DESIGN PROCEDURE AND PERFORMANCE ESTIMATION OF TIDAL CURRENT POWER SYSTEM

C-H. Jo [1], S-J. Hwang [1], J-H. Lee[1] and K-H. Lee [1]

ABSTRACT: The demand to secure an alternative resource has been increased recently rather than fossil or nuclear powers. One of the affordable ocean energy is the tidal energy since it is considered as a reliable and predictable power source. The turbine is one of the essential components which can convert the tidal current into the rotational force. The design optimization of turbine can contribute significantly to the performance of system. There are several key aspects in the design of turbine. Also the estimation of the device performance is to be carried out by the proven technologies. This paper introduces the design procedure of the tidal current turbine considering number of blades, shape, sectional profile, diameter, etc. Also the performance evaluations of the system with single and multi-arrangements are discussed.

Keywords: Tidal current power, tidal turbine, system performance, design procedure

INTRODUCTION

Ocean energy, especially the tidal current power (TCP), has greater potential and value in use than other alternative energy sources in the world. The turbine can convert tidal energy into rotational power which is delivered to power train and used for generation. Therefore turbine design and performance is one of the important factors which effects on energy efficiency. Many studies have been performed on tidal turbine design based on the blade element momentum theory (Batten et al. [1], Baltazar and Campos [2]). Bahaj et al. [3] conducted an experimental demonstration of tidal turbine performance. Jo et al. [4] compared the performance of three different types of turbines by experiment. Jo et al. [5] investigated the interference effects of multi-arrayed turbines. Faudot and Dahlhaug [6] introduced a performance analysis for tidal turbines under the effects of regular waves.

In this study, a horizontal axis turbine (HAT) was designed based on the related theories and validated using computational fluid dynamics (CFD).

THEORIES

Actuator Disk Theory

In actuator disk theory, flow is assumed to be normal, frictionless, incompressible and one-dimensional. The flow is decompressed while passing an actuator disk which extracts power from the flow. The force acing on actuator disk is calculated by Newton's law of motion. Changes of flow through the actuator disk can be described by momentum conservation law.

Eq. (1) shows the power coefficient of the actuator disk. The 'a' is the axial flow induction factor which is defined as the ratio of downstream velocity to inflow velocity.

$$C_P = 4a(1-a)^2 \tag{1}$$

Rotor Disk Theory

The power extracted from flow is used to rotate the disk. The tangential flow induction factor showed in Eq. (2) is defined as the ratio of inflow velocity at the disk to the tip speed ratio λ and the local span position μ. The λ is the ratio of blade tip speed to inflow velocity. These axial and tangential flow induction factors are important variables to design the turbine blade.

$$a' = \frac{a(1-a)}{\lambda^2 \mu^2} \tag{2}$$

Blade Element Theory

The flow mechanism around a blade element is shown in Fig. 1. The value and direction of a relative velocity Vr are determined by both inflow velocity Vw and rotating speed rω. And the blade pitch angle θ is determined by the incidence angle and the optimum angle of attack. The incidence angle also depends on the span along the blade. Thus, the pitch angle changes continuously along radial positions.

Relative velocity generates the lift and drag forces that can be decomposed into the tangential force and the thrust. The tangential force generates the torque that drives the shaft, and the thrust is the primary force which

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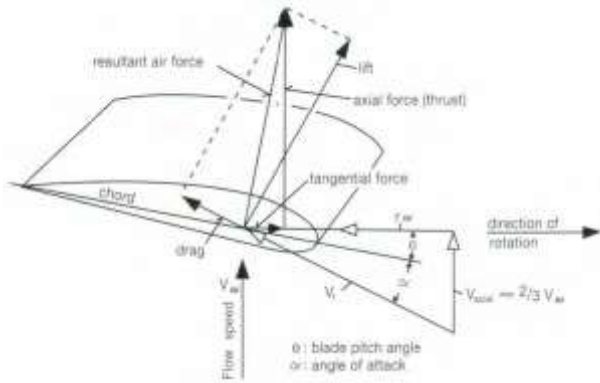induces the bending of the blade. Thus, the thrust should be considered in the structural design.

