A study of counter-rotating impulse turbine for wave energy conversion
- Effect of middle vane on the performance -

M Takao¹, K Yamada², R Sasaki¹, M M A Alam¹, S Okuhara¹ and Y Kinoue³

¹ National Institute of Technology, Matsue College, 14-4 Nishiikuma-cho, Matsue-shi, Shimane 690-8518, Japan
² Shinko Techno Engineering Co., Ltd., 2-3-1 Shinhama, Arai-cho, Takasago-shi, Hyogo 676-8670, Japan
³ Saga University, 1 Honjo-machi, Saga-shi, Saga 840-8502, Japan
e-mail: takao@matsue-ct.jp

Abstract. In an oscillating water column (OWC) wave energy converter, the oscillating water column caused by the sea wave motion is used to drive an air column in the air chamber. An air turbine for this bi-directional airflow is used to convert the pneumatic energy into the mechanical energy. A counter-rotating impulse turbine for wave energy conversion has been proposed by M. E. McCormick in 1978. In a previous study, authors investigated the effect of turbine geometry on the performance of the counter-rotating impulse turbine and clarified that the efficiency of the turbine is higher than a single rotor impulse turbine in a range of high flow coefficients. However, the counter-rotating impulse turbine has a disadvantage. Its efficiency in a range of low flow coefficients is remarkably lower due to the deterioration of flows between two rotors. In this study, middle vanes were installed between two rotors in order to achieve a further improvement in the performance, and its effect was investigated by using the computation fluid dynamic (CFD) analysis. As a result, it was found that the efficiency of the turbine was greatly improved by installing the middle vanes, and a favourable turbine geometry was clarified.

1. Introduction

Several devices have been studied under many wave energy programs in Japan, the United Kingdom, Portugal, India and other countries by making use of the principle of oscillating water column (OWC). In such wave energy devices, a water column which oscillates due to the sea wave motion is used to drive an air column as shown in Figure 1. An air turbine is used to convert the pneumatic energy of this bi-directional airflow into the mechanical energy [1].

The counter-rotating impulse turbine for wave energy conversion has been proposed by M. E. McCormick of the United States Naval Academy in 1978 (Figure 2) [2]. In a previous study, authors investigated the effect of turbine geometry on the performance of the counter-rotating impulse turbine and clarified that the efficiency of this turbine is higher than a single rotor impulse turbine in a range of high flow coefficients [3]. However, the counter-rotating impulse turbine has a disadvantage that the efficiency in a range of low flow coefficients is remarkably low due to the deterioration of flows between two rotors. In this study, middle vanes were installed between two rotors in order to achieve a further improvement of the performance of the counter-rotating impulse turbine, and the effect of middle vanes on turbine performance was investigated by using the computation fluid dynamic (CFD) analysis.
2. CFD analysis

Figure 3 shows a detail about the configuration of analyzed counter-rotating impulse turbine. The specifications of turbine rotors and guide vanes are the same as used in an impulse turbine with single rotor for wave energy conversion [4].

The specifications of the tested turbine rotors are as follows: casing diameter $D=300$ mm; chord length of 54 mm; blade thickness ratio of 0.3; tip diameter of 299 mm; hub diameter of 210 mm; hub-to-tip ratio $\nu=0.7$; mean radius $R = (1+\nu)/4$; inlet and outlet angle of rotor blade is of 70°; the number of blades are of 30; solidity at mean radius 2.02. The details of guide vanes are as follows: chord length of 70 mm; blade thickness of 2 mm; the number of vanes are of 20; solidity at mean radius of 1.75; setting angle of fixed guide vane is of 30°. In addition, although the thick-wing guide vanes were used in the McCormick's proposed turbine, it has been clarified later through a conventional research that there is no difference in the performances between the thick and thin wing guide vanes [5]. Therefore, thin blades with a simple shape were used as guide vanes in this research.

