

Performance prediction and flow analysis of pump operation of tidal unit with bilateral vanes

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Abstract. Tubular reversible pump-turbine is a special machine used in tidal power, tidal fluctuation by affecting its working head was positive and negative periodic alternation, so the operating conditions of turbine power generation not only, also includes forward and reverse pump operation. A lot of researches have been done on the turbine operation at home and abroad, and a relatively mature bi-directional “s” airfoil blade has been designed, and good power generation performance has been obtained in both forward and backward directions by adding bilateral guide vanes. However, the study on the conditions of pump operation has proved dissatisfactory. In this paper, the steady numerical simulation of a water pump-turbine with bilateral guide vanes is carried out, and the internal flow of the pump operation is analyzed. The results show that by adjusting the front and rear of guide vane and blade angle combination relationship, forward and reverse operation of the pump efficiency can be as high as 83.83% and 58.07%. The experimental research of increasing the unit pump condition is of great significance to the development of large tide unit.

1. Introduction

China has a vast territory, a long coastline and abundant tidal energy resources. With the gradual depletion of fossil fuels, the research and exploitation of tidal energy is developing rapidly, and the scale is becoming larger [1]. Tidal power generation is the main form of tidal energy utilization, which divided into single reservoir one-way, double reservoir one-way, single reservoir two-way and so on. The single reservoir two-way power generation can use the both time of ebb tide and egre tide, the generation time is longer than the one-way generation time, and the generation capacity is more than one-way 15%~20%, so it became the development mode adopted by most tide power stations at home and abroad [2].

Tubular reversible pump turbine is a turbine used in tidal power station (single reservoir two-way development mode), its main way of working is as follow: The water level of the sea is higher than that of the bay when the tide rises, and the unit applies a "sea-bay" direction to generate electricity with the forward corotation operating mode; when the tide ebbed, the water level of the sea is lower than that of the bay , then the direction changed into “bay-sea and the unit operating mode also became reverse contra-rotation ; when there is excess power in the power system, the unit can operate as a pump, pumping water from the bay into the sea, lowering the water level of the bay so that there is a higher head at the next tide egre, and the unit operating mode is reverse corotation pumping; if the tide egre time is inconsistent with the peak load time, the sea water can be pumped into the bay during the



load trough to be used for the next "bay-sea" power generation, there the unit operating mode is forward contra-rotation pumping [3]. In 2005 the State Ministry of science and technology of new tidal turbine development project included in the National 863 project, and used the pit and channel existing Jiangxia tidal power station test to install a new six mode of turbine generator unit [4], significant breakthroughs have been made in turbine runner performance, generator performance, automatic monitoring, new materials and new structures, etc.

A lot of researches have been done on the turbine operation at home and abroad, and a relatively mature bi-directional "s" airfoil blade has been designed [5], and good power generation performance has been obtained in both forward and backward directions by adding bilateral guide vanes[6]. However, the study on the conditions of pump operation has proved dissatisfactory. This paper relies on the CFD flow theory completed the unsteady numerical simulation of pump turbine with bilateral guide blade, discussed the optimal matching relation between the front and rear guide vanes and the blade angle of the forward and reverse pump conditions [7], and the optimum conditions of internal flow is analyzed in detail.

2. CFD numerical simulation

2.1. Unit model

Figure 1 is the channel model of the cross flow reversible pump turbine, the design idea of similar structure of bidirectional axial flow pump [8], are arranged on both sides of guide vanes, front guide vane is curvature airfoil, and rear guide vane is symmetrical airfoil.

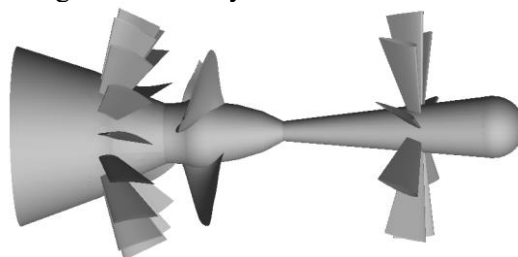


Figure 1. Flow model of bilateral guide vane unit

The specific design parameters of the unit model are shown in table 1.