Fig. 1 Flow mechanism around blade element

BLADE DESIGN

Selection of Foil

The blade design involves a sequence in which specific foils are arranged with appropriate twist angles and chord lengths along the span. The foil should be chosen or determined prior to the design of the blade. To ensure higher efficiency and stable operation, the appropriate foil should be chosen. It should generate high lift with low drag and not be sensitive to surface roughness.

To ensure structural stabilities, a thick foil of S814 was considered, which was applied to the tidal current device of Seagen developed by Marine Current Turbines (MCT). Figure 2 shows the shape of the S814 foil.
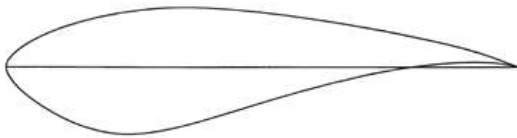

Fig. 2 Profile of the S814 foil

Key Variables

Blade design is initiated by the determination of key variables including rated power, current velocity at the site, diameter, and rated RPM. The maximum velocity in Korea is approximately 6 m/s, and design velocity was assumed to be 5.5 m/s. The remaining design variables can be calculated using Eqs. (3) and (4). P in Eq. (3) represents the power capacity, which can be defined as the product of generated power from the turbine and the efficiency of power train η. The target power capacity is 300 kW in this study, and η was assumed to be 0.85. The D represents the diameter of the turbine, which affects

both the rated power and the rotational speed. The diameter for the target capacity was calculated by assuming the efficiency of the turbine to be 0.4.
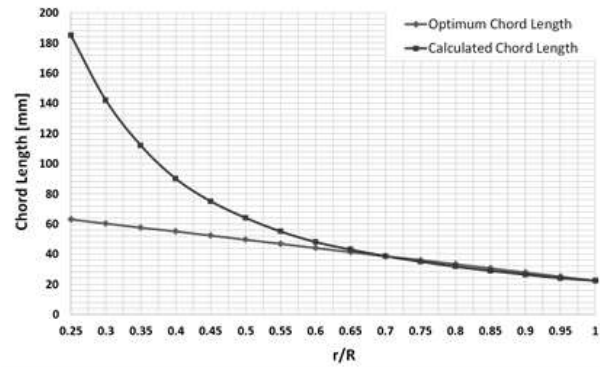

Fig. 3 Distributions of chord length

$$P = \frac{1}{8}\rho\pi D^2 U_d^3 C_P \eta \tag{3}$$

$$TSR = \frac{U_{tip}}{U_\infty} = \frac{D_\omega}{2U_d} \tag{4}$$

Blade Tip Loss

Blade tip loss is caused by tip vortices and decreases the lift force which drives turbine. Eq. (5) suggested by Ludwig Prandtl in 1919 and can be used to estimate the loss.

Initial value of the axial flow induction factor is adopted 1/3 which is applied in the ideal condition. The optimum factor was estimated by iteration procedure. The chord lengths and twist angles at each span were calculated using the optimum induction factor.

$$f_t(\mu) = \frac{2}{\pi}\cos^{-1}\left[e^{-\{(N/2)(1-\mu)/\mu\}\sqrt{1+(\lambda\mu)^2/(1-a)^2}} \right] \tag{5}$$

Chord Length

Calculation

The chord length can be determined by Eq. (6) with lift coefficient C_L, number of blades N, design tip speed ratio (TSR) λ, flow induction factors (a, a′), and the local span position μ.

$$\frac{C}{R} = \frac{2\pi}{N\lambda C_L} \times \frac{4\lambda\mu^2 a'}{\sqrt{(1-a)^2 + \{\lambda\mu(1+a')\}^2}} \tag{6}$$

Interpolation

The coordinate data of the foils at each span is calculated from the design sequences, and Fig. 3 shows the result. The red line is the result which increases too high near the root. Since the region near blade root has little contribution to the performance, calculated chord

length is modified to optimum chord length as the blue line.

