Performance Analysis and Design of Vertical Axis Tidal Stream Turbine

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Abstract

This study numerically analyses the unsteady flow around the Darrieus-type turbine by using FLUENT and deals with the application to the design of blades. Two kinds of blade sections are used in this study. Unsteady RANS equation and the turbulence model, either k-ε or k-ω model, appropriate for each blade section were employed. First for the NACA634-021 that the experimental data is available, the 2- and 3-dimensional numerical analyses have been performed and compared with the experimental result. For the optimization of the turbine, the parametric study has been performed to check the performance in accordance with the changes in the number of blades, solidity and camber. It was demonstrated that the present approach could draw the turbine characteristics better in performance than the existing turbine. Next for the NACA653-018 blade with the high lift-drag ratio from the purpose of developing highly-efficient turbine, this study has also tried to get the highly efficient turbine specifications by analyzing the performance while using 2-dimensional and 3-dimensional numerical analyses and the result was verified through the experiment. According to the present study, it was concluded that the 3-dimensional numerical analysis has simulated the experimental values relatively well and also, the 2-dimensional analysis can be a useful tool in the parametric study for the turbine design.

Keywords: Tidal Current Energy, Vertical Axis Turbine, CFD, Tip-Speed Ratio, Parametric study

1. Introduction

The tidal stream power generation, as the power generation method that produces electric energy by using the high kinetic energy created by the tidal differences and geographical features, is the method that shows a high operating rate regardless of the weather and seasonal changes. As the turbine blade of converting the motion of fluid into the mechanical torque, the key component of tidal stream power plant can be divided largely into the horizontal axis turbine (HAT) and vertical axis turbine (VAT) according to the arrangement. Although the Darrieus turbine introduced in this study has the disadvantages of rather difficult initial starting and somewhat less efficiency, it not only enables to do various analyses and verifications due to its simple design but also allows designing the optimum turbine while varying the geometric parameter. Also in order to raise the efficiency, it can be easily developed into the helical turbine or controllable pitch blades.

The tidal energy of Korea is very rich and there are many places that make fast tidal current in the regions such as Incheon and the southwest coast of Korea. There are not many countries in the world that the tidal differences occur greatly and Korea is one of the best places to develop the tidal stream power generation plant [1]. There are not many studies published on the tidal stream power generation as compared to the wind power generation and especially, it is difficult to find a systematic numerical study for the performance optimization of vertical axis turbine. Unlike the horizontal axis turbine possible to analyze with the quasi-steady assumption, the vertical axis turbine shows the inherent unsteady flow phenomenon and vortex induced vibration (VIV).

This study numerically analyzes the unsteady flow around the Darrieus Turbine by using FLUENT and deals with the study applicable to the design of blades. For two kinds of turbines, i.e., the NACA634-021 blade of [6] with existing experimental data as well as the NACA653-018 blade with the high lift-drag ratio, this study has analyzed the performance by using the 2-dimensional and 3-dimensional numerical analyses and has verified the result through an experiment.

2. Application to the Design of Darrieus Turbine with NACA 634-021 Blade Section

2.1 Model, Numerical Method and Test Condition

The model selected in this study is the #1 model of [6] that has the NACA 634-021 blade and this blade in a shape of vertical symmetry is frequently used as the

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blade of wind power turbine as shown in Fig. 1. The reason of selecting this model is to compare with the experimental as well as 2-dimensional numerical results of [6] in order to verify the present numerical results. The standard turbine model has 3 blades with the radius of R=0.455m, blade chord of c=0.068m, and blade span of H=0.68m.

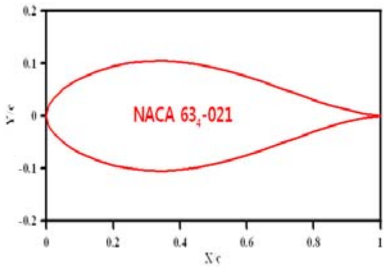
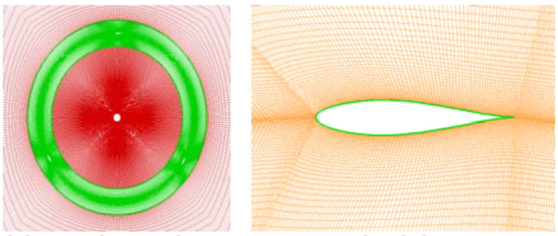


Figure 1: NACA634-021 Blade Section of #1 Model

A commercial CFD code FLUENT was used for the analysis of flow field. This study has used the unsteady incompressible Reynolds-averaged Navier-Stokes equation based on the cell-centered finite volume method and has implemented the rotation flow field by using the sliding mesh technique to rotate the space of turbine area. Prior to this study, the calculation domain has been selected to minimize the effects of finite boundaries by performing a study of optimizing the boundary conditions and calculation domain. Fig. 2 shows the grid system around the blade and rotor. The structured grid has been employed for all the grid system of rotor and the computational domain reaches to 5D in the inlet direction, 15D in the outlet direction and 5D in the vertical direction, where D denotes the turbine. The rotating area of rotor is constructed in a donut-shaped circle covering the blade, giving the velocity inlet condition for the inlet boundary and the pressure outlet condition for the outlet boundary.



