2003 유체기계 연구개발 발표회 논문집, pp. 3~12
선문대학교 아산캠퍼스 2003. 12. 5~6

Present State of Self-Rectifying Air Turbines for Wave Energy Conversion

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Abstract–This paper reviews the present state of the art on the self-rectifying air turbines, which could be used for wave energy conversion. The overall performances of the turbines under irregular wave conditions, which typically occur in the sea, have been evaluated numerically and compared from the viewpoints of the starting and running characteristics. The types of turbine included in the paper are as follows: (a) Wells turbine with guide vanes (WTGV); (b) turbine with self-pitch-controlled blades (TSCB); (c) biplane Wells turbine with guide vanes (BWGV); (d) impulse turbine with self-pitch-controlled guide vanes (ISGV) and (e) impulse turbine with fixed guide vanes (IFGV). As a result, under irregular wave conditions it is found that the running and starting characteristics of the impulse type turbines could be superior to those of the Wells turbine. Moreover, the authors have explained the mechanism of hysteretic behavior of the Wells turbine and the necessity of links for improvement of the performance of ISGV.

Keywords: Fluid machinery, Wells turbine, impulse turbine, wave energy conversion

1. INTRODUCTION

Several of the wave energy devices being studied under many wave energy programs make use of the principle of an oscillating water column (OWC). In such wave energy devices an oscillating water column due to wave motion is used to drive an oscillating air column, which is converted into mechanical energy. The energy conversion from an oscillating air column can be achieved by using a system of non-return valves for rectifying the airflow, together with a conventional turbine. However, such a system is complicated and difficult to maintain and the average cycle efficiency in an oscillating airflow is likely to be considerably smaller.

The non-return valves can be eliminated by the use of a self-rectifying air turbine, which inherently provides a unidirectional rotation for an alternating airflow. The Wells turbine [Gato and Falcão, 1988; Inoue et al., 1986a, 1986b, 1988; Kaneko et al., 1986; Raghunathan, 1995; Raghunathan et al., 1982, 1989; Setoguchi et al., 1986] is of this type and is one of the simplest and probably the most economical turbines for wave energy conversion. However, according to the previous studies, the Wells turbine has inherent disadvantages: lower efficiency, poorer starting characteristics and higher noise level in comparison with conventional turbines. Consequently, in order to overcome these weak points, a number of self-rectifying air turbines with different configurations have been proposed and improved over decades [Akabane et al., 1984; Inoue et al., 1989; Katsuhara et al., 1987; Kim, et al., 2001; Richard and Weiskopf, 1986; McCormick et al., 1992, 1993; Setoguchi et al., 1990, 1993, 1996, 2000; Takao et al., 1997]. However, a comparison of all of these characteristics has yet to be made.

This paper reviews the present state of the art of the self-rectifying air turbines for wave energy conversion. The types of turbine included in the review are summarized as follows.

Wells type turbines:
- Wells turbine with guide vanes
- Turbine with self-pitch-controlled blades
- Biplane Wells turbine with guide vanes
- Contra-rotating Wells turbine

Impulse type turbine:
- Impulse turbine with self-pitch-controlled guide vanes
- Impulse turbine with fixed guide vanes
- McCormick counter-rotating turbine

Radial turbine
Cross flow turbine
Savonius turbine

Furthermore, the performances of turbines in connection with OWC under irregular wave conditions could be used for wave power conversion in the near future. Therefore, they have been evaluated numerically and compared from the viewpoints of the starting and running characteristics. As a result, a suitable choice of the self-rectifying air turbine for wave energy conversion has been suggested.

The authors have also explained the mechanism of hysteretic behavior of the Wells turbine and the necessity of links for improvement of the performance of impulse turbine with self-pitch-controlled guide vanes.

