Shape design and CFD analysis on a 1MW-class horizontal axis tidal current turbine blade

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Abstract. This study aims to develop a 1MW-class horizontal axis tidal current turbine rotor blade which can be applied near the southwest island regions of South Korea. On the basis of actual tidal current conditions of southern region of Korea, configuration design of 1MW class turbine rotor blade is carried out by BEMT (Blade element momentum theory). The hydrodynamic performance including the lift and drag forces, is conducted with the variation of the angle of attack using an open source code of X-Foil. The purpose of the study is to study the shape of the hydrofoil used and how it affects the performance of the turbine. After a thorough study of many airfoils, a new hydrofoil is developed using the S814 and DU-91-W2- 250 airfoils, which show good performance for rough conditions. A combination of the upper and lower surface of the two hydrofoils is tested. Three dimensional models were developed and the optimized blade geometry is used for CFD (Computational Fluid Dynamics) analysis with hexahedral numerical grids. Power coefficient, pressure coefficient and velocity distributions are investigated according to Tip Speed Ratio by CFD analysis.

1. Introduction
The oceans are an untapped resource, capable of making a major contribution to our future needs. In the search for a non-polluting renewable energy source, there is a push to find an economical way to harness energy from the ocean [1].

In the southern region of Korea, there are more than one thousand islands. The local government plans to develop the region for the local people, boosting the tourism industry and to build facilities for the processing of marine products [2]. The increasing development in this region has caused the need to come up with cleaner energy and minimize the utilization of diesel power plants. By doing this, the region will benefit the environmentally friendly image, contribute to booming tourism and supply efficient and stable power for the local residents for better lifestyle.

Offshore maintenance is already a costly procedure and this, coupled with the fact that tidal arrays are typically located in remote regions at sea in areas of high flow velocity, promotes a need for minimal maintenance over long periods. Optimization of the blade design could increase the amount of energy capture, reduce structural loading and also minimize the need for maintenance [3].

In this study, the shape of the hydrofoil used and how it affects the performance of the turbines are investigated. As the forces acting on tidal turbines are very large due to the relatively high density of
water, the structural requirements for tidal turbines tend to lead to thicker sections [4]. Therefore, after a thorough study of many airfoils, a new hydrofoil is developed using the S814 and DU-91-W2-250 airfoils, which have proven to show good performance in rough conditions.

On the basis of actual tidal current conditions of southern region of Korea, configuration design of 1MW class turbine rotor blade is carried out by BEMT (Blade element momentum theory). Moreover, to examine the hydrodynamic characteristics of the turbine, CFD analysis has been conducted; power coefficient and pressure distribution have also been investigated.

2. Design of 1MW-Class Horizontal Axis Wind Turbine (HATCT) Rotor Blade

2.1. Selection of 1MW-Class Rotor Blade Hydrofoil

To design 1MW-class HATCT rotor blade applicable to Hoenggan Sudo in the south coast, three hydrofoils have been investigated. Figure 1 shows the two selected hydrofoils (S814 and DU91_W_250) and a combination of the two hydrofoils (named MNU26) with figure 2 illustrating their respective aerodynamic characteristics for the designed Reynolds number of 2 million.

S814 is a common hydrofoil used in the tidal current turbine industry; however the DU series has been mostly used in the wind turbine industry for large turbines due to structural qualities. The design objectives of the DU series of airfoils were to keep sensitivity due to leading edge contamination and contour imperfections of the nose as low as possible. The maximum lift capacity was held at moderate levels, to keep the loss of lift due to surface contamination as small as possible [4]. The new hydrofoil (denoted as MNU26) has a 26% thickness that can be applied throughout the blade length, giving good structural strength.

Figure 2 presents the distribution of lift coefficients and lift to drag ratios, which are calculated for the three hydrofoils using an open source code of X-Foil [7]. To estimate the precise hydrodynamic performance of a hydrofoil, the hydrofoil’s hydrodynamic characteristics of the part of the blade should be presented as a lift coefficient and a lift to drag ratio. The lift to drag ratio should be high enough in order to secure an excellent output performance [8]. The selected hydrofoils reach the maximum lift to drag ratio of more than 130 when the angle of attack is between 6~8°, which is relatively very high considering the Reynolds number. It can be inferred from Figure 2 that a rotor blade can be designed with a very good performance.