(a) Sliding Mesh Around a Rotor (b) Blade

Figure 2: Grid System

While most calculations were made based on the 2-dimensional unsteady flow assumption for its relative simplicity, the 3-dimensionality of finite turbine blades was also checked by performing the 3-dimensional analysis for some typical cases in order to validate the effectiveness of calculations from the excessive calculation capacity. The operational condition of turbine was based on the tip speed ratio (TSR) defined as in Eq. 1.

$$TSR = \frac{R\omega}{V} \tag{1}$$

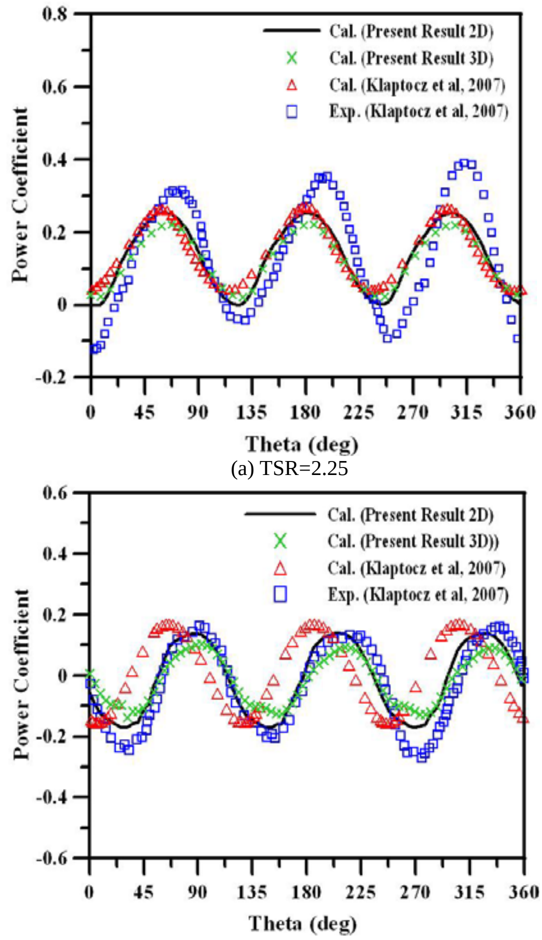
Here, ω=angular velocity, V=current speed. Turbine torque T was normalized as the power coefficient C_p.

$$C_P = \frac{T\omega}{\rho V^3 RH} \tag{2}$$

The numerical calculation was performed for TSR= 2.25 and 3.5, where experimental data were available. Design speed is V=1m/s, and its corresponding Reynolds numbers (Re) based on the blade chord and resultant velocity with the inflow of blade are 1.54×10^5 and 2.38×10^5 respectively. Although the range of Re corresponds to the laminar or transition area, this study assumes turbulent flow same as that of [6] because the free stream turbulence level in CWC is usually high enough to prevent from laminar flow.

2.2 Analysis of Standard Turbine Blade

The calculation result for the turbine blade adopted is shown in Fig. 3 and Table 1 shows the average C_p. We could see that the numerical analysis result shows a tendency similar to the existing numerical calculations and experimental values.



