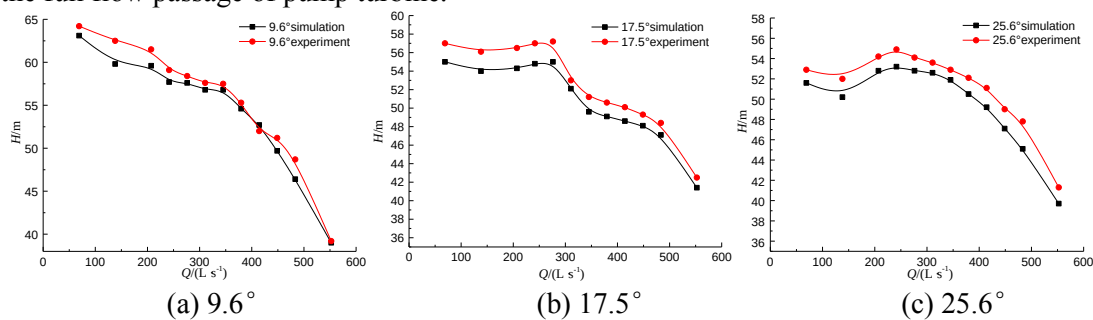


Figure 3. Comparison of head characteristic curves under different guide vane exit angles

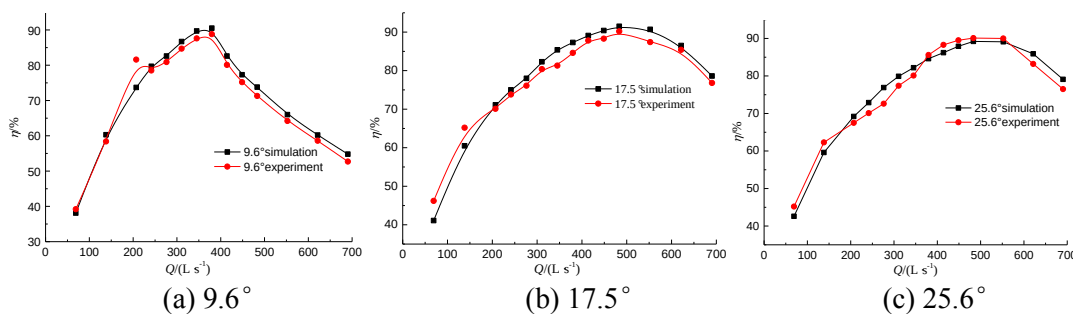


Figure 4. Comparison of efficiency characteristic curves under different guide vane exit angles.

Figure.5 shows the simulated external characteristic curves under different guide vane exit angles, it can be seen that the lift curves under large guide vane exit angles and optimal guide vane exit angles both have a certain degree of hump phenomenon, that is, the H-Q curves have positive slope regions, and the characteristic curves under different guide vane exit angles have different characteristics:

(1) Large guide vane exit angle of 25.6° : hump phenomenon exists in the head characteristic curve, in which flow $Q_d=483.3L/s$ in the design working condition enters the hump area when the pump turbine is running at $138.1L/s$ under the small flow condition and leaves the hump area when the flow rate is $241.6L/s$, the head tends to decrease gradually with the increase of the flow rate.

(2) Optimal guide vane exit angle of 17.5° : the head characteristic curve has hump phenomenon, in which the design flow rate is the same as that of the flow rate entering the hump area and 25.6° , but the hump area is wider than 25.6° , and leaves the hump area when the flow rate is $276.2L/s$, and the head characteristic curve shows obvious irregularity when the flow rate is greater than $276.2L/s$, which reflects the complexity of the internal flow field under the guide vane exit angle.

(3) Large guide vane exit angle of 9.6° : The head characteristic curve has no hump phenomenon, with the increase of flow, the head has been decreasing. Only when the flow is $241.6L/s$ - $345.2L/s$, the head decreasing trend is relatively slow, and the head curve is almost flat.