2. WAVE ENERGY CONVERSION SYSTEM

The basic wave energy conversion process can be stated in very general terms as follows: the force (or torque) of an incident wave causes relative motion between an absorber and a reaction point, which drive a working fluid through a generator prime mover. The periodic nature of ocean waves dictates that this relative motion will be oscillatory and have a frequency range of 3 to 30 cycles per minute, which is much less than the hundreds of revolutions per minute required for economic/conventional electric power generator. A variety of working fluids and prime movers are employed to convert these slow-acting, reversing wave forces into high-speed, unidirectional rotation of a generator shaft.

Primary conversion of wave energy is attained by an oscillating system, either a floating body, an oscillating solid member or oscillating water within a structure. Potentially, the most successful device used in the harnessing on wave energy has been the oscillating water column (OWC) wave energy converter. The OWC chamber, either floating or bottom standing, with the immersed end opened to the action of the sea. A reciprocating airflow is created by the action of the free surface of the water within the chamber. The conversion of this airflow into mechanical energy may be achieved by means of a number of devices.

To produce unidirectional flow, a reciprocating airflow may be rectified by a series of non-return valves. This unidirectional flow may be used to drive a conventional turbine such as a Francis turbine. An example of practical OWC wave energy converter using the conventional turbine is the navigation buoy (Fig. 1). Based on work by the Japanese wave-energy pioneer Mr. Y. Masuda, more than one thousand wave-powered navigation buoys have been produced since 1965 and marketed worldwide. Some of them have been in operation for more than 30 years. The conventional air turbine is used for the secondary energy conversion in the OWC wave energy converter. However, the

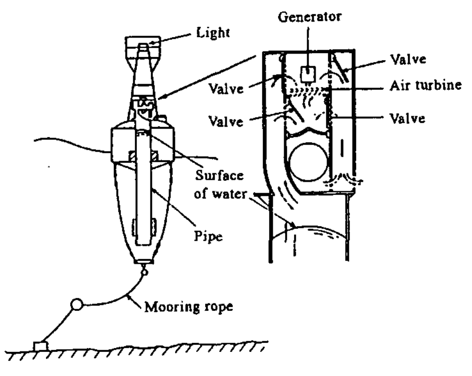


Fig. 1 Outline of navigation buoy

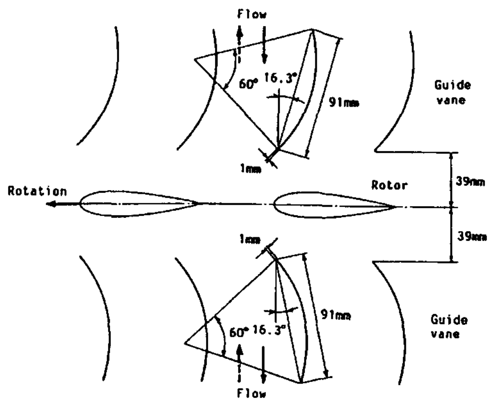


Fig.2 Wells turbine with guide vanes: WTGV

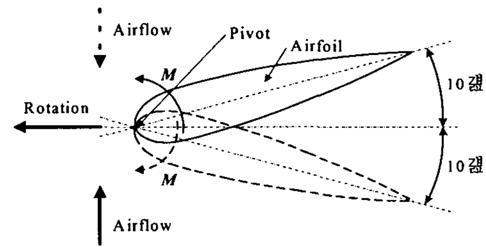


Fig. 3 Turbine using self-pitch-controlled blades: TSCB

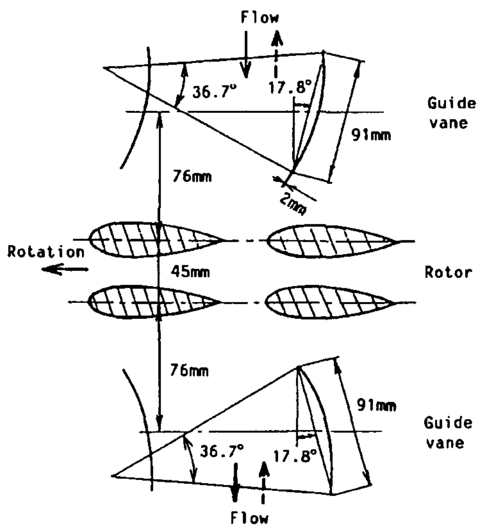


Fig. 4 Biplane Wells turbine with guide vanes: BWGV

Airflow rectification system with non-return valves is complicated and difficult to maintain. Furthermore, such a system cannot be adopted in a large-scale wave energy device because the valve becomes large.