(b) TSR=3.5

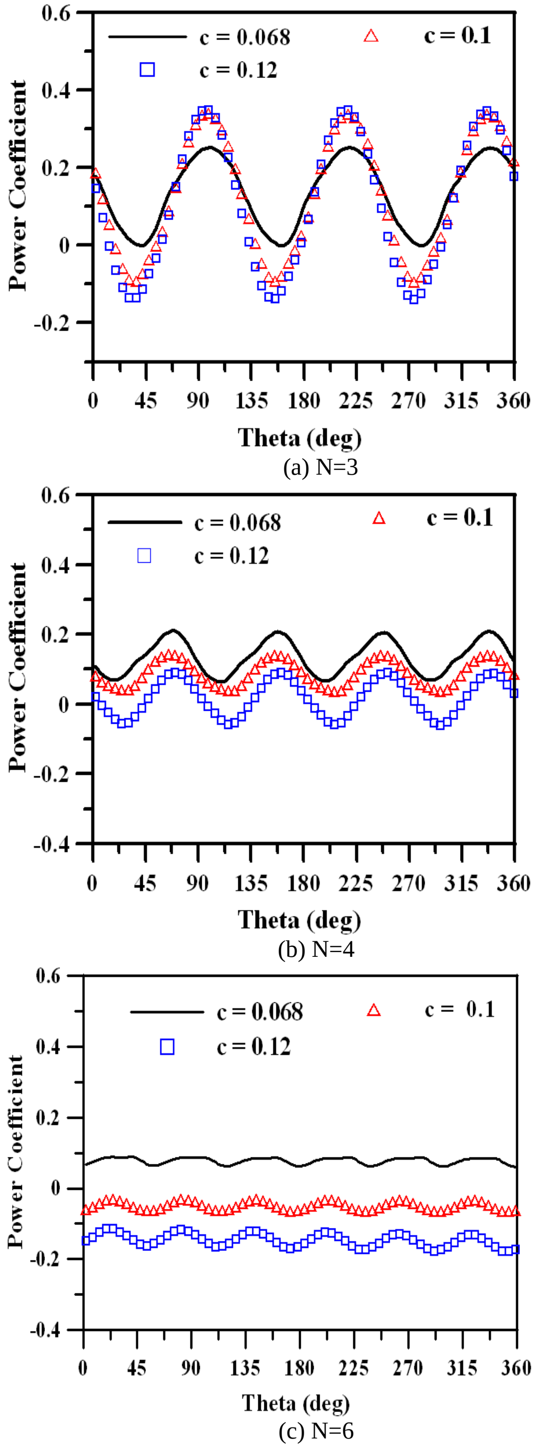
Figure 3: Comparison of Power Coefficients for #1 Model

TSR	Klaptocz et al(2007)		Cal.(Present Result)	
	Exp.	Cal.	2-D	3-D
2.25	0.138	0.149	0.134	0.123
3.5	0.0106	-0.0019	-0.016	-0.016

Table 1: Comparison of Power Coefficient for #1Model

Since the span of this turbine is relatively small, the 3-dimensional result is estimated to be about 10% less as compared to the case of 2-dimensional analysis. However since they tend to be well matched qualitatively with each other, it is judged that the 2-dimensional method could be successfully used as a purpose of distinguishing the qualitative superiority of design parameter when performing a large volume of calculations as in the parametric study.

2.3 Parametric Study



(a) N=3

(b) N=4

(c) N=6

Figure 4: Power Coefficient with Respect to Number of Blades

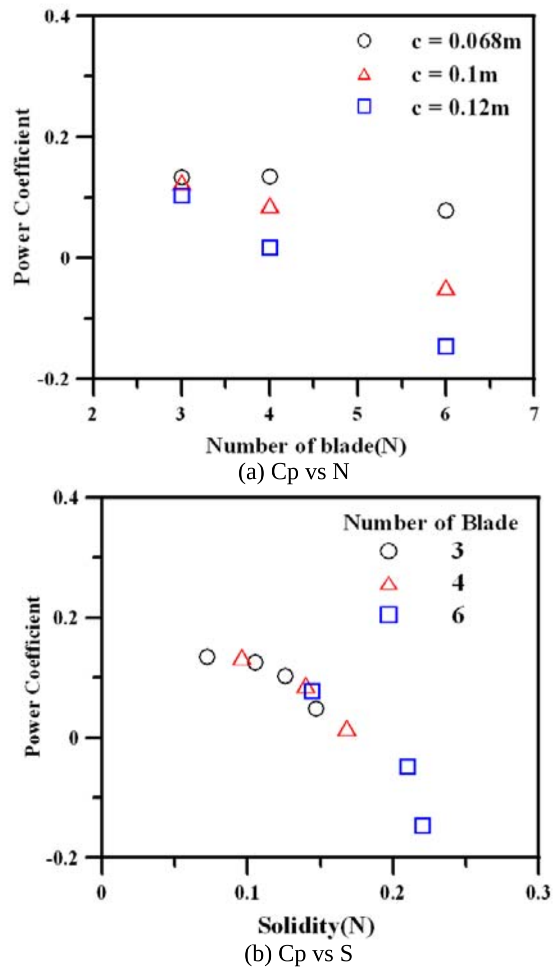
As for the design variables of rotor, the effects of solidity, pitch and camber were examined, while there are several parameters such as number of blades, chord, pitch, rotor diameter, span and blade shape. Solidity, as the ratio of the area occupied by the blade to the total area by the rotation of rotor blade, can be expressed as in the following,

$$S = \frac{Nc}{2\pi R} \tag{3}$$

where N denotes the number of blades. The 2-dimensional analysis was only performed for its simplicity and efficiency.