Twist Angle

The twist angles at each span should be calculated considering the optimum angle of attack at the design TSR. Twist angle θ, incidence angle φ, and angle of attack α are related as shown in Eq. (7). The incidence angle is calculated using Eq. (8).

$$\theta = \phi - \alpha \tag{7}$$

$$\phi = \arctan\left(\frac{1-a}{\lambda\mu(1+a')}\right) \tag{8}$$

Modeling

Three-dimensional (3-D) modeling should be conducted after calculating design variables for verifying the blade shape and CFD analysis. The curves representing the foils were generated as shown in Fig. 4. From these curves, the blade surface was formed as shown in Fig. 5
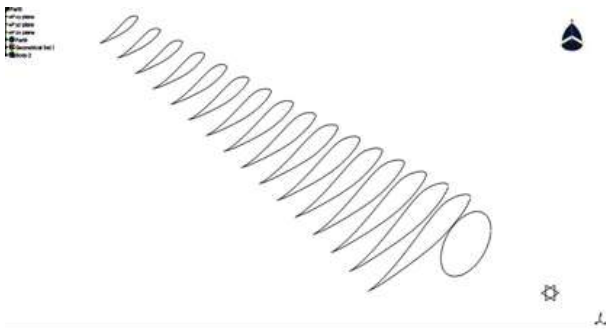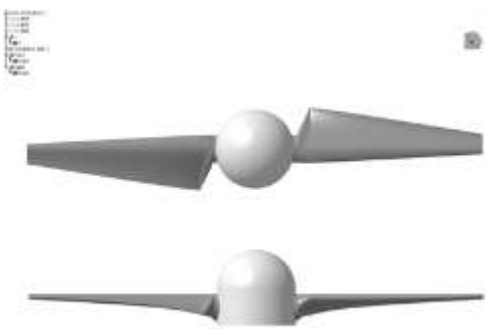

Fig. 4 Framework of the blade

Fig. 5 Solid model of the turbine

PERFORMANCE ANALYSIS

Performance of Turbine

Since the turbine is the device that converts the available power of the flow to rotating energy, its performance can be evaluated by calculating the energy conversion efficiency. The turbine power characteristic is affected by factors including the current speed, turbine size, and rotational speed, but it can be normalized and compared with others effectively as plotting the power curves for TSR. Thus, performance analysis using TSR is widely used. The power coefficient is defined using Eq. (7). The TSR is a dimensionless value representing the rotating speed based on the inflow velocity, as shown in Eq. (8). Thus,

$$C_P = \frac{T\omega}{0.5\rho AU^3} \tag{9}$$

$$\lambda = \frac{R\omega}{U} \tag{10}$$

CFD for turbine performance

Usually CFD analysis is used to verify the turbine performance prior to experiment that may require a high cost. It is possible that steady state analysis can be implemented to estimate performance of the turbine. The flow around the rotating blade can be regarded as being stationary for the time.

The computational domain for CFD analysis was generated in ANSYS CFX-V11, a commercial CFD code, as shown in Fig. 6. The analysis field was composed of two domains. An internal rotating domain encompassed the blades, and a stationary domain covered the remaining area of the flow channel.