Figure 1. Selected hydrofoils for rotor blade.

Figure 2. Hydrodynamic characteristics for selected hydrofoils Re = 2000000.
2.2. Design of 1MW-class Rotor Blade

By applying the hydrofoil of MNU26 with a high lift coefficient and a Lift to drag ratio, a 1MW-class rotor blade is designed. Table 1 presents the fundamental design parameters for the blades designed in this study.

| Design Parameters | Values       |
|-------------------|--------------|
| $P_{\text{rated}}$: Rated Power | 1 MW         |
| $C_p$: Estimated power coefficient | 0.46         |
| $V_{\text{rated}}$: Rated current velocity | 2.4 m/s      |
| $\rho$: Sea water density | 1024 kg/m$^3$ |
| $\lambda$: Tip speed ratio | 6            |
| $D$: Diameter | 20 m         |
| $N$: Blade number | 3            |
| $\omega$: Rotational speed | 11.5 min$^{-1}$ |

Table 1. Design parameters for blade design.

Figure 3. Schematic view of optimized rotor blade model: radial length and chord length of the rotor blade model.  

Figure 4. Numerical grid of blade.

With respect to the current speed of the blade design, output power $P$ can be presented as in equation (1). Since $P$ is directly proportional to the cubic root of the inflow current velocity ($V$), the current velocity for the design is set as $V_D=2.4$ m/s, which is higher than the average current velocity, with the consideration of the rotation number of a generator in order to obtain the maximum output at Tip Speed Ratio (TSR) $\lambda =6$ as defined in equation (2).

\[
P = \frac{1}{2} \rho A V_{\infty}^3 \tag{1}
\]

\[
\lambda = \frac{R \omega}{V_{\infty}} \tag{2}
\]

\[
C_p = \frac{P_T}{\frac{1}{2} \rho A V_{\infty}^3} \tag{3}
\]
Here, $\rho = \text{water density}$, $A = \text{swept area}$, and $V_\infty = \text{inflow current velocity}$. The power coefficient is given by equation (3), where $P_T$ is the product of torque and angular velocity by the turbine rotor blade, and the denominator represents the power available that is given by equation (1).

In addition, in this study, Blade Element Momentum Theory (BEMT) [9] is adopted to optimize the design for a rotor blade with the consideration of the tidal current conditions at the installation site.

Figure 3 presents the schematic view of 1MW-class optimized HATCT rotor blade designed in this study. When the blade is designed, the MNU26 hydrofoil is used for the total length of the rotor blade. The blade radius $R$ is set as 10m; the blade number $N$ as 3; and the rotational speed $n$ as 11.5min$^{-1}$.

3. Computational Analysis Method

3.1. Computational Grid

The computational grid of the blade is presented in Figure 4. Considering the computer capacity and calculation time, the total number of grids is about 7 million nodes. Hexahedral structural grids are used to increase the convergence of the calculation and its reliability.

The grid density of the blade surface is higher than that of the flow field surface because an O-Grid was used for the blade surface. A high grid density is required for the blade surface to achieve reliable CFD analysis results. However, for efficient calculation time, the flow field is formed with a lower density with a gradual change in node distance.

3.2. Calculation Conditions

For numerical analysis on the 1MW-class HATCT rotor blade performance and fluid flow, ANSYS CFX [10], a commercial CFD code, is used as a solver in this study. Table 2 presents the calculation conditions used for the numerical analysis of the rotor blade.

For the boundary conditions of the computational flow fields, the average flow rate is set at the inlet, and static pressure at the outlet. As a working fluid, sea water at 25°C is used, and the steady state calculation is conducted. The shear stress transport (SST) turbulence model is utilized, which has been well known to estimate both separation and vortex occurring on the wall of the blade. In terms of a flow analysis method including both the rotational area and the fixed area of the rotor blade in the flow field, the Frozen Rotor Interface method is applied, as a boundary condition to convey flow data to the rotational and fixed areas.