Effect of Solidity (Number of Blades and Chord Length)

Solidity is an important parameter that constructs the shape of rotor system. The effect of solidity is examined by varying the number of blades to be 3, 4 and 6 units and the chord length to be 0.068m, 0.1m and 0.12m, respectively. Fig. 4 shows the effect of the number of blades and chord length, and Fig. 5 shows the time-averaged power coefficients at various solidities.



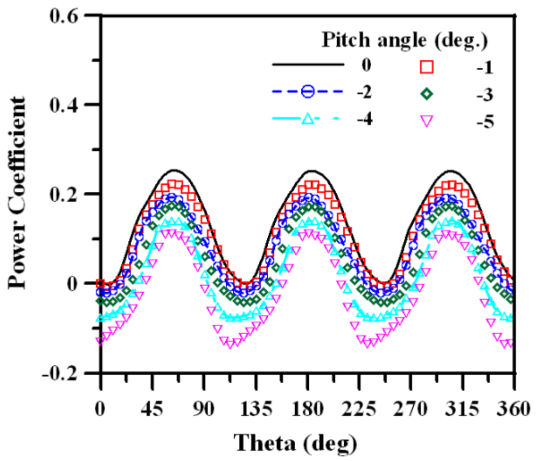
(a) Cp vs N

(b) Cp vs S

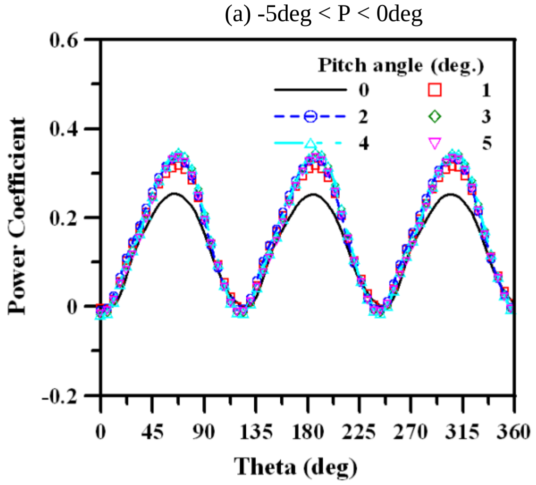
Figure 5: Mean Power Coefficient

Although no much difference in the average C_p was observed between 3 and 4 blades, very poor performance could be obtained when increasing the number of blades to 6 units. In other words, the efficiency is increased with less number of blades but the fluctuation of power coefficient gets large; and the reverse was observed with increasing the number of blades. Since the large fluctuation of C_p causes an undesirable result in the perspectives of system structure and electricity quality, it is recommended to use the model with more number of blades as long as its efficiency is somewhat within the acceptable range compared to other cases. In this perspective, the 4 bladed turbine could be a candidate in this demonstration of parametric study. Although it was anticipated that the increase of solidity could decrease the efficiency, no single proportional relationship was established between solidity and C_p.

Effect of Pitch



(a) -5deg < P < 0deg



(b) 0deg < P < 5deg

Figure 6: Power Coefficient with Respect to Blade Pitch

The blade rotates around the rotational axis and the incidence angle of flow is changed by varying the pitch using a mechanical device. Hence, the lift and drag forces could be changed and this could also influence on the efficiency. If we apply the optimum pitch on each instant in the rotational direction, the efficiency is increased and it was verified through several studies that the controllable pitch is far more efficient than the fixed pitch [2] Prior to the turbine performance analysis of controllable pitch, this study has performed the analysis by fixing the blade pitch P to several constants to see the effect of basic pitch. As for the analysis conditions, this study has set the number of blades to 3 units, diameter to 0.455m and chord length to 0.068m. For the flow velocity and TSR of 1m/s and 2.25, the calculation was performed while varying the pitch by 1° in the range of -5° to +5°. The values of C_p according to the change of pitch are shown on the Fig. 6 and Fig. 7.