In the rotating domain, an angular velocity was prescribed for each case to give a TSR. The turbine performance was analyzed using five cases for inflow velocity (3.0, 4.0, 5.0, 5.5, and 6.0 m/s). A total of 50 analysis cases were prepared and run with various values of TSR from 1 to 10. Table 1 informs the details.
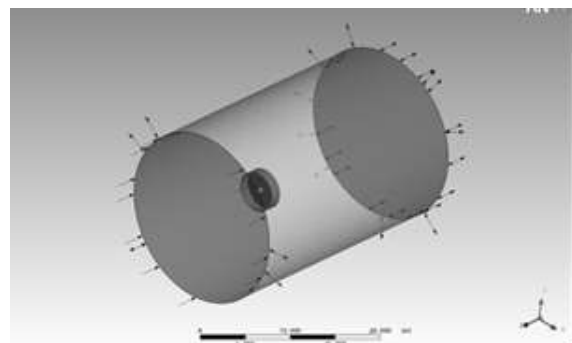

Fig. 6 Computational domain

Grid generation should be carefully conducted for smooth convergence and reliable results. The key areas of interest occurred around the rotating blades and hub for accurate estimation of the torque of the turbine. Thus, a dense tetra-prism mesh was generated in the inner zone, thereby concentrating cells around the turbine.

Table 1 Analysis information

Description	Analysis information
Working fluid	Water (isothermal, 25℃)
Inlet	Normal speed (3 to 6 m/s)
Wall	Opening
Outlet	Opening
Interface area	Frozen rotor
Turbine	Wall (no slip)
Turbulence model	SST

Results

From the results of CFD analysis, the curves of the C_P and the torque driving the generator can be obtained as shown in Fig. 7 and Fig. 8. Since the C_P represents the efficiency of the energy conversion, C_P curves are very important. In this case, the maximum power was generated at TSR 5 for all cases. Thus, the turbine should be operated at or near the TSR 5.
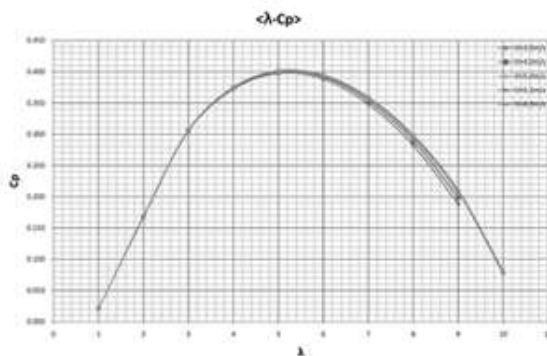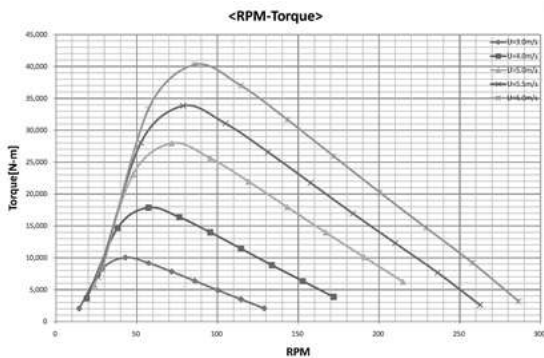

Fig. 7 Cp curves for various current speeds
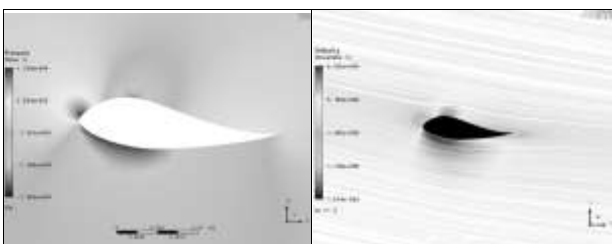

Fig. 8 Power curves for various current speeds

Fig. 9 Pressure distributions and streamlines around the foil at TSR 5

CONCLUSION

The tidal energy is a promising energy source with great potential since it is predictable and continuous regardless of weather conditions and seasons. HAT blade design involves the sequencing and arranging of specific foils with appropriate twist angles and chord lengths; its procedures and methods are presented in this paper. A performance analysis of the turbine is important and an example of verification by CFD is shown. The reliability of the blade design could be estimated using CFD by presenting the streamlines and pressure fields around the blade. The performance and torque curves were determined and the turbine characteristics were analyzed.

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