# Analysis of the Hydrodynamic Performance of Torizontal Axis Turbines

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***Abstract:*** *As the most mature trend energy capture device in the current technology, the hydrodynamic performance of the horizontal axis turbine is related to the conversion device of the entire flow energy. Based on CFD simulation software, this paper compares the influence of different TSRs on impeller energy utilization and studies the influence of turbine plus deflector on hydrodynamics. Studies have shown that when TSR increases, energy utilization also gradually increases, and the energy utilization rate of adding a shroud is much greater than that of a bare turbine.*

***Keywords:*** *Tidal power generation;  horizontal axis turbines; Hydrodynamic: Duct*

## 1. Introduction

With the continuous depletion of fossil energy sources, the development and utilisation of renewable energy sources has been the focus of research in the energy sector. As a very promising renewable energy source with a high energy flux density, tidal energy is predictable and has been extensively researched. In the development and utilization of tidal energy, the hydraulic turbine, as a key hydrodynamic component in the development and utilization of tidal energy, has a direct bearing on the hydrodynamic performance of the overall system energy utilization. Horizontal axis turbines are widely used in the development of tidal energy. Therefore, it is of great academic significance and application value to study the influence of on the hydrodynamic performance of tidal turbines. Henriques et al. carried out an experimental study of the performance of a triple horizontal axis impeller under wave coexistence conditions in a circulating water tank at the University of Liverpool, UK. The load and power of the impeller were measured for two different wave parameters and compared with measurements of the impeller under steady flow conditions only to obtain a rule of thumb for the effect of wave induction on impeller performance[1]. Barltrop et al. investigated the effect of waves on the performance of tidal energy impellers. The results show that the average values of the parameters measured over a range of wave periods are no different from the no-wave case.There is no difference between the mean values of the parameters measured during one wave cycle and the no-wave case, but the instantaneous values of drag and torque vary significantly[2]. Luksa Luznik et al. found that waves can have a significant effect on the load and energy produced by tidal wave turbines in a three-blade turbine test[3]. Galloway et al.4 studied the performance effects of a three-bladed horizontal axis turbine in waves and found a maximum increase in thrust fluctuation of 37% and an increase in torque of 35%. On this account, This paper aims at the effect of waves on the hydrodynamics of the horizontal axis[4]. Lee et al[5].used a combination of CFD and experimental methods to investigate the effect of the distance between the twin rotors on the hydrodynamic performance and energy utilisation of a counter-rotating tidal turbine. The results showed that the sliding grid method was more accurate than the MRF multi-reference system model and the actuator disk model.Tian et al[6],used a three-dimensional transient CFD method to study the interaction between waves and turbines and the effects of wave height and rotor submergence depth on the hydrodynamic characteristics of turbines. Rao Xiang et al[7],studied the effect of wave-current interaction on floating turbine arrays and found that the performance of floating tidal energy turbines is more susceptible to wave currents compared to fixed turbines. Wang et al[8],studied the effects of submergence depth, wave height and wave period on the hydrodynamic characteristics of turbines and found that multi-frequency fluctuations based on rotational frequency and incident wave frequency appeared in the axial load factor and energy utilization rate in the instantaneous values, and the amplitude of fluctuation decreases with the increase of blade tip submergence depth; according to the rotation frequency.

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## 2. Basic Theory

### *2.1 Control Equations*

For incompressible fluids, the general conservation equations for continuity and momentum, can be expressed as:

$$\frac{\partial(u_i)}{\partial x_i}=0 \tag{1}$$

$$\frac{\partial}{\partial t}(\rho u_i)+\rho u_i\frac{\partial u_i}{x_j}=-\frac{\partial p}{\partial x_i}+\mu\nabla^2 u_j+f_i \tag{2}$$

where $u_i$ , $u_j$ are the fluid flow velocities; $\rho$ is the density of an incompressible fluid; $p$ is the fluid pressure; $\mu$ is not Compressible fluid dynamic viscosity; $f_i$ is the volume force.

