The internal flow pattern analysis of a tidal power turbine operating on bidirectional generation-pumping

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Abstract. Using tidal energy can reduce environment pollution, save conventional energy and improve energy structure, hence it presents great advantage and is developing potential. Influenced by flood tide and low tide, a fully functional tidal power station needs to experience six operating modes, including bidirectional generation, pumping and sluice; the internal unsteady flow pattern and dynamic characters are very complicated. Based on a bidirectional tidal generator unit, three-dimensional unsteady flows in the flow path were calculated for four typical operating conditions with the pressure pulsation characteristics analyzed. According to the numerical results, the internal flow characteristics in the flow path were discussed. The influence of gravity to the hydraulic performance and flow characteristics were analyzed. The results provide a theoretical analysis method of the hydraulic optimization design of the same type unit as well as a direction for stable operation and optimal scheduling of existing tidal power unit.

1. Introduction

Seventy-one percent of the earth's surface is covered by oceans with rich renewable energy. Tidal current, tide, wave, wave current and ocean temperature difference can all be used for electricity generation. At present, tidal power plants are of the greatest potential [1]. Tidal power is a clean renewable energy with enormous storage yet far from fully exploited. Tidal power plant engross little area on the inshore beach with high operating regulation, stable generating capacity and convenient management [2]. Using tidal energy can reduce environment pollution, save conventional energy and improve energy structure, hence presents great advantage and developing potential [3]. Recently, much attention has been paid to the study on tidal power utilization in many countries with suitable places to build tidal power plants [4-7].

Similar to other types of hydraulic turbines, performances of tidal generation units are determined by inner flow and dynamic characteristics to a large extent. Influenced by flood tide and low tide, fully functional tidal power unit can operate in 6 modes including bidirectional generation, pumping and sluice. Therefore the units have characters of frequent operating condition transition, starting up and closing down, and load adjustment. The inner non-stationary flow and dynamic characters are very complicated. The pressure pulsations in the flow path of tidal power turbine are very important to the stable operation of the unit. Many researches have focused on pressure pulsations in the flow path for Kaplan or Francis turbines with large capacity [8-11], but few studies have been carried out to analyze the pressure pulsations for a tidal bulb tubular turbine.
In this paper, the prototype studied here was a bidirectional tidal bulb tubular turbine with a runner diameter of 2.5 m, rated head of 3 m and rated rotational speed of 125 rpm, the unit has 3 blades and 16 guide vanes. The ratio of rated head to runner diameter ($H/D$) is small comparing with general bulb tubular turbine, so the effect of gravity to the flow is obvious according to IEC regulations\cite{12}. The flow from the reservoir to the sea was defined as forward flow, and flow from the sea to the reservoir was defined as backward flow, as shown in Figure 1. The blade profile is “S” type to achieve high efficiency for forward and backward turbine mode.

![Figure 1. Bidirectional tidal bulb tubular turbine](image)

### 2. Numerical model and boundary conditions

The flow path was shown in Figure 2. The whole path was discretized with an unstructured hybrid mesh of hexahedron and tetrahedron cells. The element size was determined according to Ma and Zhou’s recommendation to satisfy the requirement for the simulation\cite{13}. The flow was modeled using the Reynolds-averaged Navier-Stokes equations with the $k-\omega$-based shear stress transport (SST $k-\omega$) turbulence model. SST $k-\omega$ model is well known and the particular implementation is described in detail by Menter\cite{14}. This model accounts for the transport of the turbulent shear stress and gives highly accurate predictions of the onset and the amount of flow separation with adverse pressure gradients. A second-order Backward Euler scheme was used for the convection terms with a central difference scheme used for the diffusion terms in the momentum equations\cite{15}.

