Numerical simulation of tubular turbine applied in wave energy extraction

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Abstract. Overtopping wave energy convertor is becoming increasing popular with the advantages of high efficiency and good reliability. As the core of this type of wave energy extraction device, a shaft tubular turbine with the design head of 15m was optimization designed based on the wave condition somewhere in Shangdong province. Navier-Stokes equations and SIMPLEC algorithm were taken for the numerical simulation of the 3D steady turbulent flow field. By taking consideration of the hydraulic performance of various methods, the best blade airfoil, blade setting angle and guide vane axis were determined. The result shows that the tubular turbine after optimization design not only has the best hydraulic performance with the highest efficiency of 85.36% but also meets the requirement for smooth operation under wave condition. The tubular turbine designed for application in wave energy extraction by numerical simulation is able to supply support for the related research of this field.

1. Wave Introduction
It is paramount to exploit renewable energy to satisfy our society as people are making great efforts to reducing pollution emissions from fossil fuels[1,2]. Ocean has a large amount of energy of more than $2 \times 10^3$TW and contains various forms of renewable energy. Waves are generated by the action of wind and wave energy is the most abundantly available and applicable in offshore areas[3]. Wave energy can be utilized by varieties of technologies and devices, such as oscillating water column device, oscillating body systems and overtopping wave energy convertor. Overtopping wave energy convertor is becoming increasing popular with the advantages of high efficiency and good reliability.

Shi, et al[4] designed two types of turbines applied in wave energy conversion and compared their laboratory performance difference. Margheritini[5] provided a wave energy converter with overtopping type made up of reservoirs and low head turbines. It is expected to produce 320 MWh/y power in the North Sea. Kofoe[6] proposed a wave energy converter called Wave Dragon. It was made up of control systems and hydro-turbines and measurement was conduct for the optimization power take off system optimization. Huang, et al[7] demonstrated that the operation performance of
axial-flow turbine is much better than curved surface vane turbine based on the results of numerical simulation. As the core of overtopping wave energy convertor, little research was investigated for how the loading, power output and efficiency of water turbines are affected by its design parameters.

Double strike and axial flow type turbines are the most two common types which are applied in wave energy conversion. Han [8] designed and optimized an axial flow type turbine and applied it wave energy extraction. Zhang, et al [1] carried research on the axial flow turbine applied in an overtopping wave energy convertor and indicated an experimentally validated CFD model which can be used to optimize turbine performance by improving parameters. However, there was litter investigation on the type of tubular turbine which has the advantage of convenient installation and maintenance, simple structure and good hydraulic performance [9, 10]. Therefore, it is significant to conduct research on the tubular turbine applied in wave energy extraction.

The objective of this paper was to provide a type of tubular turbine which can extract energy from wave energy somewhere in Shangdong province through approach of theoretical analysis and numerical simulation.

2. Wave Energy Extraction System

2.1. Description of the system

![Figure 1. Schematic diagram of wave energy extraction system](image)

1 is the float bowl, 2 is the connecting rod, 3 is the piston, 4 is the supporting structure, 5 is the control system , 6 is the retaining valve, 7 is the reservoir, 8 is the platform of wave energy extraction system.

As indicated in figure 1, wave energy extraction system consists of two main platforms, water lifting platform and wave power generation platform. As for the platform of water lifting, it is made up of the float bowl, the connecting rod, the piston, the supporting structure and the control system. The float bowl can move up and down driven by wave and convert the wave energy into potential energy of the water. This is the first step for energy conversion and water is stored in the reservoir with a high
position. As for the platform of wave power generation, it consists of tubular turbine and hydropower plant. This is the second step when the potential energy of the stored water is converted into useful energy through the tubular turbine when water is on its way to the sea. Turbines play an important role on the power take off of the wave energy extraction system.

2.2. Preliminary design

Table 1. Wave height and wave period of one year somewhere  

| Month | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 |
|-------|---|---|---|---|---|---|---|---|---|----|----|----|
| wave period | 4.7 | 4.8 | 4.5 | 4.4 | 4.1 | 4.5 | 4.7 | 4.9 | 4.8 | 4.9 | 4.8 | 4.9 |
| wave height | 1.7 | 1.2 | 1.2 | 1.1 | 1.3 | 1.3 | 1.5 | 1.4 | 1.7 | 1.7 | 1.5 |

Table 1 shows the wave height and wave period of one year somewhere which is taken as operation condition for the wave energy conversion. The averaged wave height is 1.4m and wave period is 4.7s. The effective inertia-mass of float bowl reaches $5.9 \times 10^4$kg and the bowl can move up and down under the wave condition. So the water potential energy stored by the bowl with the height of 1.4m could convert into the water potential energy stored in the reservoir with the height of 15m and capacity of 5.5m$^3$. So the flow rate passing the turbine is about 1.1m$^3$/s under design condition. Table 2 provides the average water head and flow rate somewhere by water lifting platform. The range of head is from 13.3m to 16.5m.

