Experimental Investigation on Performance of Counter-rotating Impulse Turbine with Middle Vanes for Wave Energy Conversion

Koichiro Suto, Manabu Takao, Toi Ogawa, Shinya Okuhara, Navneet Kumar, M.M.A. Alam and Yoichi Kinoue

Abstract— **In an oscillating water column (OWC) based wave energy device, a water column that oscillates due to the sea wave motion generates a bi-directional airflow in an air chamber, and finally, air turbine driven by the bi-directional airflow converts the pneumatic energy into the mechanical energy. A counter-rotating impulse turbine for bi-directional airflow was proposed by M. E. McCormick of the United States Naval Academy in 1978.**

This counter-rotating impulse turbine has a disadvantage that the efficiency in a range of low flow coefficient is remarkably low due to the deterioration of the flow between the two rotors. In a previous study, the authors proposed to install middle vanes between the two rotors in order to overcome the above drawback, and the effect of the middle vanes on the turbine performance was investigated by CFD analysis. As a result, it was found that the middle vanes installed to the turbine rectify the flow between the rotors in a range of low flow coefficient, and it drastically increases the efficiency.

In this study, the middle vanes were installed in an actual counter-rotating impulse turbine for bi-directional airflow, and its performance was elucidated by wind tunnel tests. As a result, it was found that the use of middle vanes between both the rotors increases the maximum efficiency of the turbine by more than 1.1 times.

Keywords—**bi-directional flow, counter-rotating turbine, impulse turbine, ocean energy, turbomachinery, wave energy conversion.**

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I. INTRODUCTION

THE wave energy conversion based on the oscillating water column (OWC)[1] principle is one of the most widely used wave energy utilization technologies, as shown in Fig. 1. In this principle, first, a water column in an air chamber oscillates in accordance with the movement of sea waves, and the pressure change in the air chamber generates a bi-directional airflow.

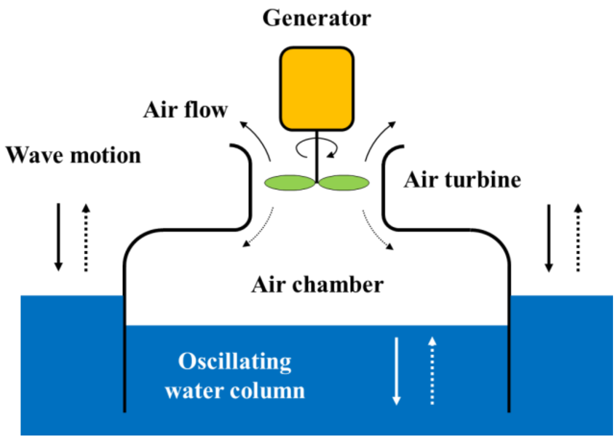

Fig. 1. Oscillating water column (OWC) based wave energy device.

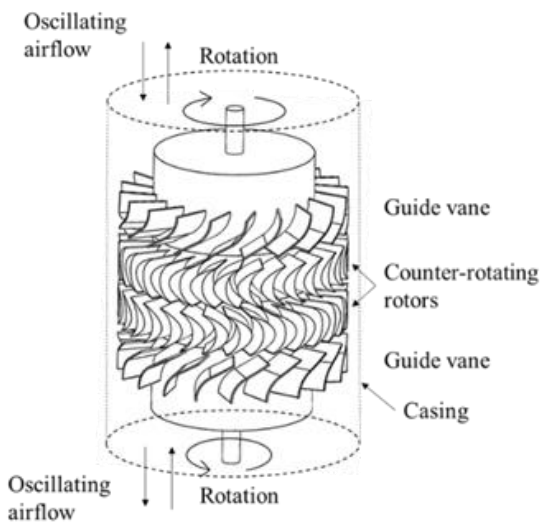

Fig. 2. Schematic of counter-rotating impulse turbine for wave energy conversion.