Table 1. Unit model parameters

Element	Value
diameter of runner(m)	0.34
number of blades	3
Hub ratio	0.38
Test head(m)	7.73
Number of front guide vanes	16
Number of rear guide vanes	9

Similar to the flow direction of the turbine generator, the pump condition can be divided into forward pump condition(bay-sea) and reverse condition(sea-bay), the flow direction of the forward pump is guided by the front guide vane and flows through the impeller and is discharged by the rear guide vane; the flow direction of the reverse pump is that flows in from the rear guide vane, then flows through the impeller and flows out from the front guide vane. In order to reduce the computing time and the burden of computer hardware, in this paper we use a single flow path to calculate. The front

guide vanes, the runner and the rear guide vanes are chosen as the computational domains, and the mesh number of each computational domain is verified to be about 500,000 by grid independence.

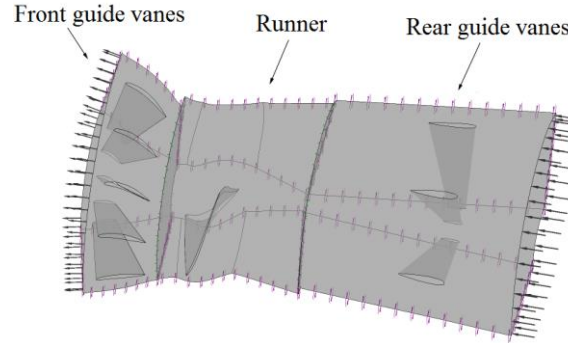


Figure 2. General view of computational domain

2.2. Numerical simulation method

The flow in the unit is three-dimensional incompressible viscous fluid. The fluid flow follows the law of conservation of mass and the law of conservation of momentum. According to the two basic conservation laws and introducing reasonable hypothesis, a governing equation describing fluid flow can be established. In order to solve the governing equations, it is necessary to give reasonable physical parameters and boundary conditions, and the appropriate numerical solution is used to solve and analyze the different flow problems. The governing equations are simultaneous in the form of partial differential, and they need to be discretized in the solution. The approximate solution can be obtained by discretization.

In this paper, the finite volume method is used to discretize the governing equations, and the standard k-ε model [9], which is widely used, is chosen. The model is put forward by LAUNDER in 1972. The transport equation is as follows:

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + \mu_t \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_j} - \rho \varepsilon \quad (1)$$

$$\frac{\partial \partial(\rho \varepsilon)}{\partial t} + \frac{\partial(\rho \varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + \frac{C_{1\delta} \varepsilon}{k} e^{i\theta} \mu_t \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_j} - C_{2\delta} \rho \frac{\varepsilon^2}{k} \quad (2)$$

Turbulent viscosity μ_t can be represented by k and ε :

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon},$$

σ_k is the Pandtl number of turbulent kinetic energy k , σ_ε is the Pandtl number of the dissipation rate ε , the empirical constants are such values: $C_{1\delta}=1.44$, $C_{2\delta}=1.92$, $C_\mu=1.0$, $\sigma_\delta=1.3$.

2.3. Algorithm and boundary condition setting

High quality tetrahedral structured mesh is used in the computational domain, the pressure term is a two order central difference scheme, velocity, turbulent kinetic energy and turbulent viscosity coefficient using two order upwind difference scheme. Solving method is SIMPLC algorithm, the algebraic equation is solved by sub relaxation iteration. The inlet boundary is given as the mass flow inlet, and the exit boundary condition is set to the pressure outlet. The sliding wall boundary condition is used in the solid wall region, and the standard wall function is used in the near wall region.