![Figure 2. Flow path of the tubular turbine](image)

![Figure 3. Inlet and outlet boundary conditions](image)
The static pressure conditions were set on the inlet and outlet according to the water level of each operation condition with gravity considered, shown as Figure 3. In the calculations, the runner zone mesh rotated at 125 rpm with the relative positions of each node in this zone unchanged. The interfaces between the guide vanes, the runner and the diffusion tube were modeled with sliding interfaces. For the unsteady calculations, the time step was 0.0048 s, which is 1/100 of the runner rotational period. The data were then recorded at every time step.

3. Numerical results for pressure pulsation

The calculations were performed for four typical operating conditions with parameters listed in Table 1.

| Operating condition    | Net head $H$ (m) | Efficiency (%) | Guide vane angle ($^\circ$) | Blade angle ($^\circ$) |
|------------------------|------------------|----------------|-----------------------------|------------------------|
| Forward turbine        | 3.0              | 81.8           | 70                          | 30                     |
| Backward turbine       | 3.0              | 77.5           | 110                         | 17.5                   |
| Forward pump           | 0.5              | 65.5           | 100                         | 15                     |
| Backward pump          | 0.5              | 60.8           | 80                          | 15                     |

During the calculations, the pressure data were saved at the recording points inside the distributor and the runner. The recording points are shown in Figure 4, and the four points RP1, RP2, RP3 and RS rotate with the runner in the blade passage, the other point GV is fixed with the flow path. The pressure pulsations obtained by simulation were analyzed using fast Fourier transform, the dominant relative frequency ($f/f_n$, where $f_n$ is runner rotational frequency) and the pressure pulsation amplitude at this frequency were then calculated.

As gravity was taken into account, the upper blade suffers low pressure, while the lower blade suffers high pressure, as shown in Figure 5. As a result, the pressure on the blade fluctuates with the rotation of the runner.
The time domain of pressure pulsation for forward turbine mode is shown in Figure 6. The relative pressure pulsation peak to peak value ($\triangle H/H$) on point RP2 is 50%, which is much higher than general bulb tubular turbine with higher value of $H/D$. Figure 7 shows that the pressure pulsation amplitudes increase with the increase of rotational radius for recording points, which also attributes to the gravity effect.

The dominant frequencies of pressure pulsation on GV are all $1fn$ and $3fn$ for four modes, which results from the rotation of the runner, as shown in Figure 8. Otherwise the amplitudes of pressure pulsation on GV for forward modes are much lower than those for backward modes, because the guide vanes locate the upstream of the runner for forward mode and downstream for backward mode. The dominant frequencies of pressure pulsation on RP are all $1fn$ and $2fn$ for four modes. Otherwise the amplitudes of pressure pulsation on RP for forward modes are higher than those for backward modes, as shown in Figure 9. The dominant frequencies of pressure pulsation on RS are all $1fn$ for four modes, the amplitudes of pressure pulsation on RS for forward modes are much lower than those for backward modes, as shown in Figure 10. Figure 9 and Figure 10 show that pressure pulsation amplitude on the
pressure side of the blade (RP for forward turbine, RS for backward turbine) is higher than that on the suction side for turbine mode. While for pump mode, pressure pulsation amplitude on the pressure side of the blade (RS for forward pump, RP for backward pump) is lower than that on the suction side.