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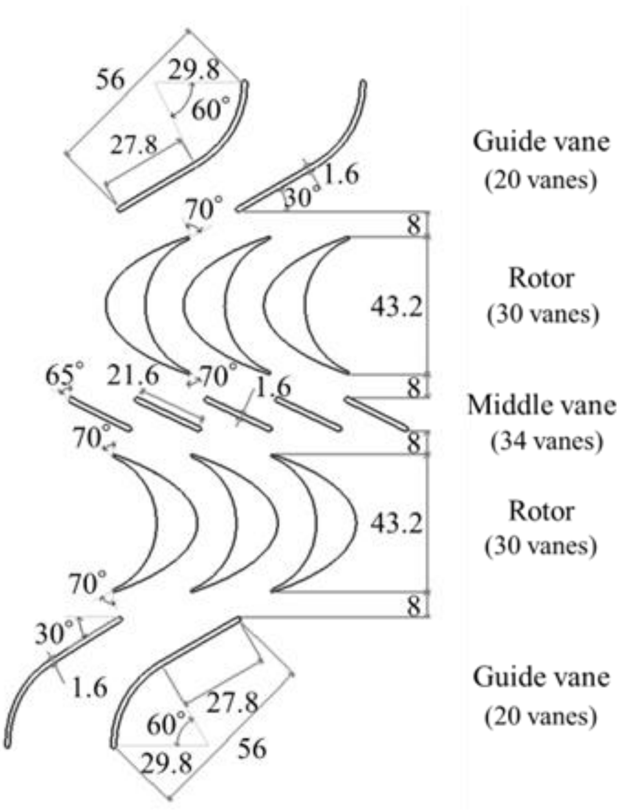


Fig. 3. Configuration of tested turbine cascade.

TABLE I
VARIABLES AND UNITS

Symbol	Quantity	Unit
ω	Rotor angular velocity	rad/s
T_o	Output torque	Nm
Δp	Pressure difference between before and after the turbine	Pa
Q	Flow rate	m^3/s
C_T	Torque coefficient	-
C_A	Input coefficient	-
η	Turbine efficiency	-
A	Cross-sectional area of the turbine channel	m^2
u	Rotor peripheral velocity at r	m/s
r	Mean radius	m
v	Axial velocity	m/s
ρ	Air density	kg/m^3
ϕ	Flow coefficient	-
T_{of}	Torque of front rotor	Nm
T_{or}	Torque of rear rotor	Nm

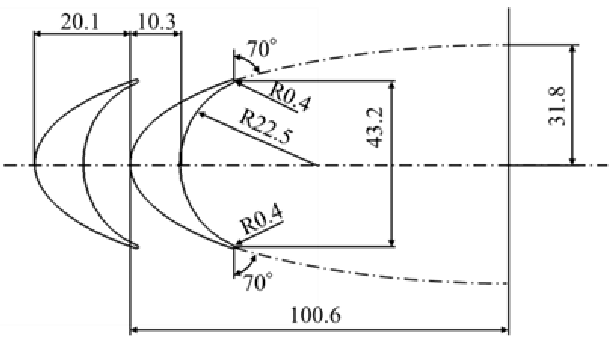


Fig. 4. Blade profile of tested rotor.

The bi-directional airflow rotates an air turbine and generates electricity. In the OWC type wave energy conversion, the air turbine always rotates in a same direction under bi-directional airflow. In 1978, M. E. McCormick[2] of the United States Naval Academy proposed a counter-rotating impulse turbine for bi-directional airflow as shown in Fig. 2. The configuration of the counter-rotating impulse turbine has two rows of impulse rotor and guide vanes as shown schematically in Fig. 2. The advantages of the counter-rotating impulse turbine are that the downstream

