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A straight-bladed vertical axis turbine for wave energy conversion: Effect of guide vane position on the performance

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Abstract. A straight-bladed vertical axis turbine for wave energy conversion has been proposed in order to develop a novel air turbine suitable for an oscillating water column based on a wave energy plant. This study aims at investigating the effect of guide vane position on the performance of a straight-bladed vertical axis turbine. The experimental study was performed using a wind tunnel. The rotor comprises four straight blades with a profile of NACA0018, chord length of 80.5 mm, pitch radius of R = 230 mm, and a blade width of 490 mm. The guide vane has a profile of a circular arc and a chord length of l = 87 mm. The distance between these two guide vanes w and the gap between the rotor blade and guide vane G were altered to investigate the effects of the guide vane solidity l/w and position G/R. In addition, we performed computational fluid dynamics (CFD) on exactly the same configuration as the experimental system to study the effect of the guide vane position. The experimental results indicated that the guide vane solidity l/w and position G/R suitable for the proposed turbine are obtained in the case of l/w = 1.09 and G/R = 0.11. The flow visualization was determined using CFD. Moreover, we confirmed the control of the flow separation by the setting of the guide vane.

1. Introduction

Generally, an oscillating water-column (OWC)-type wave power generator uses a reciprocating air current generated by the reciprocating motion of the water in the air chamber, and power generation is performed using an air turbine. This study aims at developing a wave power generator wherein a vertical axis turbine operates in a reciprocating air current (refer to Figure 1) [1]. In this study, the turbine performance test using steady flow was performed and the influence of guide vane installation angle, gap between the rotor blade and guide vane, the solidity of guide vanes and the number of guide vanes on its performance was investigated. In addition, flow visualization by numerical analysis that uses the CFD was performed.

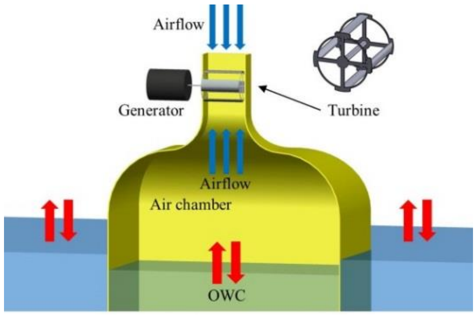
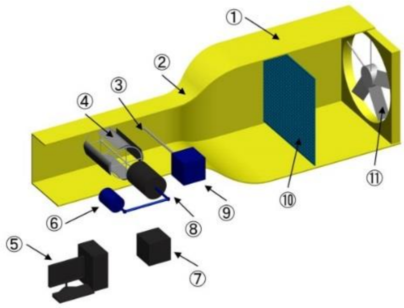


Figure 1. Oscillating water-column (OWC)-based wave energy plant with a vertical axis turbine.

2. Experimental and numerical analysis methods

2.1. Experimental method

Figure 2 illustrates a schematic of the test apparatus. To investigate the turbine performance, the airflow from the blower was rectified, and a steady flow was generated in a test section comprising a 2,400 mm-long square cross section and 498 mm on one side connected to the nozzle outlet of the wind tunnel. A wind tunnel test was conducted by maintaining a constant rotation speed of the turbine. The turbine was located 1,200 mm from the outlet. In the experiment, the torque T_o [N·m], angular velocity ω [rad/s], total pressure difference before and after the turbine Δp [Pa], and flow rate Q [m^3/s] were measured using a torque transducer (ONO SOKKI SS-050) and a Pitot tube (Testo 0635). Here, the measurement accuracy of each measuring instrument is ±0.2 %. We adopted NACA0018 as the shape of the blade, which is suitable for this turbine [1].

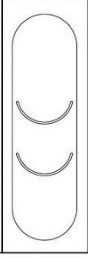
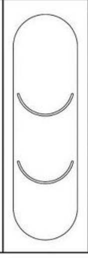
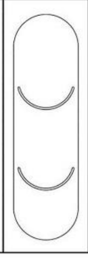
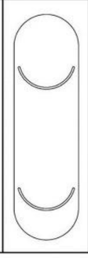
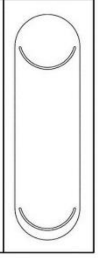


① Wind tunnel, ② nozzle, ③ Pitot tube, ④ turbine, ⑤ PC, ⑥ servomotor,
⑦ controller, ⑧ torque transducer, ⑨ pressure transducer, ⑩ mesh, and ⑪ fan

Figure 2. Schematic representation of the experimental apparatus.