On the other hand, no rectifying valve system is required if the self-rectifying air turbine is used. Many kinds of the self-rectifying air turbines have been proposed and improved over decades.

3. SELF-RECTIFYING AIR TURBINES FOR WAVE ENERGY CONVERSION

3.1 Wells Type Turbine

Figure 2 shows the Wells turbines with guide vanes [Inoue et al., 1985; Gato and Falcão, 1990: Setoguchi et al., 1998]. The turbine (turbine diameter of 1.7m, 2 tandem turbines, NACA0021, 8 blades per rotor, rated output of 30kW × 2) was adopted for the project "Mighty Whale" organized by JMSTEC, Japan [Miyazaki, 1993; Washio et al., 2000].

Figure 3 illustrates the turbine with self-pitch-controlled blades [Inoue et al., 1989; Kim et al., 2001; Takao et al., 1997]. A turbine blade is set on the hub by a pivot located near the leading edge that enables it to oscillate between two prescribed setting angles. As an airfoil set at a certain angle of incidence generates the pitching moment M about a pivot, the turbine blades can oscillate between two setting angles by themselves according to the flow direction, as shown in the figure. The turbine may be connected with the "Azores Pilot Plant" (rated output of 400kW) supported from the JOULE II [Falcão et al., 1993], where the turbine has active pitch-controlled blades.

The project using the biplane Wells turbine (Fig. 4) is making progress in Islay, U.K. [Falcão et al., 1993], where the guide vanes are not used for the turbine.

The contra-rotating Wells turbine [Beattie and Raghunathan, 1993; Raghunathan, 1995] is shown in Fig. 5. This is installed in the LIMPET system, Islay, U.K. [Alcorn and Beattie, 2001], which is the world's first commercial wave power station (capacity of 500kW, turbine diameter of 2.6m, 2 turbines, NACA0012, 7 blades per rotor). However, detailed information of the turbine characteristics has not been clarified.

3.2 Impulse Turbine

Although a number of impulse turbines have been presented [Kaneko et al., 1991], their performances have not been published.

Figure 6 illustrates the impulse turbine with self-pitch-controlled guide vanes. The turbine has guide vanes on both sides of the rotor so as to operate efficiently in an oscillating flow. They are set by pivots on the casing wall. The pivots are located at the end of the guide vane chord close to the rotor so that guide vanes are permitted to move around the pivot by aerodynamic moment induced by a move of oscillating airflow. This turbine (turbine diameter of 1.0m) has been constructed by NIOT, India [Santhakumar, 1996;

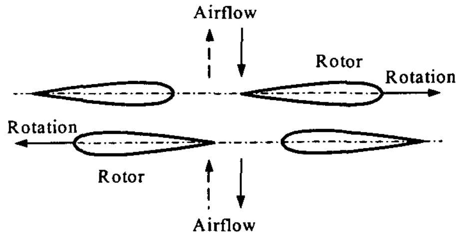


Fig. 5 Contra-rotating Wells turbine

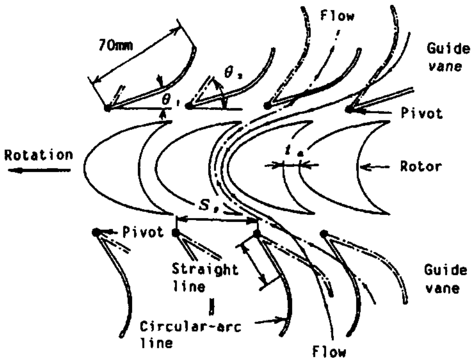


Fig. 6 Impulse turbine with self-pitch-controlled guide vanes connected by link motion: ISGV