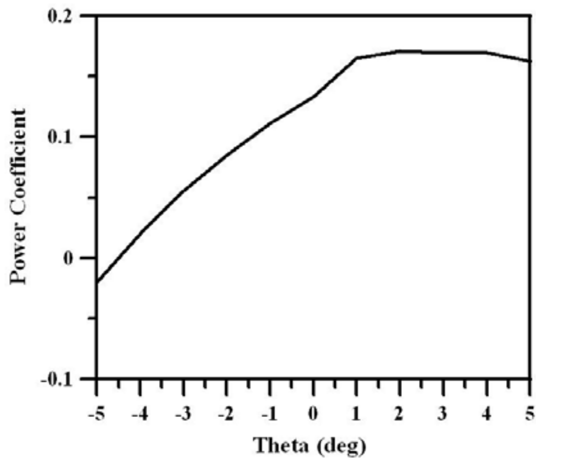


Figure 7: Mean Cp vs Pitch Angle

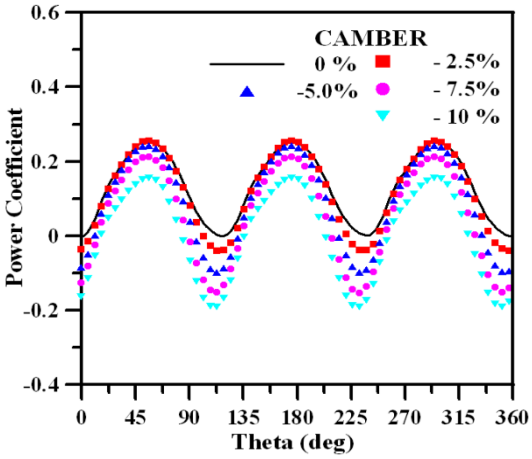
The overall efficiency was decreased in case of the negative pitch but in case of the positive pitch, the value was likely to increase. The result has shown the best efficiency at P=+4° but after the point, it has shown a decreasing tendency.

Effect of Camber

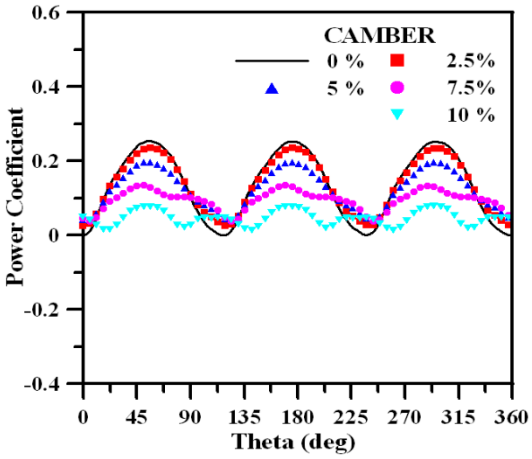
The camber f, as the parameter used frequently to adjust the lift-drag characteristics of symmetric blade, the analysis was conducted while varying the chord length by 2.5% in the range of -10% to +10%. Same as in the previous section, the number of blades was set to 3 units, diameter to 0.455m and chord length to 0.068m and the flow velocity and TSR were 1m/s and 2.25 respectively.

Fig. 8 shows the changes in power coefficient when the camber is negative. It was the highest when the value of camber is -2.5% and the peak value was decreased more as the camber gets large. However within the design scope, the effect of camber was minimal. Although the efficiency could be raised a bit by adjusting the camber finely under the respective design conditions for the sake of optimization, it may have the risk of dropping the efficiency drastically when things go wrong.

Fig. 10 shows the power coefficients obtained by each blade during the one rotation of the rotor when either there is no camber or the camber is 2.5%. Looking at the torque and performance coefficient by each blade, the maximum performance coefficient was higher than the case of no camber and the minimum efficiency was also proportionately far less when there was no camber, yielding almost same time-mean C_p for both cases.



(a) -10 % < f < 0 %



(b) 0 %< f < 10 %

Figure 8: Power Coefficient with Respect to Camber

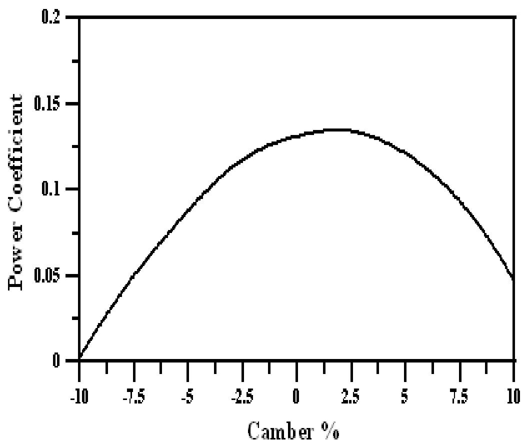


Figure 9: Mean Power Coefficient with Camber

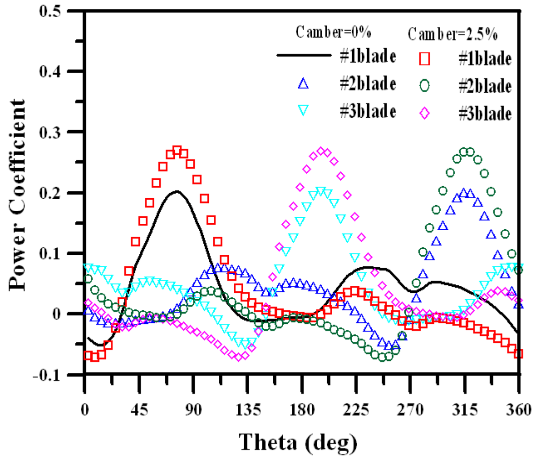


Figure 10: Comparison of Camber Ratio 0% and 2.5%

Design Parameters of Optimum Rotor

When TSR=2.25 and Re= 2.69×10^5 after comparing the performances according to the changes of solidity, blade pitch and blade camber, the design parameters of the optimum rotor in terms of efficiency can be arranged as in Table 2.