3. Performance prediction of pumping condition

The pump turbine can be used as pump and as a water turbine, depending on its working conditions, so the characteristic curve is different from that of the traditional turbine or water pump, it is composed of two parts: the hydraulic turbine operating characteristic curve and the pump operating characteristic curve represented by n_{11} and Q_{11} as parameter variables. When the unit operates under the pump condition, the change of the guide vanes opening has little influence on the flow rate, therefore, the adjustment of the working conditions is more likely to depend on the blade angle. So to first determine an optimal efficiency of the front and rear guide vanes opening conditions, then calculate different blade angle conditions, finally draw the characteristic curve of pump model.

3.1. Forward pump condition

The flow direction of the forward water pump is guided by the front guide vane and flows through the runner wheel and is discharged by the rear guide vane. In determining the optimal guide vane opening conditions, we need to fix a suitable blade angle and then through the calculation of multiple conditions, at the same time compare the front and rear sides of the guide vane opening effect on efficiency. Specific process is as follows: The blade angle is set to 30 degrees and the rear guide vane is set to 60 degrees. The simulation results of design and calculation of different forward guide vanes opening obtained as shown in figure 3, you can see the same in other conditions, the efficiency of the unit with forward guide vanes opening increase showed first increased and then decreased, and the highest in front of guide vane is about 80 degrees when.

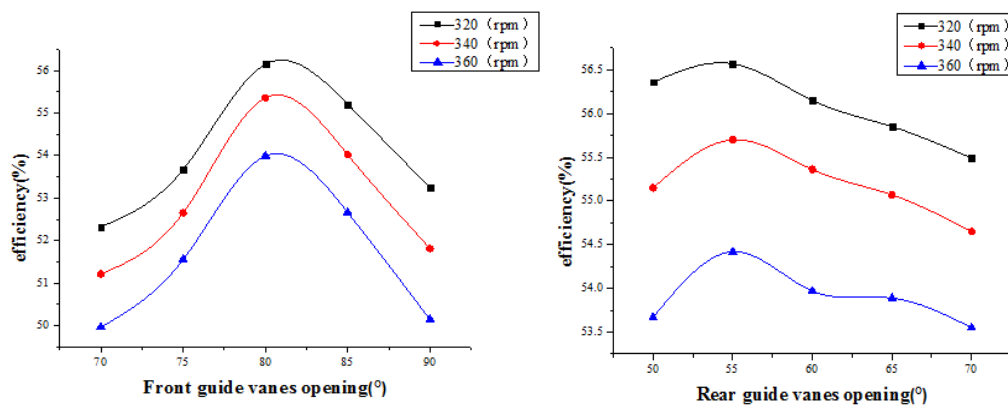


Figure 3. Simulation results of forward pump guide vane opening

We set the blade angle and the front guide vane opening respectively fixed at 30 degrees and 80 degrees, to calculate different rear guide vanes opening, which simulated results shown in figure 3. We can see when other conditions remain unchanged, the efficiency reach the highest with rear guide vanes opening near 55 degrees. At the same time, the two guide vanes opening degrees are fixed, and the 30 working conditions are calculated with the blade angles of 30 degrees, 32 degrees, 34 degrees, 36 degrees, 38 degrees, and the rotating speed of the wheel units being 240rpm, 260rpm, 280rpm, 300rpm, 320rpm and 340rpm respectively, and finally we draw the model synthetic characteristic curve of pump operating conditions such as shown in figure 4:

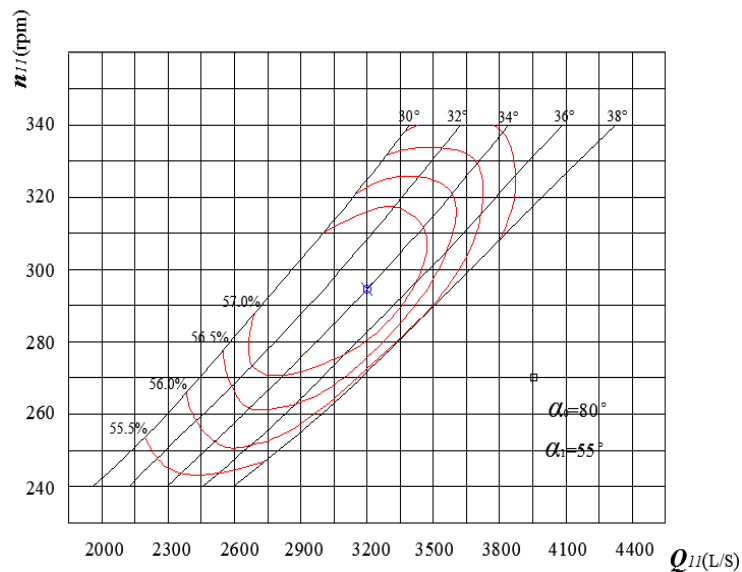


Figure 4. Forward water pump model characteristic curve

From the operating characteristics of forward pump working curve in Figure 4 can be seen, When the blade angle is 34 degrees and the runner speed is 295rpm, the efficiency of the unit is better. By calculation, the flow rate is 3209L/S, and the unit efficiency is 58.07%.

3.2. Reverse pump condition

The flow direction of the reverse water pump is that flows in from the rear guide vanes, and flows through the impeller and flows out from the front guide vanes. The rear guide vane is symmetrical airfoil, so they can be set in the opening of the 90 degree to reduce the impact on the flow, then the blade angle is fixed at 18 degrees, calculation and compare impact of different forward guide vanes opening on efficiency, the simulation results are shown in Figure 5, and ultimately determine when the front guide vane is at 70 degrees, the optimal efficiency of the unit reach highest.

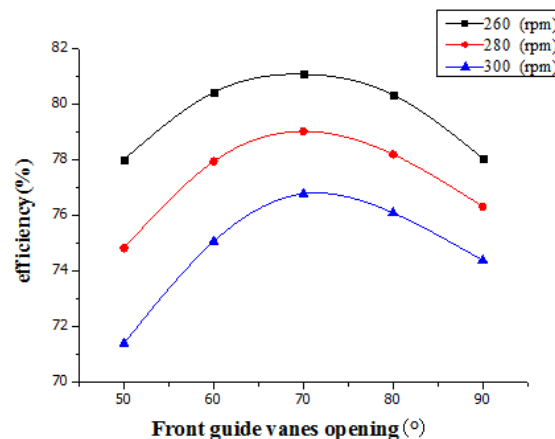


Figure 5. The front guide vane opening - efficiency curve

Similarly, the front guide blade is fixed at 70 degrees, and the rear guide blade is fully open at 90 degrees, then the design and calculation of unit runner speed at 200rpm ~ 260rpm, blade angle from 16 degrees to 24 degrees within the scope of the 35 groups of operating conditions, draw out the reverse pump working model characteristic curve as shown in figure 6, from which we can see the efficiency of the unit is optimal when the blade angle is 22 degrees and the runner unit speed is 220rpm, the unit flow rate is 2028L/S and the unit efficiency is 83.83%.