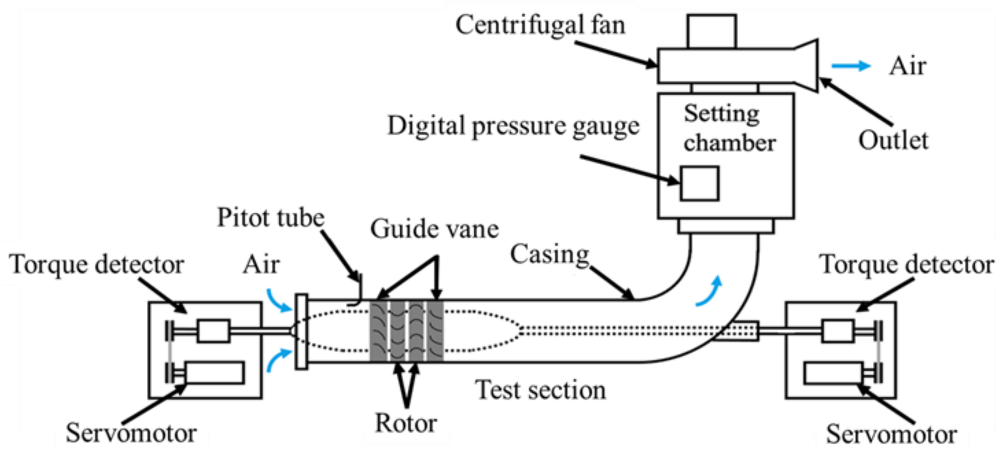


Fig. 5. Wind tunnel test equipment (plan view)

rotor collects the airflow from the upstream rotor to provide torque, and the two rows of impulse rotors invert in phase with each other to offset the torque generated externally. Therefore, the casing is easy to install. On the other hand, the disadvantages of the counter-rotating turbine are that it is noisy, and the shaft structure is complex, making it expensive to manufacture. In addition, field experiments have shown that the efficiency of this turbine is higher than that of the Wells turbine in small waves [3]. In a previous study, authors investigated the effect of turbine geometry on the performance, and it was clarified that the turbine's efficiency of the turbine is higher than that of an impulse turbine with a single rotor in a range of high flow coefficient [4]. In addition, two rows of impulse rotors are phase-reversed so that the generated torques cancel each other out. As the hydrodynamic characteristics of the counter-rotating impulse turbine are unknown, the basic performance and flow conditions inside the blade train were investigated by a CFD analysis, and the turbine has significantly lower efficiency at low flow coefficient due to difference between the downstream rotor. In order to overcome this drawback, the method of installing middle vanes between two rotor rows has been proposed by the authors[5]. However, the detailed performance of this turbine with middle vanes has not been investigated by the wind tunnel test, though it has been estimated by using CFD analysis. In this study, the middle vanes were installed between two rotor rows in an actual counter-rotating impulse turbine for wave energy conversion, and its performance was elucidated by the wind tunnel test.

II. Tested Turbine and Experimental Method

THE rotors and guide vanes of the counter-rotating impulse turbine used in this study have the optimal geometry that was reported in previous studies, and their cascade is shown in Fig. 3. This turbine consists of two rows of two-dimensionally shaped impulse-type rotors with a blade profile consisting of a circular arc and a portion of an ellipse, and two rows of guide vanes with a blade profile consisting of a circular arc and a straight line in front and behind the rotors, with middle vanes to control the air flow between the rotor blade rows.

The tested guide vanes are thin blades with a thickness of 1.6 mm, installed in front of and behind two rows of rotor blades at a distance of 8 mm from the rotors, with a tip diameter of 238.2 mm, chord length of 56 mm, length of the straight section of 27.8 mm, circular arc radius of 29.8 mm, a setting angle of 30°, the number of blades per row is 20, and solidity of 1.75. The tested rotors shown in Fig. 4 has the following specifications: tip diameter of 238.8 mm, chord length of 43.2 mm, inlet (outlet) angle of 70°, thickness ratio of 0.3, hub diameter of 168 mm, hub-to-tip ratio of 0.7, number of blades per stage of 30, and solidity of 2.02.

The tested middle vane is a 1.6 mm-thick flat plate blade placed between two rotor rows at a distance of 8 mm from the rotor rows, with setting angle of 65°, chord length of 24.8 mm, and solidity of 1.12.

The test apparatus [6] used in this study is shown in Fig. 5. A centrifugal fan and a settling chamber connected to a turbine test section in a wind tunnel generating steady flow. In the experiments, the rotational speed of the rotor was varied in 100 rpm increments over a rage of 0 rpm to 1400 rpm ($0 \le \omega \le 147$ rad/s, ω: rotor angular velocity), and the torque T_o, the pressure difference between before and after the turbine Δp, and the flow rate Q were measured.

Moreover, the turbine performance at steady flow was calculated by the torque coefficient C_T, the input coefficient C_A, the efficiency η, and the flow coefficient ϕ, expressed by the following equations.