Table 2. Water head and flow rate of one year somewhere  

| Month | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 |
|-------|---|---|---|---|---|---|---|---|---|----|----|----|
| head  | 16.5 | 13.9 | 13.9 | 13.3 | 14.4 | 14.4 | 14.4 | 15.5 | 15.0 | 16.5 | 16.5 | 15.5 |
| rate  | 1.06 | 0.86 | 0.94 | 0.90 | 1.06 | 0.96 | 1.18 | 0.94 | 0.94 | 1.00 | 1.00 | 0.94 |

Pit type tubular turbine was chosen because of its advantage of convenient installation and maintenance, simple structure and good hydraulic performance.

2.3. Parameter of the turbine

The tubular turbine here consists of inlet passage, guide vanes, runner blades and outlet passage as shown in figure 2. The main working parameters of the turbine were designed with a head of 15m, the flow rate of 1.1m$^3$/s, rotating speed of 850r/min and a diameter of 0.8m.

Figure 2. Whole flow passage of the tubular turbine.
3. Numerical Simulation

3.1. Simulation method

3.1.1. Basic Equations and mesh. For the fluid flow analysis of the entire tubular turbine, the continuity equation and Reynolds-averaged Navier-Stokes equation have been used in the following form [11,12]:

\[
\frac{\partial u_i}{\partial x_i} = 0
\]

(1)

\[
\frac{\partial (U_j U_j)}{\partial x_j} = -\frac{1}{\rho} \frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j} \left[ \nu \left( \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) \right] - \frac{\partial}{\partial x_j} \left( \frac{u_i u_j}{\rho} \right)
\]

(2)

Where \( U \), \( \rho \), \( \nu \) and \( p \) is velocity, density, kinematic viscosity and pressure respectively. The renormalization group (RNG) \( k - \varepsilon \) turbulence model was used for the steady state numerical simulation[13].

The turbine blades were resolved using a body fitted, tetrahedral grid. A sliding mesh interface between the state domain and the rotor enables rotor motion. The mesh in the critical areas was finer such as around the blades. Different mesh strategies were adapted in studying the mesh independence considering the effects of total cell numbers on the turbine performance. Refining the mesh indicated in table 3 has a negligible effect (less than 2%) on the results of turbine efficient, implying that this level of mesh could get a convergent result of the numerical simulation.

Table 3. Mesh of different domains.

| Computational domain | Mesh elements(10^3) | Computational domain | Mesh elements(10^3) |
|----------------------|--------------------|----------------------|--------------------|
| Inlet passage        | 551                | Runner blades        | 514                |
| Guide vanes          | 553                | Outlet passage       | 400                |

3.1.2. Boundary Conditions. At the boundary conditions, pressure inlet and pressure outlet were chosen with no-slip condition for solid boundary and standard wall functions for the region of near wall. There were no buoyant forces taken into consideration and segregated steady implicit solver was used for this simulation. The residual target of convergence criteria was set as 10-4.

3.2. Optimization method

3.2.1. Different airfoil of blades. The blade of the tubular turbine is a space-twisted type. Each section of the blade is rotated around the z axis by a certain angle to change the blade’s twist angle while the shape of blade and the other flow-passing parts keep constant. In order to get the different airfoils of blades, the blade section is warped by the degree of -2°, -1° and +1° so that three types of airfoils could be obtained as proposed in figure 3.

Figure 3. Different section of the blade.
3.2.2. **Different setting angles of blades.** The blade’s setting angle is defined as the angle of the blade bone of the tangent line along the flow direction to streamwise direction. The setting angles of blades were chosen as -4°, -2°, 0°, +2° and +4° for the numerical simulation with the constant angle of guide vane. Figure 4 shows the five different setting angles of blades.

![Figure 4. Different setting angles of Blades.](image_url)

3.2.3. **Different axis positions of guide vane.** The guide vanes play an important role in the turbine performance, different axis positions of guide vanes could affect the turbine efficiency considerably. The guide vane’s axis positions were set as -4°, -2° and +2° while the other flow-passing parts keep constant.

4. Results and discussion

4.1. **Different airfoil of blades**

It can be seen from table 4 that blade airfoil has a significant effect on the turbine performance. The efficiency and flow rate increase when the blade’s distortion angle becomes larger. The turbine shows a best overall performance with the runner blade airfoil (3). The highest efficiency is 84.38% and minimum hydraulic loss is 1.14m.

| Types  | Δh/m  | η/%  | Q/(m³/s-1) |
|--------|-------|------|------------|
| airfoil (1) | 1.170 | 83.26 | 1.06       |
| airfoil (2)  | 1.253 | 83.94 | 1.14       |
| airfoil (3)  | 1.142 | 84.38 | 1.18       |

The pressure difference between the runner blades reflects the power capability. Figure 5 and figure 6 show the pressure distribution on the pressure surface and suction surface of the turbine blades under the same guide vane position.
Figure 5 indicates that the static pressure distributions of the three blades are reasonable and vary smoothly from the inlet edge to the outlet edge. The blades rotate driven by the torque because of its low pressure zone in the root and high pressure zone around the tip. With the increase of blade’s distortion angle, the area of low pressure area decreases and the area of high pressure area expand. The airfoil (3) has a larger high pressure zone and a smaller low pressure zone, indicating that the work capacity of airfoil (3) is better than the airfoil (1) and (2). It can be seen from figure 6 that the area of the suction surface of the airfoil (1) and (2) blade is larger than that of the airfoil (3), which means that the punching angle is larger at the inlet and the flow impact suction surface is formed by vortex and degassing leading to a local hydraulic loss. With the decrease of the area of low pressure and the improvement of cavitation resistance, the blade with airfoil (3) has a better hydraulic performance.