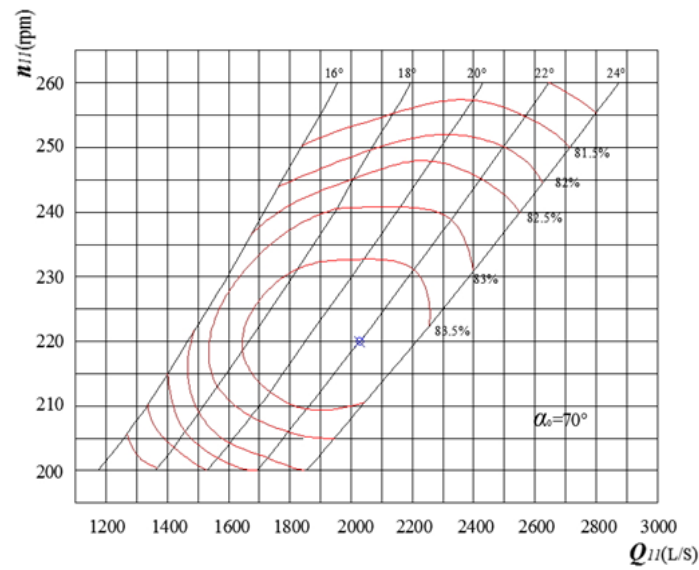


Figure 6. Reverse water pump model characteristic curve

4. Internal flow analysis of optimal working conditions

From the performance prediction of the forward and reverse pump operation, the efficiency of the unit reverse pump is much higher than that of the forward pump. By comparing the streamline distribution under the optimal working conditions (as shown in Figure 7), it can be seen clearly that the overall flow distribution in the reverse pump condition channel is smoother than that in the forward pump condition except for minor flow shedding at the outlet of the front guide vane passage, especially after the flow through the runner, the flow began to become chaotic, which is the main reason for the low efficiency of the unit forward pump.

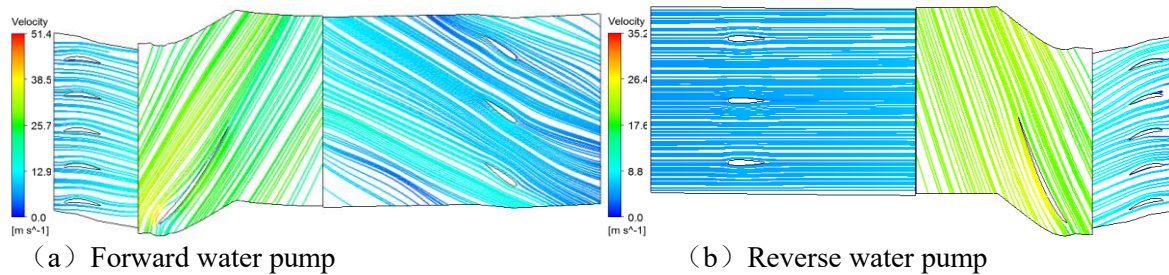


Figure 7. Forward and reverse water pump condition streamline distribution

4.1. Flow analysis of the runner region

In order to further investigate the causes of the flow variation, we select the velocity and pressure distributions along the X axis of the blade pressure surface and the suction surface, as shown in figure 8 and figure 9.

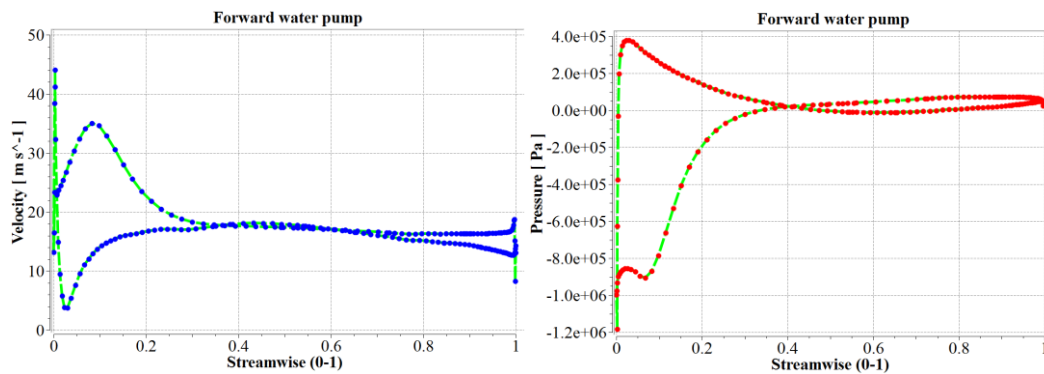


Figure 8. Flow velocity and pressure distribution near blade (forward water pump)