Current Speed	Number of blade	Chord	Pitch	Camber (of Chord length)
1 m/s	4	0.068m	+4deg	2.5%

Table 2: Selected Parameters.

Fig. 11 shows the result that has compared the numerical analysis and experimental result of [6] with the typical results of parametric study and the demonstrative optimum design conducted in this study. Table 3 also shows the average C_p. Although the optimum parameters drawn from the result of this study are limited in efficiency, the result shows that they can be applied relatively well onto the design simply by the 2-dimensional analysis.

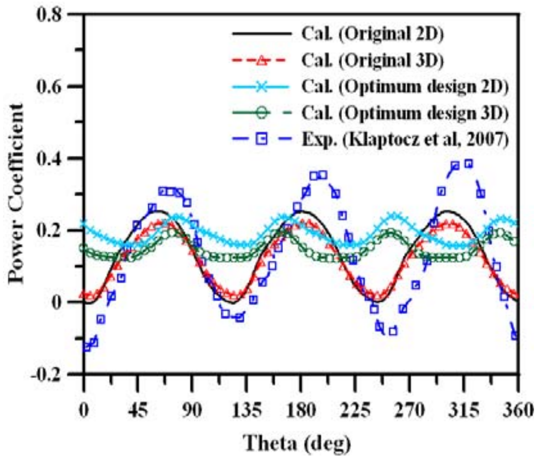


Figure 11: Power Coefficient at Optimum Designing Condition

3. Application to the Darrieus Turbine with NACA 653-018 Blade Section

3.1 Model, Numerical Analysis and Calculation Conditions

Numerical analysis for the design of Darrieus Turbine with the blade shape of high lift-drag ratio has been made together with the verification experiment in the circulating water channel. The selected model is called here as the #2 turbine model with NACA 653-018 blade section. As shown in Fig. 12, it is a symmetric foil and widely used in the wind turbine due to the excellent lift-drag ratio and stall performance. As for the turbine, this study has selected the 3-bladed rotor model for its higher efficiency according to the numerical result of [3] and has also examined the 4-blade turbine for the sake of comparison. After setting the turbine radius R to 0.4m, blade chord to 0.07cm and spans respectively to 0.6m and 0.85m, the latter case was designed in a shape that is projected by 5cm above the water surface to make a surface-piercing turbine with its span of 0.8m.

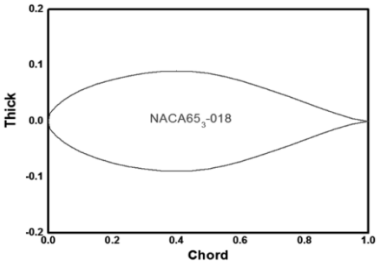
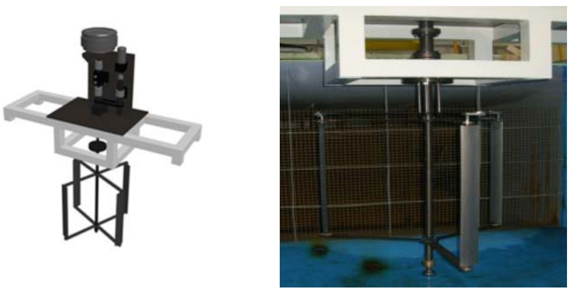


Figure 12: NACA 653-018 Blade Section of # 2

While the numerical approach is mostly identical to the details in the previous section, this study has only used one k-ω SST model as for the turbulence model. Both the 2-dimensional and 3-dimensional analyses were carried out at various TSRs. The 2-diemnsional analyses were carried out ranging from 2.6 to 4.2 , 3-diemnsional analyses were carried out at TSR 2.8, 3.2, 3.6. The Reynolds numbers (Re) at the flow velocity of V=1m/s correspond to the range from 1.96×10^5~2.94×10^5, respectively. Turbulent flow assumption was also made.