The hydrodynamic performance of HATT is characterized by three dimensionless parameters: power (*Cp*) and axial load coefficient (*Cz*),and Tip-Speed-Ratio (*TSR*). The *Cp* reflects the energy absorption capability of the tidal current turbine and the *Cz* represents the fraction of the axial load experienced in the turbine. The TSR is the ratio between the rotor's linear angular ($\omega_T$) and environmental flow speed:

$$\mathrm{Cp}=\frac{\mathrm{M}\omega_{\mathrm{T}}}{0.5\rho \mathrm{AU}^3} \tag{3}$$

$$\mathrm{Cz}=\frac{\mathrm{F_Z}}{0.5\rho \mathrm{AU}^2} \tag{4}$$

$$\lambda=\frac{\omega \mathrm{T_R}}{\mathrm{U}} \tag{5}$$

where *M* is the rotor torque, $\omega_T$, the angular speed, $\rho$, the fluid density,*A*, the swept area, and *Fz* is the axial load of the rotor.

The airfoil shape is very important for the turbine blade, and the shape of the airfoil is closely related to the hydrodynamic characteristics of the turbine. The lift and drag coefficients of different airfoil shapes are different, and the lift-to-drag ratio varies greatly, which directly affects the hydraulic turbine's. This has a direct impact on the hydraulic turbine performance. The NACA0018 wing was chosen for this study. The main parameters of the blades are shown in Table 1:

*Table 1: Main blade parameters*

| Number of blades | Blade chord length/m | Wheel diameter/m | Blade diameter/m |
|---|---|---|---|
| 3 | 0.0275 | 0.18 | 0.8 |

## 3. CFD Numerical Simulations

### *3.1 Creation of Computational Domains and Meshing*

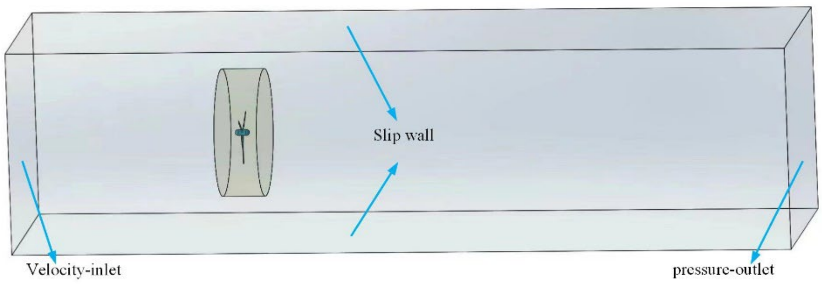


*Figure 1: Computational domain and boundary conditions*

The simulations adopt a three-bladed HATT.The diameter of the turbine is 0.4m, and of the hub is 0.02m. An overset mesh is adopted to simulate the rotation of HATT, using two independent domains: a rectangular background and a cylindrical rotating (around the blades), as can be seen in Fig.

1.Expanding the background domain guarantees development of flow but at more computational cost. Using a cost-benefit analysis, the distance of the HATT rotation plane from the inlet is set to 4D, and from the outlet, to 8D. The inlet is set as velocity inlet, the outlet is set as pressure outlet and the wall is set as non-slip wall. Inlet and outlet turbulence densities are both set at 5%. The grid is divided as shown in Fig.2. The computational domain consists of a stationary domain and a rotational domain. The turbine (including the impeller)is placed in the rotational domain, The shape of the rotating domain is set to cylindrical with a radius of 0.4m and a height of 0.05m, while the rest of the turbine is in the stationary domain. Rotation field in unstructured form, rest of the grid in structured form.