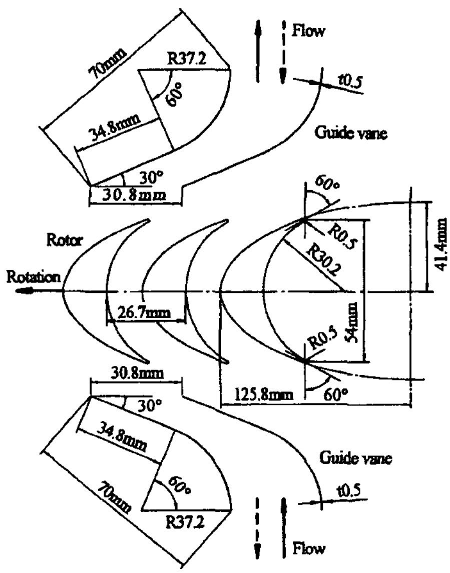


Fig. 7 Impulse turbine with fixed guide vanes: IFGV

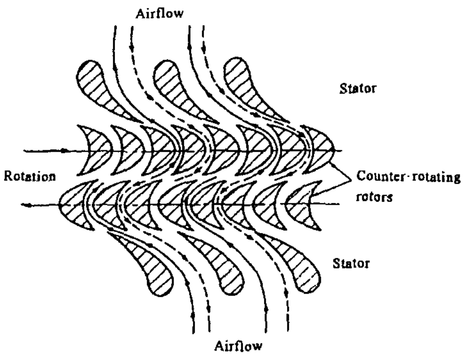


Fig. 8 McCormick counter-rotating turbine

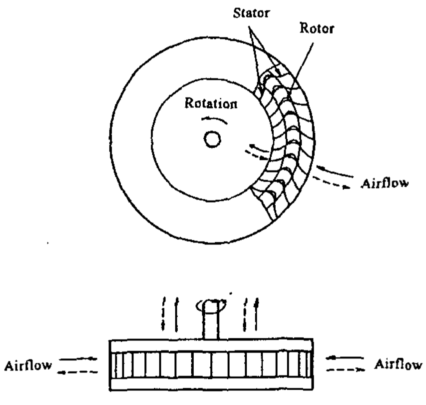


Fig. 9 Radial turbine

Santhakumar et al., 1998].

Figure 7 shows the impulse turbine with fixed guide vanes. This is being planned to be constructed in India, China and Ireland [Thakker et al., 1999].

The outline of the McCormick turbine [McCormick, 1981] is sketched in Fig. 8. The prototype model of the McCormick turbine (30 nozzles each turbine, 60 blades per rotor, turbine diameter of 0.99m, 450 rpm nominal design, 1200 rpm with gears) was constructed and tested [Richard et al., 1986], and average efficiencies near 0.3 appear to have been attained. The disadvantage may be the balance of gearing cost and its noise generation may be a severe problem at specific sites.

3.3 Radial Turbine

Figure 9 shows the impulse type radial turbine. The efficiency of the turbine seems to be higher according to the previous studies [McCormick et al., 1992; McCormick and Cochran, 1993], though detailed turbine characteristics are not found in the literature. However, according to a recent research, the efficiency is not so good [Setoguchi et al., 2002].

3.4 Cross Flow Turbine and Savonius Turbine

Many versions of cross flow turbine and Savonius turbine have been proposed so far. In order to obtain some fundamental data for the turbines, the tests were performed under steady flow conditions [Akabane et al., 1984; Katsuhara et al., 1987]. However, in general, they are inferior to Wells type turbines in the starting and running characteristics [Kaneko et al., 1991].

4. SUITABLE TURBINE FOR WAVE ENERGY CONVERSION

One of the objectives of this chapter is to compare the performances of turbines which could be used for wave power conversion in the near future. Here we should note that the performance of the wave power converter depends on the energy absorption efficiency of OWC , which is closely related to the pressure difference across the turbine, as well as the turbine efficiency. Therefore, the performances of turbines in connection with OWC under irregular wave conditions are evaluated numerically by using a quasi-steady analysis [Inoue et al., 1986a; Setoguchi et al., 1993], and compared from the viewpoints of the starting and running characteristics.