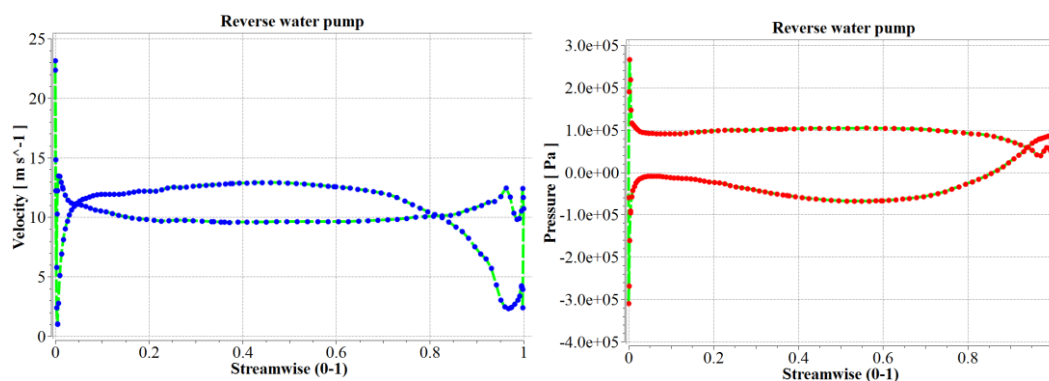


Figure 9. Flow velocity and pressure distribution near blade (reverse water pump)

Through comparison and analysis, it can be seen that under reverse pump condition the flow velocity and pressure distribution near the blade are relatively uniform, and the flow regime is good, however, when the unit is running as a forward water pump, the head of the blade has shown obvious flow velocity and pressure changes, in order to observe the flow situation of the area more concisely and intuitively, we intercept the velocity profile of the blade surface (see figure 10) and the pressure distribution nephogram (see figure 11) under the forward water pump condition.

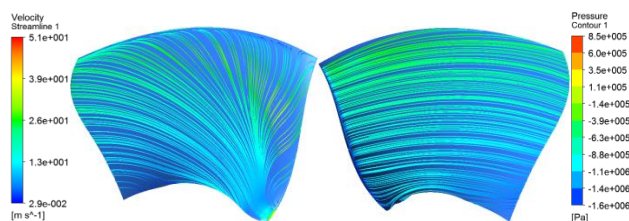


Figure 10. Streamline distribution of forward water pump blade surface

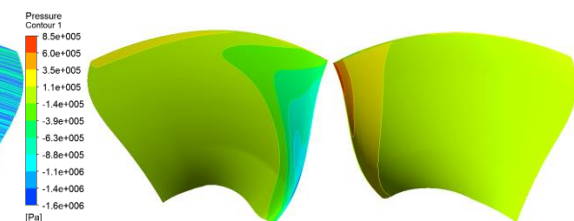


Figure 11. pressure distribution of forward water pump blade surface

From the diagram, it is obvious that the distribution of the streamline at the head of the blade is very uneven, and the head of the pressure head appears slight impact, while the suction surface is a small range of flow shedding and continues to extend backwards. The resulting high-pressure and low-pressure zones are prone to local cavitation and adversely affect the operating life of the blade and even the unit. This may be due to a mismatch between the water inlet angle of the blade head and the direction of the flow, therefore, we can optimize the design of the blade heading and the tail, such as symmetric "s" airfoil blade that is more suitable for bidirectional flow [10], to improve the flow condition and cavitation performance of the unit forward pump.

4.2. Flow analysis of the guide vanes region

Due to the different flow direction, the forward and reverse water pump flows into the rear guide vanes region and the front guide vanes region respectively after passing through the runner, the flow regime of these two regions will also affect the unit efficiency and lift.