Figure 3 illustrates a schematic of the test turbine used in this study. This straight-bladed vertical axis turbine has advantages of rotating in the same direction even in reciprocating airflow, of having higher efficiency than other vertical axis turbines, and of not rotating at higher speed and having less noise than Wells turbine. To correspond to the reciprocating airflow, the guide vanes were installed on both the upstream and downstream sides of the turbine. The guide vane was an arc blade with a chord length l = 87 mm and a thickness of 3 mm along with a radius of curvature r = 45 mm, which is suitable for this type of guide vane [2, 3]. The case illustrated by Figure 3 is a Type I configuration, and the case in which the Type I vanes are reversed is a Type II configuration. First, to investigate the influence of the gap G between the rotor blade and guide vane, values of G = 25, 35, and 45 mm were investigated. The value of G/R = 0.11, 0.15 and 0.2 for G = 25, 35, and 45 mm, respectively, when expressed as a ratio of the rotor pitch circle radius R. Next, to investigate the influence of the solidity of guide vanes, we changed the guide vane interval w to 80, 105, 125, 188, and 250 mm for the two guide vanes used in our experiment (refer to Table 1). The solidity λ of guide vane is defined as $\lambda = l/w$. Furthermore, to investigate the influence of the number of guide vanes, we changed the number of guide vanes n to 1, 2, and 3. The installation angle θ of the guide vane was set to $\theta = 5°$ and $-5°$ based on the findings of a previous study [2]. At the highest efficiency point, the Reynolds number based on the chord length is 1.2×10^5.

Table 1. Solidity of guide vanes.

w	80	105	125	188	250
λ	1.09	0.829	0.696	0.464	0.348
					

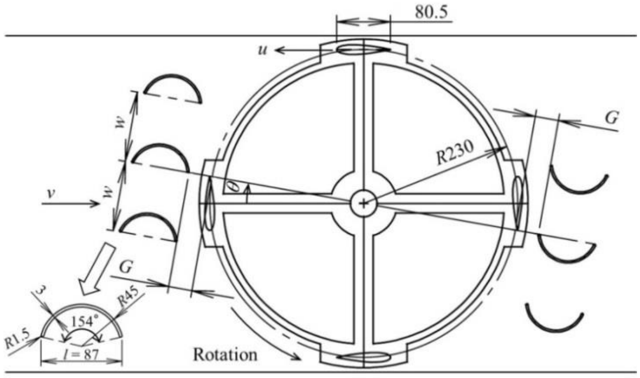


Figure 3. Turbine configuration (Type I).

2.2. Numerical analysis method

A numerical analysis was performed using Particleworks software and employing the moving particle semi-implicit (MPS) method. The MPS method is a particle method that targets incompressible flow, in which a fluid is discretized by particles and the particle motion is calculated using the Lagrangian method. Compared to the finite volume method, the MPS method does not require mesh generation. Therefore since it is easy to derive calculated results from the introduction of the calculation model, it was adopted for numerical analysis.

The length of the test section used in the experiment was 1,200 mm, and the length in the span direction was 150 mm. In addition, in order to improve the visualization accuracy, the particle diameter was set to 8 mm.

3. Results and discussion

3.1. Experimental results and discussion

The turbine performance under steady flow conditions evaluated by turbine efficiency η, torque coefficient C_T and input coefficient C_A against flow coefficient ϕ. The definitions of these parameters are as follows:

$$C_T = T_o / \{\rho(v^2+u^2)AR/2\} \tag{1}$$

$$C_A = \Delta pQ / \{\rho(v^2+u^2)Av/2\} = \Delta p / \{\rho(v^2+u^2)/2\} \tag{2}$$