The types of turbine included in this comparative study are as follows: (a) Wells turbine with guide vanes named WTGV in this paper, Fig. 2 [Inoue et al., 1985; Gato and Falcão, 1990]. (b) Turbine with self-pitch-controlled blades named TSCB, Fig. 3 [Inoue et al., 1989; Kim et al., 2001; Takao et al., 1997]. (c) Biplane Wells turbine with guide vanes named BWGV, Fig. 4 [Setoguchi et al., 1990]. (d) Impulse turbine with self-pitch-controlled guide vanes named ISGV, Fig. 5 [Setoguchi et al., 1993, 1996]. (e) Impulse turbine with fixed guide vane named IFGV, Fig. 6 [Setoguchi et al., 2000]. In this study the contra-rotating Wells turbine was not adopted because the turbine characteristics have not been clarified so far.

4.1 Details of Turbine Geometries

The details of turbines adopted in the study are as follows: (a) WTGV (Fig. 2); blade profile: NACA0020, tip diameter: D = 298mm, hub-to-tip ratio: ν = 0.7, aspect ratio: AR = 0.5, chord length of rotor: l_r = 90 mm, rotor solidity at mean radius r_R: σ_{rR} = 0.67 and guide vane solidity at mean radius: σ_{gR} = 1.25. (b) TSCB (Fig. 3); NACA0021, D = 298mm, ν = 0.7, AR = 0.6, l_r = 75 mm, σ_{rR} = 0.75 and preset angle of 10°. (c) BWGV (Fig. 4); NACA0020, D = 298mm, ν = 0.7, AR = 0.5, l_r = 90 mm, σ_{rR} = 0.45 and AR = 1.25. (d) ISGV (Fig. 6); D = 298mm, ν = 0.7, t_a/S_r = 0.4 (t_a: width of flow path at mean radius, S_r: rotor blade space at r_R) (see Fig. 6), l_r = 54 mm, inlet (or outlet) angle of rotor: γ = 60°, σ_{rR} = 2.02, σ_{gR} = 2.27, setting angle of upstream guide vane: θ_1 = 17°, setting angle of downstream guide vane: θ_2 = 72.5° and sweep angle of rotor: λ = -7.5°. (e) IFGV (Fig. 7); D = 298mm, ν = 0.7, t_a/S_r = 0.4, l_r =

54 mm, γ = 60°, σ_{rR} = 2.02, σ_{gR} = 2.27, guide vane setting angle of 60° and λ = -7.5°. Note here that the configurations considered for these turbines are the ones found to be most promising in previous studies [Kim et al., 2001; Setoguchi et al., 1990, 1996, 1998, 2000]. Furthermore, all of them can start [Inoue et al., 1986a] by themselves.

4.2 Turbine Characteristics under Steady Flow Conditions

Turbine characteristics under steady flow conditions were obtained by model testing, and evaluated with torque coefficient C_T, input power coefficient C_A and flow coefficient ϕ, which are defined as:

$$C_T = T_o / \{\rho_a (v_a^2 + U_R^2)\, b l_r z r_R / 2\} \tag{1}$$

$$C_A = \Delta p Q / \{\rho_a (v_a^2 + U_R^2)\, b l_r z v_a / 2\} \tag{2}$$

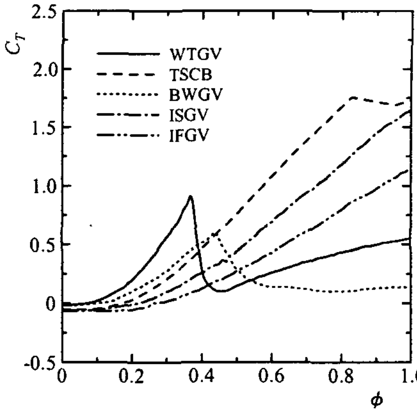
$$\phi = v_a / U_R \tag{3}$$

where T_o : output torque, ρ_a: density of air, b: rotor blade height, U_R: circumferential velocity at r_R, v_a: mean axial flow velocity, r_R: mean radius, z: number of rotor blades, Δp: total pressure drop between settling chamber and atmosphere, and Q: flow rate. The test Reynolds number based on blade chord was about 2.0×10^5 at peak efficiency for WTGV, TSCB and BWGV (Wells type turbines), and approximately 0.4×10^5 for ISGV and IFGV (impulse type turbines).