3.2 Experimental Equipments and Measurements



(a) Schematics (b) experiment set-up

Figure 13: Schematic & experiment set-up of #2 section model system

Fig. 13 shows the measuring system's schematic plan(a) and the set-up of experiment for the model #2. The test section of circulating water channel was 2.8m(L)×1.8m(B)×0.9m(H), and the experiment has conducted in the range of 0.6~1.2m/s of flow velocity while its maximum is 2 m/s. The flow uniformity in the test section was about 1% and the turbulence intensity was about 2%. As for the major experimental devices, there are the torquemeter with the capacity of 20N-m, power brake of 100N-m for generator loading, clutch that separates the rotor from the shafting for no loading condition, pony motor for quick self-starting and tachometer. In order to minimize the rotor vibration, the upper part of rotor was supported by the traverse system of circulating water channel and the lower part on the floor of water channel by using a ball bearing.

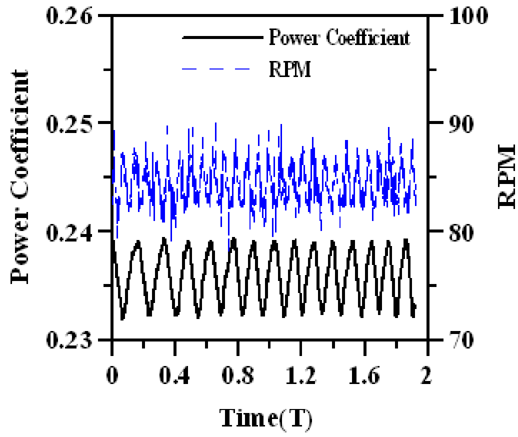
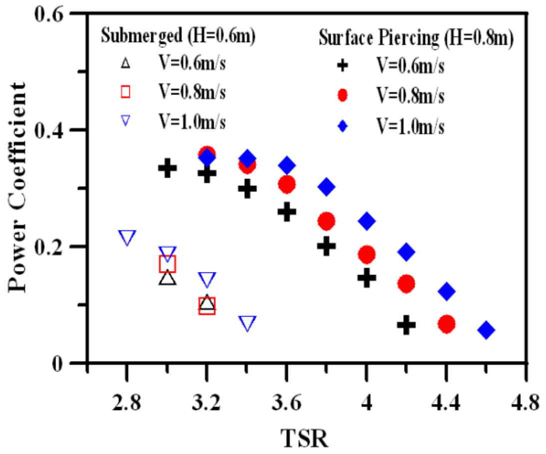


Figure 14: Sample Measurements

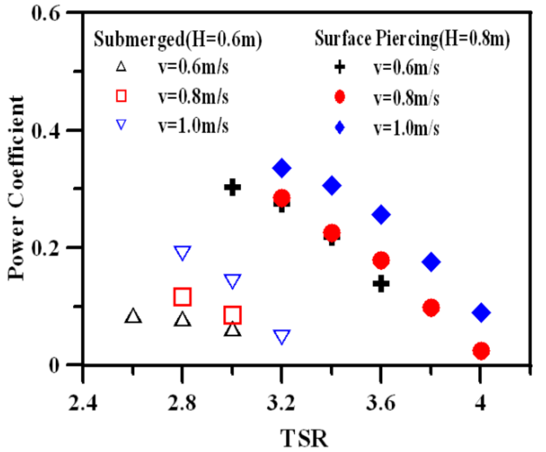
Fig. 14, as the sample of power coefficient and RPM with time, shows the experimental result when the 4-blade turbine is at the flow velocity of 1m/s and TSR=3.5. Unlike the numerical result that normally shows a large fluctuation of torque, the measured torque shows only a slight variability. Instead, it could be seen that the rotational speed is changed by a maximum of 8 rpm, although it was assumed to be a constant in numerical analysis. This, as the RPM variation of about 9%, could be regarded as reflecting the uncertainty attributable to the shaft assembly with an excessive moment of inertia and the bearing system with insufficient absorption capability against the vortex induced vibration. From this point of view, it is obvious that a certain discrepancy exists between the test conditions of numerical analysis and experiment. While it is necessary to study further to enhance the dynamic characteristics by reducing the measuring apparatus as simple as possible, it is judged that the average torque and average RPM could yield a valid quantitative result through appropriate static calibrations of torquemeter and tachometer.

3.3 Results and Discussions

First, Fig. 15 shows the turbine efficiency with respect to TSRs by varying the flow velocity, number of blades, and span. Even with the same number of blades and same span, some differences in power coefficient are apparent in case of identical TSR, showing the effect of flow velocity. Further study is necessary to elucidate the dependency of Reynolds number in efficiency. When the span is 0.6m, the rotational speed of turbine was about 80rpm at the flow velocity of 1m/s. The rotational speed tends to be systematically slower compared to the longer span. The surface-piercing blade with the span of 0.8m holds the rotational speed of about 100rpm at the flow velocity of 1m/s, showing the evidently faster rotational speed.



