Effect of hub-to-tip ratio on the performance of bi-directional impulse turbine with flow collector for tidal energy conversion

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Abstract. The bi-directional impulse turbine and the bi-directional flow collector for tidal energy conversion is investigated in this paper. The bi-directional impulse turbine with fixed guide vanes is adopted because the turbine has a high efficiency and an advantage of maintenance. The turbine characteristics of the combined system of impulse turbine and collector are investigated experimentally by using the water tunnel. This system is proved to produce the power by a tidal flow experimentally. Three types of collector A, B and C are investigated, where the maximum radius of collector A is smaller than the ones of collector B and C. The velocity ratio of collector A is much smaller than the one for the cases of collector B and collector C, and the output power of collector A is very small compared to the other collectors. Nevertheless, the effect of flow collector is large because velocity ratio of collector A is much larger than the one without collector. Among three cases of 0.5, 0.6 and 0.7 of hub-to-tip ratio, the difference of turbine performance is not so large, but it is observed that the case for 0.5 of hub-to-tip ratio has inferior performance and the case for 0.6 of hub-to-tip ratio has the performance among them.

1. Introduction
The need for the use of natural energy has increased all over the world in recent times with attention on environmental issues. There are many natural energy that can be used as renewable energy, such as solar energy, wind energy, ocean energy and so on. For the ocean renewable energy, many types are focused on, such as ocean thermal energy, ocean current, wave motion energy, tidal energy and so on. Among them, many studies have already been conducted on the wave energy conversion system [1-6]. For the tidal energy conversion, the system using a dam is common, where, by dividing an arm of the sea by the dam, bi-directional flow can be made. Installing the water turbine directly in the tidal flow, however, is another possible way for the tidal energy conversion. This water turbine should work on the situation of the bi-directional flow according to the tidal flow. In this study, it is considered that the turbine system does not change the orientation but the turbine system itself can work in the bi-directional flow. That system does not need the orientation change.

In the case of installing the water turbine directly in the tidal flow, collecting the flow is very important because it is generally known that the power available from a stream of water is proportional to the cube of the free stream velocity of the current. For the wind turbine, the technique of accelerating wind speed using a diffuser with a brim has been developed [7,8]. As this special diffuser system is for one-way flow, the bi-directional diffuser is considered in this research for the flows that periodically change the direction. For the turbine system, the bi-directional impulse turbine
Figure 1. Experimental apparatus.

Figure 2. Bi-directional impulse turbine.

Figure 3. Three types of bi-directional flow collector (diffuser).
with fixed guide vanes [3], developed for air flow wave energy conversion system, has been adopted. The effects of hub-to-tip ratio as well as the shape of the collector are investigated in this paper.

2. Experimental apparatus and procedure

Figure 1 shows the test rig for the water turbine and the diffuser. The water tunnel at Institute of Ocean Energy, Saga University (IOES) is used for this study. The depth and the width of water flow are 0.7 m and 1.0 m, respectively. The mainstream velocity of water flow \( v \) is set as 0.8 m/s and the accurate value of \( v \) is measured by a Pitot-tube. The axial velocity in the turbine \( v_a \) is also measured by another Pitot-tube. The impulse turbines with hub-to-tip ratio of 0.5, 0.6 and 0.7 were placed at the center of the test section and tested at a constant rotational speed, i.e., constant angular velocity \( \omega \) under steady condition. The pressure drop in the turbine \( \Delta P \), the output torque \( T \) and the axial force on the turbine and the diffuser \( F_a \) are measured. The casing radius \( r_c \) in the turbine is 85 mm.

Figure 2 shows the outline of the impulse turbine with fixed guide vanes for the water flow test rig. The turbine geometry is symmetrical in the rotor centerline so as to rotate the same direction for a bi-directional flow. Tested rotor is an impulse type rotor which number \( z_r \) is 26 and the inlet and outlet angle \( \gamma \) is 50 deg., respectively. Tip clearance of the rotor is 0.3 mm. Tested guide vanes consist of thin metal plate and 26 blades which has dimensions such as the setting angle \( \theta \) of 37.5 deg., the circular arc radius \( R_a \) of 23.8 mm, angle of the arc \( \delta \) of 52.5 deg., respectively.

Figure 3 shows the photographs of three types of the diffuser, that is, flow collector A, flow collector B and flow collector C, respectively. The spreading angle of A and B in the straight part is 31 degree, whereas the one of collector B is 45 degree, respectively. The maximum radii are 1.2 times of rotor diameter for collector A, while 2.0 times of rotor diameter for collector B and collector C, respectively. The ratio of the cross section area of collector to the test section area is 0.29 for the collector B or C. The investigation of the effect of this area ratio is desired because the value of 0.29 is not fully small.

The experimental results are converted to the dimensionless values as follows.

\[
\phi = \frac{u}{\dot{\omega}} \quad \text{(1)}
\]
\[
\psi = \frac{\Delta P}{\rho u^2 / 2} \quad \text{(2)}
\]
\[
\tau = \frac{T \omega}{\rho u^3 A / 2} \quad \text{(3)}
\]
\[
\eta = \frac{T \omega}{\Delta P Q} = \frac{\tau}{\phi \psi} \quad \text{(4)}
\]
\[
C_f = \frac{F_a}{\rho u^2 A / 2} \quad \text{(5)}
\]

Where \( \phi \) is flow coefficient, \( \psi \) is pressure drop coefficient, \( \tau \) is torque coefficient, \( \eta \) is turbine efficiency and \( C_f \) is axial force coefficient, respectively.