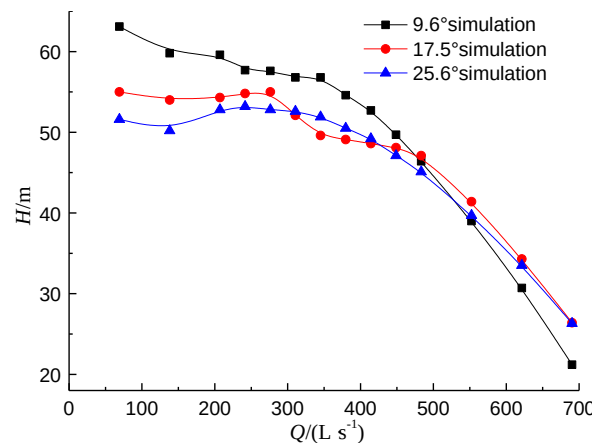


Figure 5. Simulated external characteristic curves under different guide vane exit angles.

Fig.6 is a simulated efficiency characteristic curve under different guide vane exit angles, it can be seen that the efficiency value of small guide vane exit angle is slightly higher than that of large guide vane exit angle and optimal guide vane exit angle in small flow area. The efficiency curves of the optimal guide vane exit angle of 17.5° and the large guide vane exit angle of 25.6° show an obvious parabolic shape with downward opening, and the high efficiency region is wider, while the high efficiency region of the efficiency characteristic curve of the small guide vane exit angle of 9.6° is very narrow.

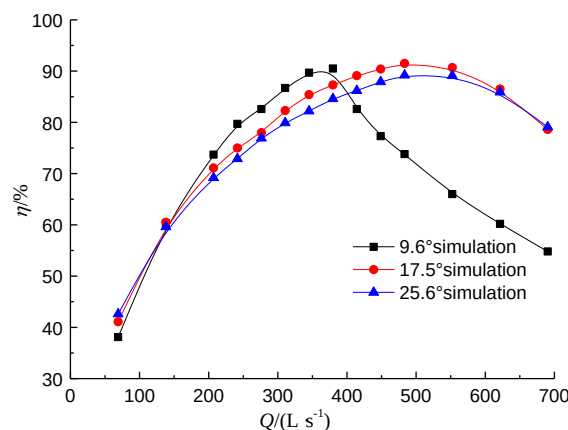


Figure 6. Simulated efficiency characteristic curves at different guide vane exit angles.

4.2 Analysis of internal flow characteristics at the highest point of hump

Figure.6 shows that when the outlet angles of the guide vanes were 9.6° , 17.5° and 25.6° , the peak flow rates in the hump area were 345.2 L/s, 176.2 L/s, and 241.6 L/s respectively.

4.2.1 Draft tube internal flow field. Fig.7 shows the velocity streamline and pressure distribution of the tailpipe meridian plane at different guide vane outlet angles, it can be found that the velocity change corresponds to the pressure change. Due to the rotation of the impeller, the inlet of the impeller forms a low pressure zone, under the action of the pressure difference, water enters from the draft tube in a way of depressurization and acceleration. It can be seen that the internal flow of the draft tube is relatively smooth at the highest point of the hump with three different guide vane openings. There is no spiral backflow, and the water flow smoothly enters the impeller from the draft tube. Comparing the static pressure distribution cloud diagram of the meridian surface, the peak of the hump where no backflow occurs in the draft tube, and the pressure cloud distribution is also relatively regular. In the diffusion section of the draft tube, due to the loss of resistance along the passage, the static pressure gradually decreases along the flow direction. When the water reaches the elbow section of the draft tube, due to the centrifugal force, a high pressure zone is formed outside the elbow section and a low

pressure zone is formed inside the elbow section. When the water reaches the draft tube cone section, the pressure gradually decreases along the flow direction.