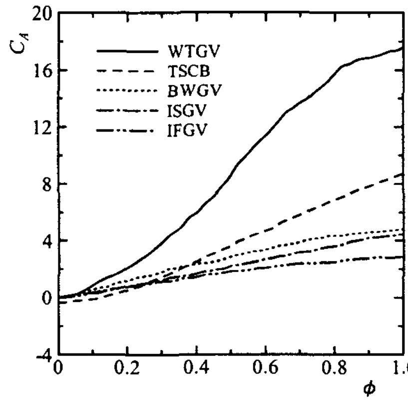
Figure 10(a) shows C_T-ϕ characteristics for the five turbines. Abrupt decreases in C_T characteristics due to rotor stall are observed for all the Wells type rotors such as WTGV, TSCB and BWGV. The value of ϕ where rotor stall starts is the largest for TSCB and the value of ϕ at C_T = 0 for TSCB is larger than other Wells type turbines. This is because relative inflow angle for rotor is lower than the case that rotor blades are fixed at 90 degrees of stagger angle. On the other hand, for both the impulse type rotors such as ISGV and IFGV, the value of C_T increases with increasing ϕ, and the value of C_T at region of large ϕ is larger than the Wells type turbines. The value of ϕ at C_T = 0 is larger than the Wells type turbines.

Figure 10(b) shows C_A-ϕ characteristics for the five turbines. The value of C_A for WTGV is the largest in the five turbines at any flow coefficient. This means that the pressure in air chamber is higher than other turbines and should be taken care for the maintenance of bearing because of larger thrust force. On the other hand, for TSCB, BWGV, ISGV and IFGV, the value of C_A is rather small, especially for ISGV and IFGV. This means that the pressure increase in the air chamber is small when impulse type turbines are adopted for wave power generator devices.

Concerning the turbine efficiency η under steady flow conditions, we can easily take η-ϕ characteristics from Fig. 10 because of $\eta = C_T / (C_A \phi)$. The peak efficiencies of five turbines were 0.492 (WTGV), 0.496 (TSCB), 0.534 (BWGV), 0.564 (ISGV) and 0.390 (IFGV). However, it should be noted that η does not give the useful information about the suitable turbine for wave power conversion. This is because turbine



(a) Torque coefficient



(b) Input coefficient

Fig. 10 Turbine characteristics under steady flow conditions

characteristics depend on the efficiency of the air chamber, i.e., the ratio of power of OWC and incident wave power.

4.3 Simulation of Turbine Characteristics under Irregular Flow Conditions

Although the characteristics of the turbine used in the study were clarified under steady flow conditions as shown above, the results have not yet provided sufficient information about the suitable turbine for wave power conversion. This lack of information exists because the performance of the wave power converter also depends on the OWC's energy absorption efficiency, which is closely related to the pressure difference across the turbine, namely to the C_A-characteristics of the turbine.