3. Experimental results and discussions

3.1. Effect of flow collector shape

The effect of flow collector is investigated for the case of \( \nu = 0.7 \). Figures 4 to 8 show the experimental results of velocity ratio, pressure drop coefficient, turbine efficiency, axial force coefficient, torque coefficient, respectively. Also, Figure 9 shows the predicted output power \( P_{FS} \), which was evaluated for the case of full scale, that is, the rotor diameter is 10 m and tidal flow velocity is 1.7 m/s. The marks of circle, square, triangle and diamond in the figures show the values for the cases of collector
(diffuser) A, collector (diffuser) B and collector (diffuser) C, and without collector (diffuser), respectively.

Figure 4. Effect of collector shape on velocity ratio.

Figure 5. Effect of collector shape on pressure drop coefficient.

Figure 6. Effect of collector shape on turbine efficiency.
Figure 7. Effect of collector shape on axial force coefficient.

Figure 8. Effect of collector shape on torque coefficient.

Figure 9. Effect of collector shape on predicted full-scale power.
Figure 10. Effect of hub ratio on velocity ratio for collector C.

Figure 11. Effect of hub ratio on pressure drop coefficient for collector C.

Figure 12. Effect of hub ratio on turbine efficiency for collector C.
Figure 13. Effect of hub ratio on axial force coefficient for collector C.

Figure 14. Effect of hub ratio on torque coefficient for collector C.

Figure 15. Effect of hub ratio on predicted full-scale power for collector C.
For the case of collector A, the velocity ratio is much smaller than the ones of collector B and C. Also, the output power of collector A is very small compared to the other collectors. Nevertheless, the effect of flow collector is large because velocity ratio of collector A is much larger than the one without collector in Figure 4. The value of the maximum radius is considered to be the important parameter in order to collect the flow.

For the case of the collector C, the axial force coefficient of Figure 7 is much larger than the one for the cases of the collector A and the collector B. That fact shows that the constructive cost of the collector C becomes larger than the ones of the other collectors. The installation of the brim is considered to be the important point to reduce the flow resistance for collector B.

3.2. Effect of hub-to-tip ratio

In terms of the effect of hub-to-tip ratio, Figures 10 to 14 show the experimental results of velocity ratio, pressure drop coefficient, turbine efficiency, axial force coefficient, torque coefficient, respectively. Also, Figure 15 shows the predicted output power $P_{FS}$, which was evaluated for the case of full scale.

Among three cases, the difference is not so large, but it is observed that the case for $\nu=0.5$ has inferior performance and the case for $\nu=0.6$ has superior performance. In Figure 15, the maximum values of $P_{FS}$ are 95.0 kW at $\phi=1.45$ for $\nu=0.7$, 99.6 kW at $\phi=1.50$ for $\nu=0.6$, 83.0 kW at $\phi=1.45$ for $\nu=0.5$, respectively. Therefore, the value of hub-to-tip ratio of 0.6 is best in this investigation.

4. Conclusions

The combined system of the bi-directional impulse turbine and the bi-directional flow collector for tidal energy conversion is investigated in this paper. Conclusions are summarized as follows.

(1) The bi-directional impulse turbine with fixed guide vanes is adopted in this study because the turbine has high efficiency and does not need the orientation change.

(2) The velocity ratio of collector A is much smaller than the one for the cases of collector B and collector C, which shows that the output power of collector A is very small compared to the other collectors. Nevertheless, the effect of flow collector is large because velocity ratio of collector A is much larger than the one without collector. The value of the maximum radius is considered to be the important parameter in order to collect the flow.

(3) Among three cases of 0.5, 0.6 and 0.7 of hub-to-tip ratio, the difference of turbine performance is not so large, but it is observed that the case for $\nu=0.5$ has inferior performance and the case for $\nu=0.6$ has the best performance among them. The maximum values of $P_{FS}$ are 99.6 kW at $\phi=1.50$ for $\nu=0.6$.

5. References

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### 6. Nomenclature

| Symbol | Description                                                                 | Unit          |
|--------|----------------------------------------------------------------------------|---------------|
| $A$    | Cross sectional area at the turbine ($= \pi r_c^2$)                        | m$^2$         |
| $C_f$  | Axial force coefficient                                                   | [-]          |
| $F_a$  | Axial force of turbine and collector                                      | N            |
| $\Delta P$ | Pressure drop in the turbine                                            | Pa           |
| $R_g$  | Radius of circular arc of guide vane                                      | m            |
| $r_c$  | Characteristic length ($= r_c$)                                           | m            |
| $r_b$  | Radius of brim in the collector                                           | m            |
| $r_h$  | Radius at the casing in the turbine                                       | m            |
| $u^*$  | Characteristic velocity ($= r^* \omega$)                                  | m/s          |
| $v$    | Mainstream velocity                                                       | m/s          |
| $v_a$  | Axial velocity in the turbine ($= Q/A$)                                    | m/s          |
| $z_r$  | Rotor blade number                                                        | [-]          |

### Acknowledgement

This research is supported by Saga Prefecture, Japan, which fund name is the Saga prefecture fund for the introduction of renewable energy. The authors greatly acknowledge the support stated above.