The specifications of the middle vanes are as follows: chord length $l=27$ mm; blade thickness of 2 mm, number of vanes are $z$; solidity at mean radius $\sigma = (lz/(2\pi R))$; setting angle of middle vane is $\delta$. In this research, the setting angle $\delta$ and solidity $\sigma$ of middle vanes were changed to investigate their effects on the performance characteristics of the counter-rotating impulse turbine. Table 1 shows the specifications.
of middle vanes used in this study.

The numerical analysis was conducted using a commercial CFD tool of SCRYU/Tetra developed by Cradle Co., Ltd. The computational domain used in this study consists of a casing, a cone and the turbine. Considering the computational cost, a half of the domain in axial symmetry was computed. The domain was composed of approximately 5,800,000 mesh elements. The Reynolds averaged Navier-Stokes (RANS) equations were used as the governing equations, and the low Reynold’s number SST $k$-$\omega$ model was used to predict the turbulent stresses. The flow was considered as steady-state. The no-slip boundary condition was set to the solid boundaries. The flow rate at the inlet was kept constant at $Q=0.320\,\text{m}^3/\text{s}$, and the outlet was opened to the atmosphere.

The numerical model used in the present CFD work had already validated through a comparison of the predicted turbine performance with the experimental results in a previous study [3].

3. Performance evaluation

The turbine performance under steady flow conditions was evaluated by the torque coefficient $C_T$, input coefficient $C_A$, efficiency $\eta$, and flow coefficient $\phi$. The definition of these parameters are as follows:

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

Figure 4. Effect of setting angle of middle vane on turbine characteristics ($\sigma=1.08$).
\[ C_A = \frac{\Delta p Q}{\rho (v^2 + u^2) Av/2} = \frac{\Delta p}{\rho (v^2 + u^2)/2} \]  

\[ \eta = \frac{T_0 \omega}{(\Delta p Q)} = C_t/(C_A \phi) \]  

\[ \phi = \frac{v}{u} \]  

\[ \phi = 0.55 \]  
\[ \phi = 0.95 \text{ (at peak efficiency)} \]  
\[ \phi = 2.23 \]  

(a) Middle vane-less

\[ \phi = 0.55 \]  
\[ \phi = 0.95 \text{ (at peak efficiency)} \]  
\[ \phi = 2.23 \]  

(b) With middle vane (\( \delta = 65^\circ, \sigma = 1.08 \))

**Figure 5.** Streamlines around the rotor at the downstream side (\( 2r/D = 0.93 \)).

\[ \phi = 0.55 \]  
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**Figure 6.** Pressure distributions on the rotor surface at the downstream side.
where $A$, $u$, $v$ and $\rho$ denote the flow passage area \(=\pi D^2 (1-\nu^2)/4\), circumferential velocity at mean radius \((=R\omega)\), axial flow velocity \((=Q/A)\) and density of air, respectively.

4. Results and discussion

Figure 4 shows the effect of setting angle of middle vanes on turbine characteristics. Comparing the performances of the turbine with and without middle vanes, the turbine with middle vanes has a higher torque coefficient $C_T$ than the turbine without middle vanes. Similarly, the input coefficient $C_A$ is higher for the turbine with middle vanes. However, as the flow coefficient decreases, $C_A$ approaches to the value for the turbine without middle vanes. Therefore, the turbine with middle vanes showed a better efficiency than the turbine without middle vanes in a range of low flow coefficients, and the maximum efficiency also found increased as well. Moreover, the counter-rotating turbine with middle vanes showed a higher efficiency than the single rotor impulse turbine in the whole range of flow coefficients. Based on this result, a suitable setting angle $\delta$ of middle vanes was investigated. In Figure 4, $C_T$ increases with the increase of $\delta$ because the swirling component of the flow at inlet of the downstream side rotor is increased. Likewise, $C_A$ is increased with the increase of $\delta$. Therefore, the efficiency found increases with the increase of $\delta$, and in the case of $\delta=65^\circ$, the turbine shows the highest efficiency of $\eta=0.52$.