4.2. Different setting angles of blades
Efficiency and energy loss are important indexes to measure turbine performance. Table 5 shows the computational results of different blade setting angles by numerical simulation. The different blade setting angle has an influence on the turbine efficiency and the efficiency improves with the increase of blade setting angle. The turbine reaches its highest efficiency of 85.28\% when the blade’s setting angle is +4°. The head loss is varies a litter bit when the blade’s setting angle is +4° and -4°. The different blade angles have a certain influence on the flow capacity of the turbine. The flow capacity reduces with the increase of blade setting angle, therefore the flow rate is close to the design rate when the blade’s setting angle are +2° and +4°.
Table 5. Results of Blade’s different setting angles.

| Φ/(°) | Δh/m  | η/%  | Q/(m³s⁻¹) |
|-------|-------|------|-----------|
| -4    | 1.092 | 82.25| 1.30      |
| -2    | 1.111 | 82.67| 1.23      |
| 0     | 1.267 | 83.16| 1.17      |
| +2    | 1.242 | 83.58| 1.13      |
| +4    | 1.029 | 85.28| 1.09      |

Figure 7 shows the streamline diagram of full flow passage by different blade setting angles. The streamlines are smoothly with no backflow and local vortex. At the outlet passage, there is radial vortex when the blade’s setting angle is -4°, the radial vortex is improved with the setting angle of 0° and +2°. When the blade’s setting angle is +4°, there is no apparent vortex or radial vortex and the flow pattern is smooth. Therefore, the turbine has the best hydraulic performance with the blade’s setting angle of +4°.

![Streamline Diagram](image)

Figure 7. Streamline diagram of full flow passage.

4.3. Different axis positions of guide vane

The results of different guide vane opening angles by CFD simulation are shown in table 6. Different guide vane opening angles have a certain influence on hydraulic performance of the turbine. When the opening angle of guide vane is set as -2°, the efficiency is highest of η= 85.36% and the hydraulic loss is lowest of Δh =1.153m.

Table 6. Results of different Guide Vane’s Axis Positions.

| Φ/(°) | Δh/m  | η/%  | Q/(m³s⁻¹) |
|-------|-------|------|-----------|
| -4    | 1.216 | 84.36| 1.06      |
| -2    | 1.153 | 85.36| 1.11      |
| 0     | 1.267 | 83.16| 1.17      |
| +2    | 1.279 | 82.98| 1.19      |

Figure 8 and figure 9 show the pressure distribution on the pressure surface and suction surface of guide vanes. According to figure 8, the pressure is getting lower with no mutation from the inlet edge to outlet edge. The pressure increases from the root of the guide leaf to the outer edge of the guide blade. The pressure is the lowest near the root edge. Therefore, the pressure distribution of the guide
vane is more uniform when the opening angle is -4° and -2°. Based on figure 9, the minimum pressure of suction surface is at the root of the guide vane while the pressure at the outer edge of the guide blade is greatest. With the decreasing of the opening angle of guide vane, the flow rate gradually reduces. Under the condition of small rate flow, the opening angle of guide vane has a great influence on the ability of changing the flow direction. When the opening angle of guide vane is -4°, the suction surface of the guide vane appears to cause flow reduction and the negative pressure appears which will lead to cavitation. The results show that the opening angle of guide vane should not be set too large nor small, the hydraulic pressure distribution of the turbine is more reasonable with the guide vane of -2° and the maximum efficiency is 85.36%.

![Figure 8. Pressure distribution of guide vane pressure surface.](image1)

![Figure 9. Pressure distribution of guide vane suction surface.](image2)

5. Conclusions
A shaft tubular turbine applied in wave energy extraction based on the wave condition somewhere in Shangdong province was designed and optimized in this paper. The tubular turbine has a satisfied hydraulic performance with the airfoil twist angle of +2°, blade setting angle of +4° and guide vane opening angle of -2°. The main conclusions are as follows.

- The tubular turbine designed in this paper has a good performance with high power output, efficiency and stable flow pattern which is suitable for the extraction of energy from wave.
- The blade airfoil with the twist angle of +2° has a huger pressure difference between pressure surface and suction surface of the turbine blades, indicating that the optimized blade airfoil could improve the work capacity and performance of the turbine.
- The blades with setting angle of +4° have a higher efficiency and less hydraulic loss, also there is no apparent vortex or radial vortex and the flow pattern is smooth.
- The opening angle of guide vane should not be set too large nor small, the hydraulic pressure distribution of the turbine is more reasonable with the guide vane of -2° and the maximum efficiency is 85.36%.

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