$$\eta = T_o\omega/(\Delta pQ) = C_T/(C_A\phi) \tag{3}$$

$$\phi = v/u \tag{4}$$

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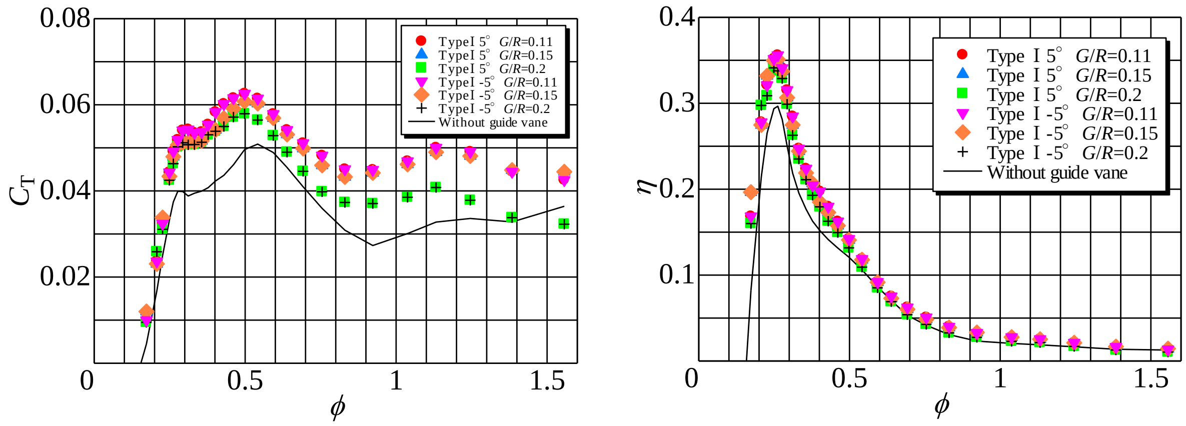
where ρ is the air density [kg/m^3], A is the flow cross-sectional area [m^2], R is the rotor pitch circle radius [m], u is the circumferential velocity of the rotor [m/s], v is the average cross-sectional flow velocity [m/s], and ω is the angular velocity of the rotor [rad/s].

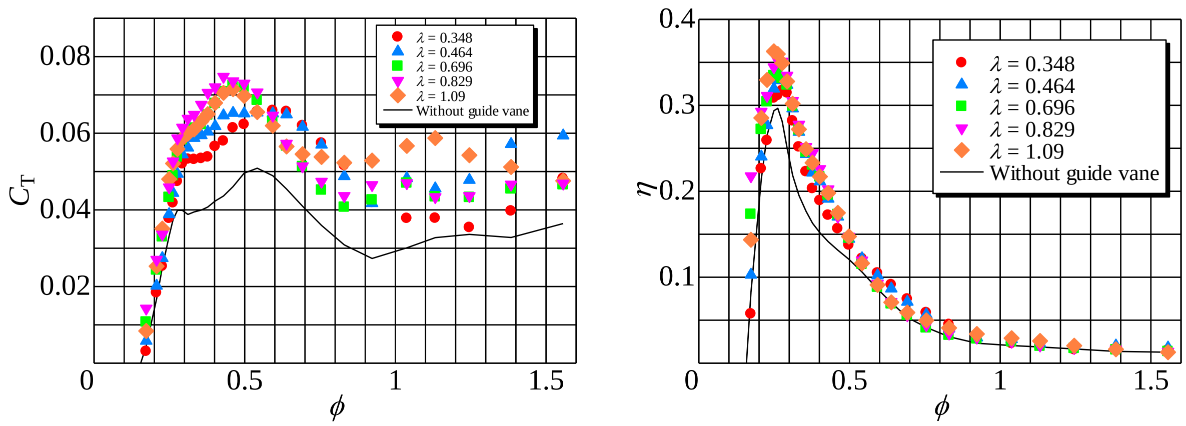
Figure 4 illustrates the influence of the gap between the rotor blade and guide vane, Figure 5 illustrates the influence of the solidity of guide vanes, and Figure 6 illustrates the experimental result of the influence of the number of guide vanes. In each figure, (a) depicts the torque coefficient, and (b) depicts the turbine efficiency. Here, the results from Type I, which was more efficient than Type II, are presented.



(a) Torque coefficient (b) Turbine efficiency

Figure 4. Influence of the gap between the rotor blade and guide vane on the torque coefficient (a) and the turbine efficiency (b).