Table 2. Calculation Conditions for 1MW HATCT blade design.

| Condition                              | Values                                      |
|----------------------------------------|---------------------------------------------|
| Inlet current speed                    | 1.2 ~ 4 m/s [TSR: 3 ~10]                    |
| Outlet static pressure                 | 0 Pa                                        |
| Rotational speed                       | 1.2 rad/s [11.5 min$^{-1}$]                 |
| Turbulence model                       | SST                                         |
| Rotational area and rotor blade interface | Frozen rotor                             |
| Surface of blade                       | No slip wall                                |

4. Calculation Results and Discussion

4.1. Power Coefficient

Equation (3) is utilized in plotting the power coefficient curves for the three types of rotor blades investigated. The power coefficient curves are shown in Figure 5.

From the power coefficient curves, a good estimation can be made on the performance of the
turbines. The maximum efficiency of all 3 turbines is about the same at around 46%. However, the DU91-W-250 blade achieves maximum efficiency at a lower TSR compared to the other two turbines. The S814 and MNU26 blade achieves maximum efficiency at TSR 7. This means that at a low current speed the S814 and MNU26 turbines perform better than the DU91-W-250 turbine. Another significant feature observed of the MNU26 turbine is that its efficiency is greater than 40% from TSR range of 5 to 7, meaning it can perform better over a wide range of tidal stream.

4.2. Pressure Distribution

The pressure coefficients of MNU26 turbine are shown in Figure 6. The figure presents the pressure coefficients at each local radius, $r/R$, of the blade radius at TSR 6. The maximum pressure difference between the pressure and suction surfaces is observed at 80% of the local radius. This implies that the maximum power of the blade is generated from about 80% of the blade length.

The occurrence of cavitation can also be estimated from the pressure coefficient curve. The negative pressure coefficient has to be larger than the cavitation number in order for cavitation to occur. The cavitation number is $\sigma = (p_{out} - p_v) / (0.5\rho V_\infty^2) = 2.14$, [11]. From Figure 6, it can be inferred that the cavitation number is greater than the critical pressure coefficient so the MNU26 blade is safe from cavitation.
4.3. Streamline Distribution on the Blade Surface

Figure 7 presents a clear streamline distribution on the blade surface for TSR range 3~9 of the MNU26 turbine. From this figure it can be observed that the lower TSR shows a stronger radial flow formed near the root suction side. Also, TSR 3 shows radial flow formed across more than 75% of the blade starting from the root. This radial flow creates an irregular flow on the blade surface and affects the hydrodynamic characteristics of the turbine by decreasing the output power and power coefficient.

However, for larger TSR (5~7) the radial flow near the hub is quite small and has little effect on the performance of the turbine. In addition, a larger TSR means a low current velocity, which results in low torque on the blade, so as seen in figure 5 that results in lower power efficiency from TSR 8. In order for the turbine to run at excellent efficiency the TSR should be in the range of 5~7. All three rotor blades show similar streamlines so it can be inferred that the optimum TSR range for large HATCT is between 5~7.

5. Conclusions

The following conclusions can be made from this study:

A 1MW HATCT is designed using a new hydrofoil which is a combination of DU91-W2-250 and S814 hydrofoil. The new hydrofoil shows very good hydrodynamic characteristics adopting the merits of the latter hydrofoils at the designed Re of 2,000,000.

The chord lengths and twist angles is optimized to provide a high power efficiency at a wider range of the tip speed ratio, i.e. the efficiency is greater than 40 percent from TSR 5~7, which is a significant advantage to generating more power.

From the pressure coefficient on the blade surface it can be seen that most of the power is generated at 80 percent of the blade length. Comparing the cavitation number and pressure coefficient concludes that cavitation does not occur on the MNU26 hydrofoil.

The streamlines show smoother transition of fluid flow on the blade from TSR 5. A very low TSR means the velocity is quite high and flow is turbulent, which causes flow separation at TSR3.

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