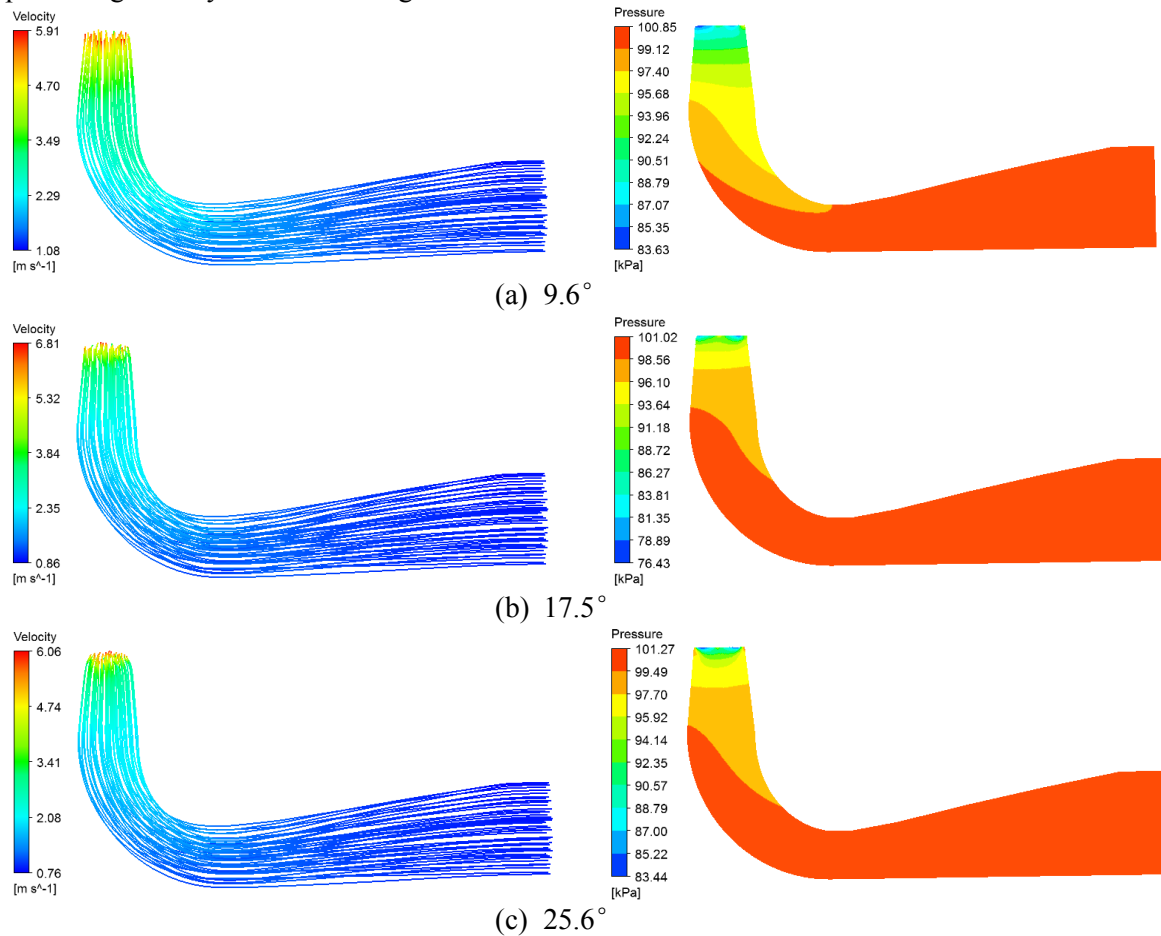


Figure 7. Meridional velocity streamline and pressure distribution of draft tube.