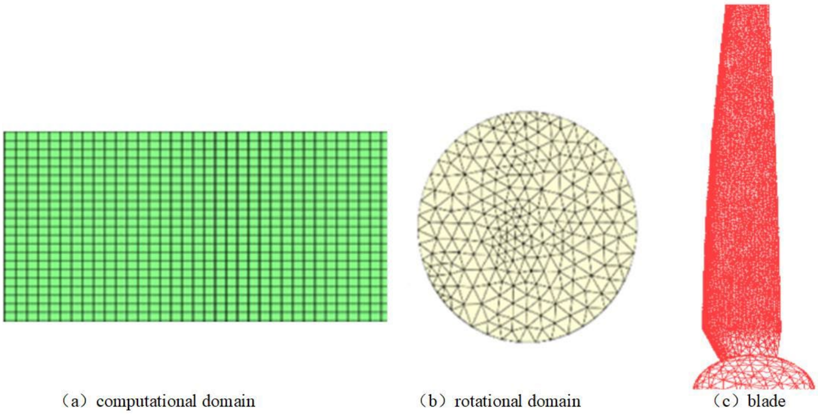


*Figure 2: Schematic diagram of mesh topology*

The external flow field walls are set up as free-slip walls to obtain unbounded uniform flow. The runner wall is set as a fixed wall; The SIMPLEC algorithm is used, with the time term in first-order backward differential implicit format and the pressure term in second-order windward format. The pressure-velocity coupling is based on the SIMPLEC algorithm, with the time term in first-order backward differential implicit format, the pressure term in second-order windward format and the momentum term discretization is in first-order windward format; the turbulence model is SST k-ω model, which has a higher accuracy in rotating flows, and it also takes into account the orthogonal divergence term It also takes into account the orthogonal divergence term so that the equations are suitable for both near and far wall surfaces. The solver uses a transient solver with a time step of 1 degrees of impeller rotation.

### *3.2 Grid-independence Verification*

In numerical calculations, the fineness of the grid division and area division determines the accuracy and speed of the calculation, in order to better investigate the flow field characteristics around the impeller in unbounded flow conditions, the first step is to determine the accuracy of the simulation, the grid division in the unbounded flow field is relatively simple, the grid is divided into three qualities, according to the number of grids into 165W, 225W and 297W (W: million) grid number model. As shown in Fig.3, using the torque coefficient as a reference value.

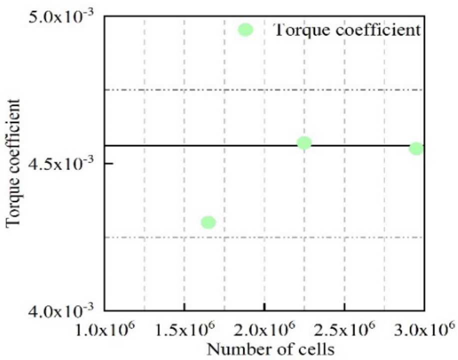


*Figure 3: Grid-independence verification*

As can be seen in Figure 3, when the number of grids is greater than 2.25 million. The torque factor remains constant as the number of grids continues to increase. Torque coefficient value remains at

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0.00456. In this paper. In view of the computational costs and the influence of the number of grids on the hydrodynamics, the later numerical simulations are calculated using a grid of 225w.

## 4. Results and Analysis

### *4.1 Effect of Speed Ratio on Hydrodynamics*

TSR is the ratio of blade rotation speed and incoming flow speed, which is one of the important factors affecting the hydrodynamic performance of hydro turbines. For a turbine with a specific form, structural size, size and airfoil shape, the speed ratio will directly determine the combined speed and angle of attack of the blade at different position angles, thereby affecting the force of the blade and the performance of the turbine.