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

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

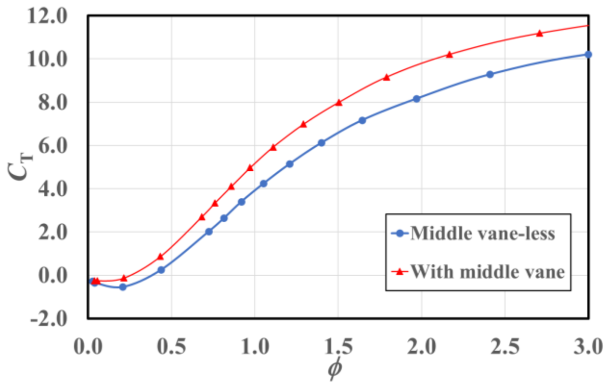
$$\eta = T_o\omega \,/\, (\Delta pQ) = C_T \,/\, (C_A\phi) \tag{3}$$

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

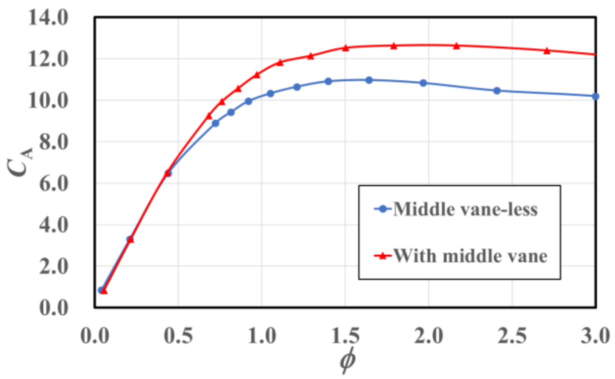
Here, A is the cross-sectional area of the turbine channel $\{= \pi D^2(1 - v^2)/4 = 0.02307 m^2\}$, D is inner diameter of casing, To is the output torque, u is the rotor peripheral velocity at r, v is the axial velocity ($= Q/A$), Δp is the total pressure difference before and after the turbine, ρ is the air density, and ω is the rotor angular velocity.

III. Experimental Results and Discussion

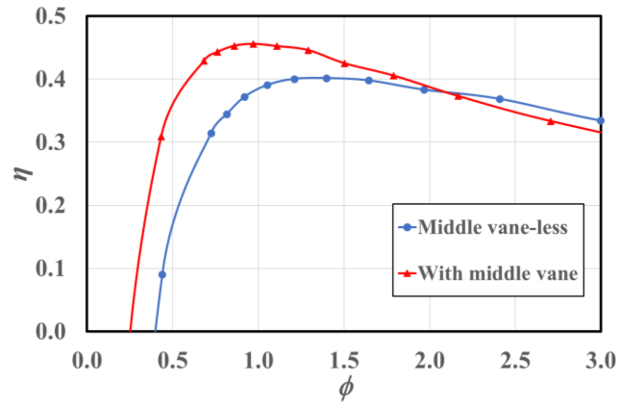
FIG. 6 shows the effect of the middle vanes on torque coefficient under steady flow conditions. The vertical axes of the figures (a) to (c) are the torque coefficient C_T, the input coefficient C_A and the efficiency η, respectively. The horizontal axes of all the graphs are the flow coefficient ϕ. As shown in Fig. 6(a), the installation of the middle vanes increases C_T. The reason is assumed that the middle vanes rectify the airflow between the rotor rows and the torque generated by the rotor at downstream side increases. On the other hand, in Fig. 6(b), C_A increase by the installation of the middle vanes because the collision losses at its inlet and friction losses between the vanes increase the pressure difference Δp. Fig. 6(c) indicates the peak efficiency of the turbine are 0.401 and 0.456 in the case of without middle vane and in the case of the turbine with middle vane, respectively. This result is attributed to the fact that the rate of increase in torque due to the installation of middle vanes was greater than that of the pressure difference. Therefore, it can be concluded from the above results that the turbine performance is improved by the installation of the middle vanes.