(a) N=3

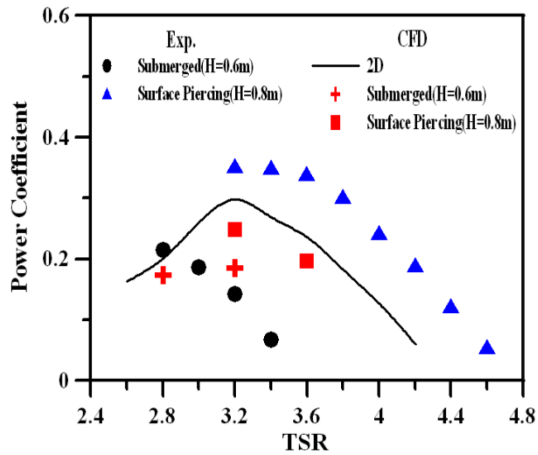


(b) N=4

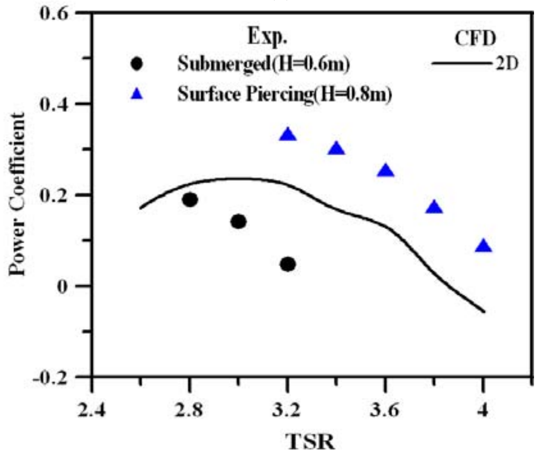
Figure 15: Measured Cp with Respect to Span and Current Velocity

Through this experiment, we could see that the operable TSR range can vary greatly by the interaction between longer turbine blades and free-surface. Power coefficient becomes also higher with a longer span of 0.8m. It is therefore quite possible that the special installation method such as a surface-piercing VAT can be properly utilized to get the best performance, especially when the water depth is limited or not sufficient.

Fig. 16 shows the numerical and experimental results obtained while considering the changes in the number of blades and span. A big influence of protruding blades is noticeable. As for the numerical analysis, the 2-dimensional and 3-dimensional numerical analyses were performed. When the span was 0.6m, it shows a difficulty in using the 2-dimensional analysis due to the large 3-dimensionality of blades. On the other hand when the span of 0.8m is projected above the water surface, we judge that the 2-dimensional analysis result could be acceptable since the protruding blade of 0.8m in span corresponds to the blade of 1.6m operated in infinite depth. Although the numerical analysis and experiment in general has not shown a nice matching yet, we assume that this in part could be applied to the design of vertical axis turbine. Further study is to be performed to validate the effectiveness and the detailed operating characteristics of a surface-piercing VAT.



(a) N=3



(b) N=4

Figure 16: Comparison Between Calculated and Measured Results

4. Conclusions

This study deals with the numerical and experimental analyses for performance prediction and design of Darrieus-type VAT. It is summarized as follows;

(1) The unsteady flow analyses have been performed by a commercial software Fluent for the #1 model with NACA634-021 blade section, where the experimental data is already available. The validation of numerical analysis has been completed by the successful 2-dimensional and 3-dimensional numerical results. In order to demonstrate the rotor optimization technique by properly selecting the blade parameters

such as solidity, pitch and camber of turbine blade, the 2-dimensional numerical method has been utilized for its simplicity. The optimum turbine with 4 blades was designed by reflecting the designer's considerations to prove the effectiveness of the present approach as a design tool.

(2) For the NACA653-018 blade with high lift-drag ratio, the numerical and experimental study has also been carried out. The 2-dimensional and 3-dimensional numerical results were verified through the experiment performed at a circulating water channel. The experimental result reveals that the rotational speed of turbine blades is not constant, but shows the variation of about 8%. The measured torque was fluctuating rather properly due to the large moment of inertia. More sophisticated experimental system is needed to enhance the quality of experiment. However some meaningful results were obtained in time-averaged sense that the influence of blade span and/or the effect of surface-piercing blade is found to be very noticeable so that the special installation method is possible especially when the water depth is limited. we could see that the influence of flow velocity has also existed in part under same TSR. Overall, it could be seen that the numerical analysis and experiment show similar tendency and useful result in a sense.

(3) Through the future studies on special turbines such as surface-piercing VAT, ring-type VAT and controllable pitch VAT, we plan to continue the studies of improving turbine performance.

Acknowledgements

This research is financially supported by The Ministry of Knowledge Economy of Korea, managed by Korea Institute of Energy Technology Evaluation and Planning. (2009T100100556 and 2009T100100245)

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