4.2.2 Internal flow field of impeller. Fig.8 shows the meridional plane velocity vector diagram and the blade deployment velocity distribution diagram of the impeller under different guide vane exit angles. It can be seen from the figure that there are backflow phenomena in the impeller basin under the three different guide vane exit angles, and the flow is complicated. Backflow and secondary flow occurred near the lower ring at the impeller inlet. The velocity changes greatly and the flow is unstable. Backflow also occurred near the outlet of the impeller crown. Backflow and secondary flow will affect the main flow and prevent the main flow from passing smoothly, thus increasing the flow loss. In addition, the secondary flow and reflux itself will also dissipate some energy, which will further increase the energy loss in the impeller basin. With the increase of guide vane exit angle, impeller exit velocity increases, backflow and secondary flow areas increase and intensity increases, and impeller basin losses will continue to increase.

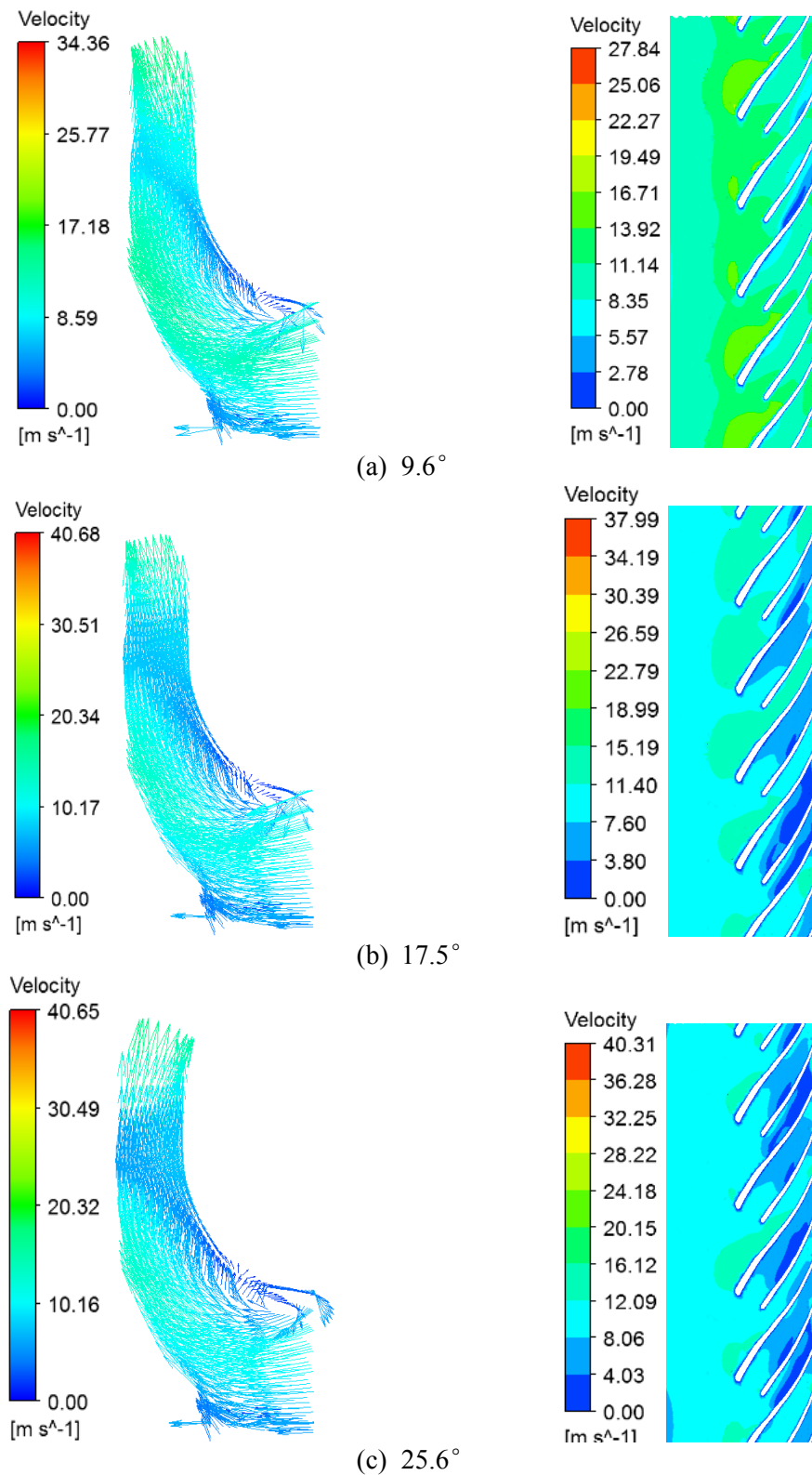


Figure 8. Meridional plane velocity vector diagram of impeller.

4.2.3 Internal flow field of double row cascade. Fig.9 shows the velocity distribution of the movable guide vane and the fixed guide vane along the middle section of the inlet height direction under different guide vane outlet angles. It can be seen that there is a high speed area at the inlet of the

movable guide vane at the highest operating point of the hump, the water flows into the movable guide vane at a faster speed, and the flow velocity has obviously decreased when the water flows into the fixed guide vane. With the increase of the exit angle of the guide vane, the high speed area of the inlet of the movable guide vane becomes smaller. When the exit angle of the