(a) Torque coefficient



(b) Input coefficient

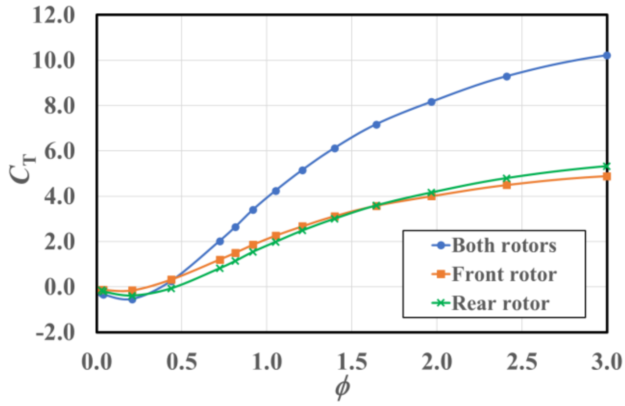


(c) Efficiency

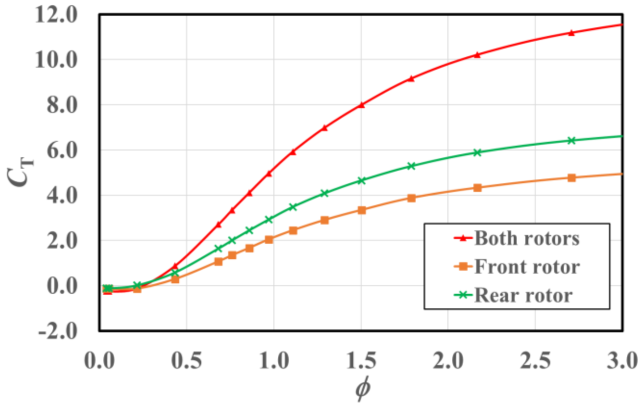
Fig. 6. Effect of middle vane on the turbine characteristics under steady flow conditions.

Fig. 7 shows the respective torques of the front and rear of rotors without and with the middle vanes, as well as the combined torque of front and rear torques. The middle vane installation rectifies the downstream flow and increases the torque of the rear rotor.

Fig. 8 shows the corresponding percentages of the torques of the front and rear rotors in total torque. Here, T_o refers the total torque, T_{of} means the torque in the front rotor, and T_{or} means torque in the rear rotor. In the case of



(a) Middle vane-less



(b) With middle vane

Fig. 7. Effect of middle vane on torque coefficient of each rotor.

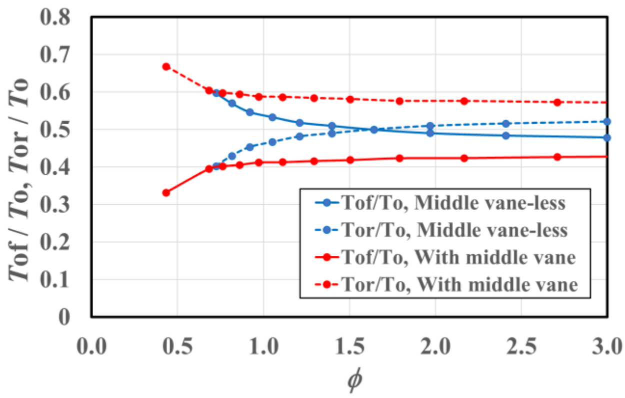


Fig. 8. Torque generated by the front and rear rotors.

without middle vane, the torque in the front rotor accounts for a large proportion of the total torque in the low flow coefficient. As the high flow coefficient, the proportion of torque in the rear rotor increases. On the other hand, in the case with middle vanes, the torque of rear rotor accounts for a larger proportion in the total torque at whole range of flow coefficient, and the ratio of the torque of rear rotor to that of front rotor is approximately 6:4. The reason is due to the flow rectification in the downstream rotor by the installation of the middle vanes.

IV. CONCLUSIONS

THE effect of middle vanes on the performance of counter-rotating impulse turbine for wave energy conversion was investigated by a wind tunnel test under a steady flow condition, in this study. The obtained results are summarized as follows:

(1) The peak efficiencies obtained are 0.401 and 0.456, respectively, for the turbine without and with middle vanes.

(2) The peak efficiency can be increased to 1.14 times by installing the middle vane.

(3) The middle vanes rectify the airflow between the rotor blade rows and increases the torque generated by the downstream rotor.

(4) The middle vane increases the pressure difference across the turbine due to collision losses at inlet and friction losses in the vanes.

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