**Figure 7.** The frequency domain of pressure pulsation on points RP1, RP2 and RP3

**Figure 8.** The frequency domain of pressure pulsation on point GV for four modes

**Figure 9.** The frequency domain of pressure pulsation on point RP2 for four modes

**Figure 10.** The frequency domain of pressure pulsation on point RS for four modes

### 4. Conclusions

The 3D unsteady flow in a bidirectional tidal power turbine was simulated using SST $k-\omega$ turbulence model with gravity considered. The results show that the upper blade suffers low pressure, while the lower blade suffers high pressure because of gravity influence, and the pressure on the blade fluctuates with the rotation of the runner, the pressure pulsation amplitudes increase with the increase of rotational radius. The relative pressure pulsation peak to peak value on pressure side point RP2 is 50% for forward turbine mode, which is much higher than general bulb tubular turbine with higher value of $H/D$. The dominant frequencies of pressure pulsation on point inside the guide vanes are $1fn$ and $3fn$, which results from the rotation of the runner, but the amplitudes of pressure pulsation on GV for forward modes are much lower than those for backward modes, because the guide vanes locate the upstream of the runner for forward mode and downstream for backward mode. The pressure pulsation amplitude on the pressure side of the blade is higher than that on the suction side for turbine mode. While for pump mode, pressure pulsation amplitude on the pressure side of the blade is lower than that on the suction side.

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References

[1] Zhejiang Electricity Corp., 2001, Jiangxia experimental tidal power plant [M]. Hohai University Press.

[2] Wang, Z. W., Yang, X. S., Xiao, Y.X., 2010, Hydraulic performance optimization of bidirectional tidal power turbine. Journal of drainage and irrigation machinery Engineering, 28(5): 417-421.

[3] Cheng, J. G., 2005, Hydraulic generating design principle for Tidal power station. Shanghai Medium and Lange Electrical Machines, (3):15-18.

[4] Ferreira, R. M., Estefen, S. F, 2009, Alternative concept for tidal power plant with reservoir restrictions. Renewable Energy, 34(4): 1151-1157.

[5] Lee, D. S., Oh, S. H., Yi, J. H., et al., 2010, Experimental investigation on the relationship between sluice caisson shape of tidal power plant and the water discharge capability. Renewable Energy, 35(10): 2243-2256.

[6] Xia, J. Q., Falconer, R. A., Lin, B. L., 2010, Impact of different operating modes for a Severn Barrage on the tidal power and flood inundation in the Severn Estuary, UK. Applied Energy, 87(7): 2374-2391.

[7] Bae, Y. H., Kim, K. O., Choi, B. H., 2010, Lake Sihwa tidal power plant project. Ocean Engineering, 37(5-6): 454-463.

[8] Guedes, A., Kueny, J. L., Ciocan, G. D. and Avellan, F., 2002, Unsteady rotor-stator analysis of a hydraulic pump-turbine-CFD and experimental approach. Proceedings of the 21st IAHR Symposium on Hydraulic Machinery and Systems, Lausanne, Switzerland.

[9] Wang, Z. W., Zhou, L. J. and Huang, Y. F., 2002, The rotor-stator interaction flow simulation on three gorges hydraulic turbines. Proceedings of the 21st IAHR Symposium on Hydraulic Machinery and Systems, Lausanne, Switzerland.

[10] Liu, S. H., Li, S. C. and Wu, Y. L., 2009, Pressure Fluctuation Prediction of a Model Kaplan Turbine by Unsteady Turbulent Flow Simulation. Journal of Fluids Engineering, 131(10): 101102(1-9).

[11] Luo, Y. Y., Wang, Z. Z., Zhang, J., et al., 2013, Vibration and fatigue caused by pressure pulsations originating in the vaneless space for a Kaplan turbine with high head, Engineering Computations, 30(3): 448-463.

[12] Liu, Y. Z., Chang, J. S., 2008, Influence of gravity on flowfield analysis and hydraulic performance evaluation of bulb turbine. SHUILLI XUEBAO, 39(1): 96-102.

[13] Ma, W. S. and Zhou, L. J., 2006, The effect of grid on the result of CFD simulation of turbine. Journal of Hydroelectric Engineering, 25(1): 72-75.

[14] Menter, F. R. and Rumsey, C. L., 1994, Assessment of two-equation models for transonic flows. AIAA Paper, Colorado Springs, CO, pp. 94-2343.

[15] Page, M., Theroux, E. and Trepanier, J. Y., 2004, Unsteady rotor-stator analysis of a Francis turbine, Proceedings of the 22nd IAHR Symposium on Hydraulic Machinery and Systems, Stockholm.