movable guide vane is smaller than 9.6° , only a small amount of eddy current is generated in the inner flow passage of the double-row cascade. When the outlet angle of the movable guide vane is 17.5° , backflow occurs at the inlet of the movable guide vane, and the secondary flow and vortex flow in the flow passage are obviously increased and the flow becomes disordered. When the exit angle of the guide vane increases to 25.6° , the high-speed flow area at the entrance of the movable guide vane is already very small, the complicated flow such as backflow and vortex almost fills the entire flow passage of the double-row cascade, and the energy loss is also quite serious.

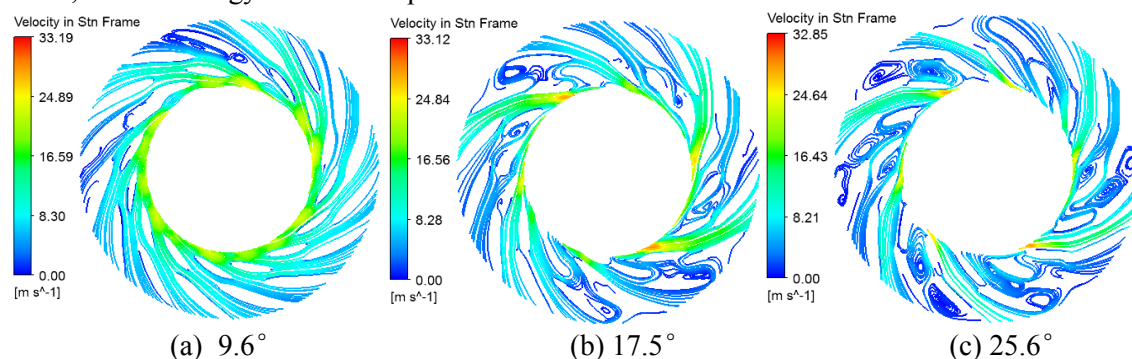


Figure 9. Velocity distribution of double row cascade with different guide vane exit angles.

5. Conclusion

In this paper, the first domestic turbine model with split-blade pump is taken as the research object, and the relationship between unstable flow characteristics, internal flow characteristics and unstable external characteristics under unstable operating conditions in the hump area is analyzed, which provides a theoretical basis for the study of the application of splitter blades in hydraulic machinery. The main conclusions are as follows:

(1) Based on the experimental study and simulation of a pump turbine model pump working condition of a pumped storage unit in China, the external characteristic curves of 9.6° , 17.5° and 25.6° guide vane exit angles were obtained. It was found that the hump phenomenon was the least obvious at 9.6° guide vane exit angles, and it was obvious at 17.5° and 25.6° guide vane exit angles, and occurred at small flow conditions. With the increase of the exit angle of the guide vane, the head of the small flow condition obviously decreases and the hump area narrows.