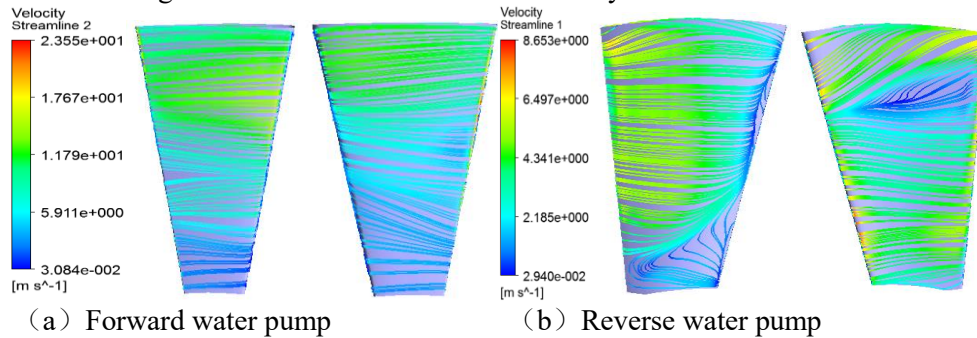


Figure 12. Streamline distribution of the guide vane surface in the forward and reverse pumps

You can see from figure 12, in the forward pump condition, the runner flows not very smoothly, but in the rear guide vane near uniform flow, the flow of the water can be better close to the wall of the guide vane. On the contrary, the flow of the front guide vanes region near the hub flange is not ideal when the unit operates in reverse pump condition, this is because of that the structure near the hub is a hump, and the flow of the water passes through the rotating blades, which makes the flow of the rotor become worse, leading to the instability of the flow near the wheel hub region; in the region near the flange, as the radius of the flow area suddenly increases, the flow of the wall will occur diffusion and separation of flow, and once the impact of the lead vane, it will develop into a local vortex phenomenon.

Another reason may be that the blade of the runner is a distorted type X airfoil blade, water flow angles at different blade heights are different, when the flow of the middle zone coincides with the inlet angle of the guide vane, the water flowing out of the upper and lower wall regions will enter the forward guide vanes region in different flow directions, thus, the head of the front guide vane is impacted and is accompanied with an unstable flow regime, such as a vortex. Here, in the region near the flange, you can see the vortex clearly.

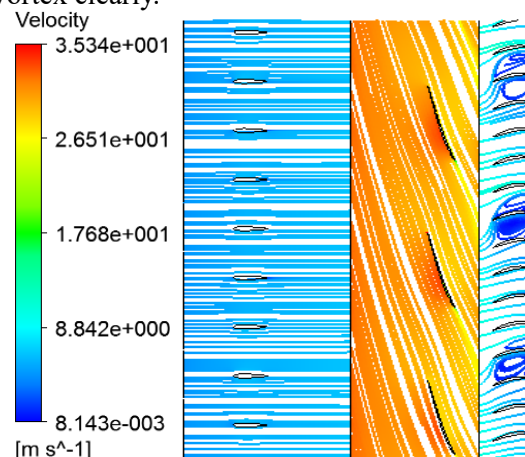


Figure 13. Position of vortex in full flow channel

We calculate a full channel model to further confirm the location of the vortex, the velocity distribution at the height of the guide blade at $0.9H$ is selected as shown in figure 13, we can see that the vortices do not exist in each clearance of the guide vanes, and only a large area of vortex appears behind the clearance of each runner blade, which is probably the result of the wake flow of the blade and the cross influence of the guide vanes. The calculation shows that the head loss of front guide vane

region is 0.39 meters, the head water loss of the front guide vanes region accounts for 5.04% of the unit head, which cannot be ignored in the pump condition of the unit.

5. Conclusion

In this paper, the pump performance of a new type of pump turbine with bilateral guide vanes is evaluated by constant numerical calculation, the internal flow regime is also analyzed. The result shows:

- By adjusting the front and rear guide vane and blade angle combination relationship, forward and reverse pump operating efficiency up to 83.83% and 58.07%, the unit with bilateral guide vanes can show good performance in the aspect of pump operation.
- Due to the defects of the blade structure design, the unit's forward and reverse pump conditions show great differences, especially under the forward pump condition the impact and separation of flow in the blade head become the main cause of the low efficiency of the unit, we can optimize the design of blade airfoil in the future.
- The operation efficiency of the unit reverse pump is high, but the flow in the near wall region of the front guide vanes is not stable, accompanied by partial reflux and vortex phenomena. How to reduce the hydraulic loss caused by the unsteady flow in this area is the direction of the optimization work.

Acknowledgement

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