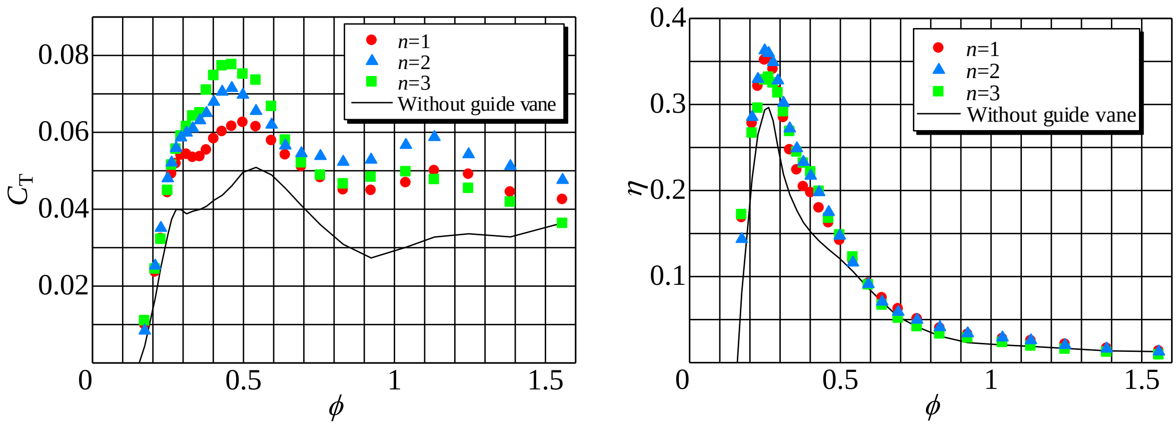
In Figure 4, it is clear that the Type I configuration, for which $\theta = -5°$ and $G/R = 0.11$, exhibits the highest turbine efficiency compared with the other installation conditions. It can also be seen that the turbine efficiency increases as the gap between the rotor blade and guide vane decreases at $\theta = -5°$, so decreasing the gap between the rotor blade and guide vane increases the turbine efficiency.



(a) Torque coefficient (b) Turbine efficiency

Figure 5. Influence of the solidity of guide vanes.

Figure 5 illustrates the influence of the solidity of guide vanes in the Type I configuration, where $\theta = -5°$ and $G/R = 0.11$, showing the highest turbine efficiency in Figure 4. The highest efficiency was at λ = 1.09, when the spacing between the guide vanes was the closest, indicating that the turbine efficiency decreases as the guide vane spacing is increased. This is probably because the airflow hitting the turbine blades was increased by bringing the guide vane intervals closer to each other, thus enhancing the wind collecting effect.



(a) Torque coefficient (b) Turbine efficiency

Figure 6. Influence of the number of guide vanes.

Figure 6 illustrates the influence of the number of guide vanes in a Type I configuration, where $\theta = -5°$ and G/R = 0.11 for λ = 1.09, which exhibits the highest efficiency as determined by Figures 4 and 5. The number of guide vanes was varied between 0 and 3, and the highest efficiency was achieved with two guide vanes. In the three guide vane configuration, the additional increase in the number of guide vanes caused a conspicuous decrease in the turbine efficiency.

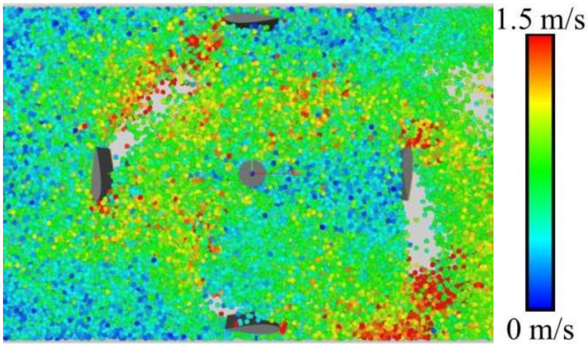
In addition, in (b) of each figure, an improvement in the turbine efficiency was seen when the flow coefficient was $0.2 < \phi < 0.5$, because the torque coefficient increased near that point as seen in (a) of each figure.