Numerical simulation with a flow velocity of 2m/s to investigate the effect of velocity ratio on hydrodynamics,As can be seen in Fig.5, the energy utilization and axial load factor of the turbine change for different control speed ratios.and there is an optimum velocity out, which is the velocity ratio corresponding to the maximum energy utilization rate. In a real tidal environment, the tidal flow rate is constantly changing with time. Therefore, in order to keep the impeller in the best working condition, the impeller speed has to be adjusted by the control system so that the impeller speed ratio is stabilised at the optimal speed ratio. The axial load coefficient is related to its control speed ratio. The axial load coefficient curve increases rapidly with the increase of the speed ratio in the lower speed ratio region.

### *4.2 The Effect of Duct on Hydrodynamics*

Design of a venturi-type duct and comparison of the effect of the presence or absence of the duct on the hydrodynamics of the turbine through continuous optimization of the parameters.The shape of the duct and the duct mesh is shown in Fig. 4.

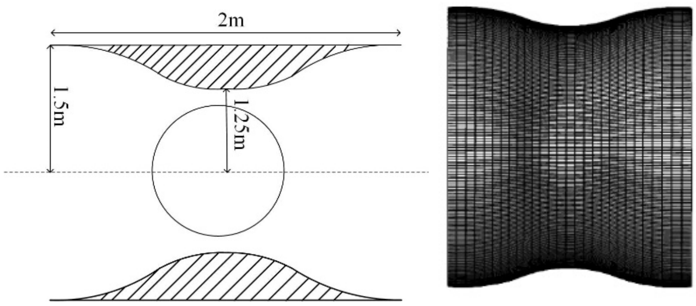


*Figure 4: The shape and mesh of the duct*

The inner wall of the deflector is set in an axisymmetric pattern with an overall length of 1D, the inlet and outlet of the duct are both 0.75D from the axis of rotation and the throat is1.25m from the axis of rotation.

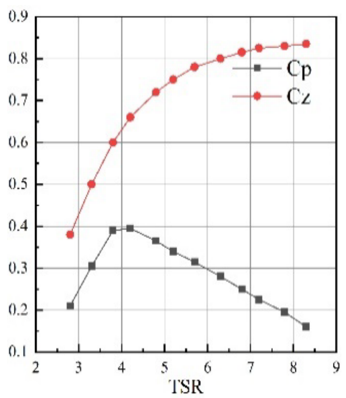


*Figure 5: TSR and Cp, Cz curve*

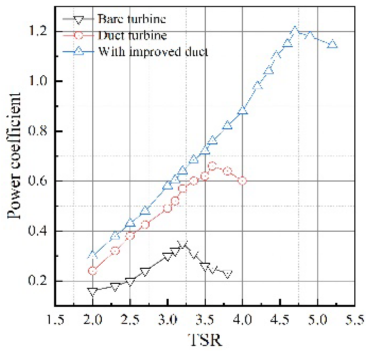


*Figure 6: Cp-TSR curve of duct and unduct*

The Fig.6 shows a significant increase in the power coefficient obtained from the ducted horizontal axis turbine compared to the bare horizontal axis turbine.The bare turbine maximum power coefficient is 3.2 TSR, The maximum operating point in the power output of the horizontal axis turbine with the improved duct, which is equivalent to 4.7 TSR. It is obvious that the addition of a duct can improve the energy efficiency of the turbine.

## 5. Conclusion

This paper presents a numerical study of the hydrodynamics of horizontal axis turbines based on CFD software, and analyses the effects of speed ratios and the addition of deflectors on the energy utilization and thrust coefficients of the turbines. The following conclusions were drawn:

(1) As the speed ratio increases, the energy utilization of the turbine gradually increases, and when it increases to 0.4, Cp starts to gradually decrease when increasing the speed ratio。The axial load on the impeller increases gradually with increasing speed ratio, but the rate of increase decreases with increasing speed ratio.

(2) With the optimization of the turbine's design duct, the optimized duct design achieves a good energy gathering effect, which is caused by the design of the deflector shrinking at the velocity inlet and stretching at the pressure outlet. This design allows for the convergence of energy as the water flows into the inlet of the deflector, releasing pressure at the outlet.

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