Figure 5 shows the streamline near the blade tip of the rotor at downstream side ($r=140$ mm), while
Figure 6 shows the pressure distributions on the rotor surface at downstream side. In a case of high flow coefficient ($\phi=2.23$), the air smoothly flows into the rotor at the downstream side with or without middle vanes. Therefore, they show a similar pressure distribution. The turbine efficiency at a high flow coefficient with middle vanes is slightly lower than that of without middle vanes because the influence of pressure loss due to the installation of middle vanes is significant. On the other hand, in a case of low flow coefficients ($\phi=0.55$), the flow inside the turbine greatly differs depending on the presence or absence of the middle vanes. In the turbine without middle vanes, the air flow collides with the leading edge of the rotor and largely exfoliates at the rotor pressure surface. Thus, the low pressure area (the encircled part) is generated near the blade tip on the rotor pressure surface. Furthermore, the high pressure area (the encircled part) is generated on the rotor suction surface because the flow separated from the leading edge of the rotor collides with the suction surface. As a result, a negative torque is generated to the rotor at the downstream side. On the contrary, in the turbine with middle vanes, the air smoothly flows into the rotor at the downstream side. Therefore, the turbine with middle vanes obtains a higher torque than that of without middle vanes because the low pressure area on the rotor pressure surface and the high pressure area on the rotor suction surface become small. In addition, the shock loss at inlet of the rotor at downstream side is greatly reduced. For these reasons, the turbine efficiency at a low flow coefficient with middle vanes becomes higher than that of without middle vanes.

Figure 7 shows the effect of middle vane solidity on the turbine characteristics. From figure, $C_T$ is increased with the increase of middle vane solidity $\sigma$ because the air from the middle vane flows into the downstream side rotor at an ideal angle that leads to the increase in the torque. Similarly, $C_A$ is increased with the increase of $\sigma$. Therefore, the efficiency increases with the increase of $\sigma$, and the turbine shows the highest efficiency of $\eta=0.52$ in case of $\sigma=1.08$.

5. Conclusions

The performance analysis of a counter-rotating impulse turbine for wave energy conversion was investigated by using CFD analysis. The results obtained are summarized as follows:

1. The flow collision at the inlet of the rotor at downstream side can be suppressed by the installation of middle vanes, and it may result in the increase in torque.
2. The efficiency in the range of low flow coefficients can greatly be improved by the installation of middle vanes, and the peak efficiency may increase as well.
3. The counter-rotating impulse turbine with middle vanes shows a higher efficiency than the impulse turbine with single rotor in the whole range of flow coefficients.
4. The suitable setting angle and solidity of middle vanes in the counter-rotating impulse turbine could be considered to be 65° and 2.02, respectively.

Actually an air turbine for wave energy conversion is operated in an irregular flow condition. Therefore, it is necessary to investigate the turbine performance under unsteady airflows as a future study, though a turbine performance under unsteady flow condition can solely be evaluated by a steady flow investigation.

Further, the turbine size used in this study is a model size, and it is smaller than the prototype. Thus, it seems that the efficiency of a prototype turbine could be higher than the predicted value obtained in this study.

References

[1] Takao M and Setoguchi T 2012 Int. J. of Rotating Machinery Vol. 2012 717398
[2] McCormick M E 1981 Ocean Wave Energy Conversion (New York: Wiley-Interscience)
[3] Takao M, Yamada K, Alam M M A, Okuhara S and Setoguchi T 2017 Turbomachinery Vol. 45 pp 30-36 (in Japanese)
[4] Setoguchi T, Takao M, Kinoue Y, Kaneko K, Santhakumar S and Inoue M 2000 Int. J. of Offshore and Polar Eng. Vol. 10 pp 145-152
[5] Setoguchi T, Santhakumar S, Maeda H, Takao M and Kaneko K 2001 Renewable Energy Vol. 23 pp 261-292