3. 2. Numerical analysis results and consideration

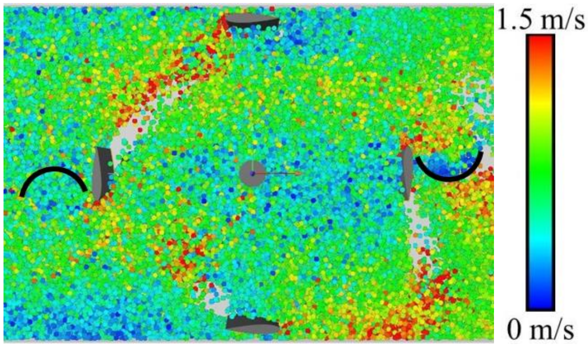
Figure 7 illustrates the flow visualization results as calculated by the MPS method, which is a type of CFD. Figure 7(a) illustrates the case without a guide vane, 7 (b) illustrates a case in which the number of guide vanes is n = 1, 7 (c) illustrates a case in which the number of guide vanes is n = 2, and (d) illustrates a case in which the number of guide vanes is n = 3. The numerical analysis results illustrated in Figure 7 are for when the flow coefficient is ϕ = 0.262 and the turbine efficiency is at its highest point.

In the case without any guide vanes (a), separation occurs only in the downstream flows of the turbine blades on both the upstream and the downstream sides, but in (b), (c) and (d), where the guide vanes are installed, the upstream side and the downstream side the separation at the downstream side of the turbine blade was suppressed on the side of the turbine blade. The suppressing separation was particularly large when the number of guide vanes was n = 2 (b) or 3 (c).

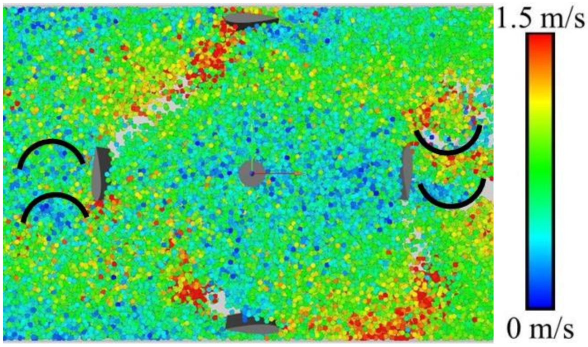
These calculated results seem to support the data plotted in Figure 6 (b). However, the reason why the turbine efficiency is low, despite the fact that the number of guide vanes is n = 3 has a greater positive effect than the suppression in efficiency caused by separation, is due to the high input coefficient C_A (not presented). Therefore, even though the torque coefficient C_T has a high value, as obtained by the deterrent suppression effect in Figure 6 (a), the turbine efficiency seems to be low in relation to the input coefficient C_A.



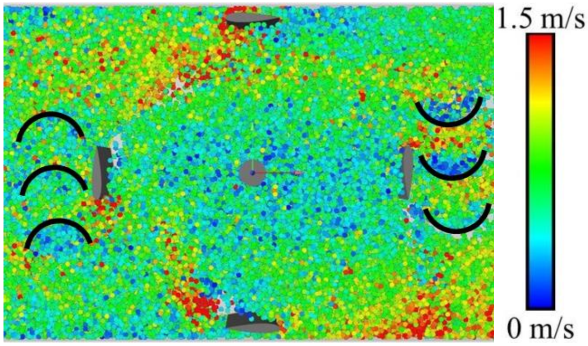
(a) Without guide vane



(b) $n = 1$



(c) $n = 2$



(d) $n = 3$

Figure 7. Flow visualization at $\phi = 0.262$ (Type I: $\theta = -5°$ and $G/R = 0.11$) as calculated for turbines with (a) 0, (b) 1, (c) 2 and (d) 3 guide vanes.

4. Conclusion

In the scope of this study, the most suitable guide vane setting conditions were $G/R = 0.11$, $\theta = 5°$, $\lambda = 1.09$, and $n = 2$. Moreover, from the visualization results of the flow using the MPS method, it was confirmed that separation is suppressed at the downstream side of the turbine blade, and installation of guide vanes is effective for improving the turbine performance.

Acknowledgements

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