(2) Under the condition of small flow rate, the exit angle of small guide vane has the highest efficiency, but the high efficiency zone is the narrowest. However, the efficiency of small guide vane exit angle is far less than the optimal exit angle and large guide vane exit angle under the condition of large flow rate.

(3) The highest hump point under the three guide vane exit angles was selected to analyze the internal flow field. It was found that there were backflow and secondary flow near the lower ring at the impeller inlet and secondary flow near the impeller outlet at the upper crown. There are a large number of whirlpools and backflow in the double-row cascade flow passage. The flow condition is bad and the energy loss is serious. Therefore, it is believed that the hump phenomenon is related to the flow separation and vortex in the impeller and double-row cascade flow passages.

References

- [1] Tao L, Zhang Y, Xiaoze D U. A review of rotating stall of pump turbines[J]. Journal of Hydroelectric Engineering, 2015, **34**(11):16-24.
- [2] Liu X Y, Wang L H. Optimization of the Reversible Pump-turbine:A Review[J]. Hydropower

- and New Energy, 2016(6):39-41.
- [3] Tagare D. Electricity Power Generation: The Changing Dimensions[M]. Wiley-IEEE Press, 2011.
- [4] Yuan S Q, Fang Y J, Yuan J P et al. Review on vibration problems of power units in commissioned pumped storage projects in China[J]. Journal of Hydroelectric Power, 2015, **34**(11):1-15.
- [5] Mei Y Z. Pumped-storage power generation technology[M]. Machinery Industry Press, 2000.
- [6] Bueno C, Carta J A. Wind powered pumped hydro storage systems, a means of increasing the penetration of renewable energy in the Canary Islands[J]. British Poultry Science, 2006, **32**(2):353-362.
- [7] Yang J, Pavesi G, Yuan S et al. Experimental Characterization of a Pump-Turbine in Pump Mode at Hump Instability Region[J]. Journal of Fluids Engineering, 2015, **137**(5).
- [8] Wang L Q, Liu J T, Zhang L F et al. Study of pump-turbine's pumping mode at different openings of guide vane[J]. Machinery Industry Press, 2012, **31**(2):222-227.
- [9] Ran H J, Zhang Y, Luo X W et al. Numerical simulation of the positive-slope performance curve of a reverse hydraulic-turn in pumping mode[J]. Machinery Industry Press, 2011, **30**(3):175-179.
- [10] Li D, Wang H, Xiang G et al. Unsteady simulation and analysis for hump characteristics of a pump turbine model[J]. Renewable Energy, 2015, 77:32-42.
- [11] Shu L F. Numerical study of the hump characteristics of a pump-turbine[D]. Harbin Institute of Technology, 2013.
- [12] O. Braun, JeanLouis Kueny, F. Avellan. Numerical Analysis of Flow Phenomena related to the Unstable Energy-Discharge Characteristic of a Pump-Turbine in Pump Mode[J]. 2005.
- [13] Lai X J, Qu Z J, Zhou J et al. 3-D hydrodynamic model for shallow water on unstructured grids[J]. Advances in Water Science, 2006, **17**(5):693-699.
- [14] Gong Y, Wang Y S. The revision for the k- ϵ turbulence model and its application effect[J]. Journal of Engineering Thermophysics, 1989, **V10**(1):114-116.
- [15] Ren J, Zhang Y, Shen Z F. Numerical simulation of off-design condition of vane pump based on modified RNG k- ϵ model[J]. Ship science and technology, 2014(10):101-105.
- [16] Fan W C, Li K. A new wall function[J]. Journal of Engineering Thermophysics, 1989, **V10**(1):114-116.
- [17] Lei Y B, Yuan Y X, Zhao J X. Application of Wall Function in Computation of Complex Flow Field[J]. Journal of Ballistics, 2003, **15**(2):1-5.