In order to clarify the suitable turbine for wave energy conversion, it is necessary to evaluate the turbine

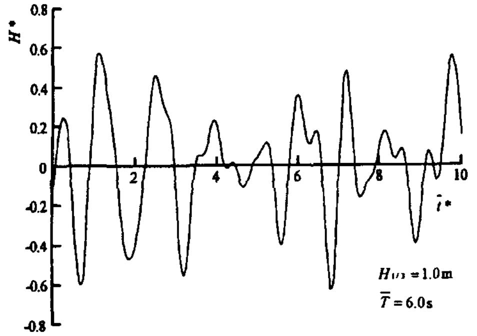


Fig. 11 Test irregular wave

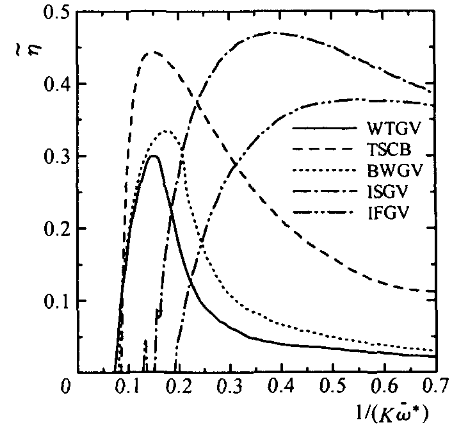


Fig. 12 Comparison of conversion efficiency of wave energy

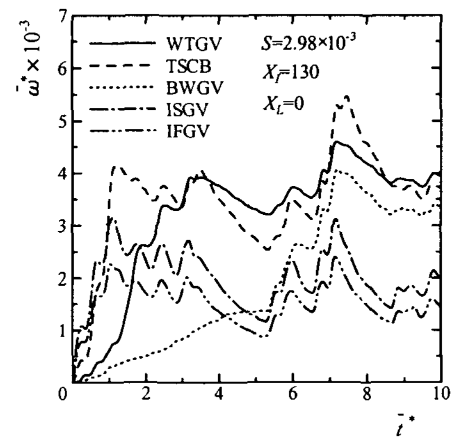


Fig. 13 Starting characteristics under irregular wave conditions

characteristics in connection with an OWC under irregular flow conditions.

For an exact evaluation of the energy absorption efficiency of OWC, it is important to take into account the inertia of water in the OWC, the resistance characteristics of a flow passage, the momentum flux in the water column and so on. However, for evaluation of the turbine suitable for wave power conversion, the inertia, resistance and so forth are ignored, so that the efficiency of air chamber can be calculated only when considering the turbine's C_A-characteristics, without providing the OWC's specified configurations. Hence, the method presented by Setoguchi et al. (2000) is used for the simulation, i.e., the energy conversion efficiency of the system $\tilde{\eta}$ is obtained by multiplying the efficiency of air chamber $\tilde{\eta}_c$ and turbine efficiency $\tilde{\eta}_t$. Note here that the objective of this study is to compare the performances of the turbines relatively. In this case, it is considered that the above method is suitable to evaluate the chamber's efficiency.

For evaluation of the starting characteristics of the turbine, behavior of the turbine rotor from the rest (i.e., $\bar{t}^* = t/\bar{T} = 0$, $\bar{t}^*$: dimensionless time under irregular wave condition, t: time, $\bar{T}$: mean period of wave under irregular wave condition) to $\bar{t}^* = 10$ is obtained by solving the equation of motion for a rotating system of the turbine under irregular wave conditions through the use of the Runge-Kutta-Gill method [Inoue et al., 1986a; Setoguchi et al., 1993]. The simulation of the starting characteristics was performed as a load-free condition (i.e., $T_L = 0$, T_L: loading torque).

The irregular wave test adopted is shown in Fig. 11, where H^* is non-dimensional incident wave ($=H/H_{1/3}$, H: incident wave height, $H_{1/3}$: significant wave height). This is based on the ISSC (International Ship Structure Congress) spectrum, which is commonly done in the field of ocean engineering [ISSC, 1964]. The significant wave height $H_{1/3}$, the wave mean period $\bar{T}$ and the air chamber area ratio m ($=A_t/A_c$, A_t: turbine flow passage area, A_c: air chamber cross-sectional area) are 1.0m, 6.0s and 0.0234, respectively. In these calculations, the flow condition is assumed to be quasi-steady, because the validity of quasi-steady analysis in the prediction of turbine characteristics under unsteady flow conditions has been demonstrated [Inoue et al., 1986a; Setoguchi et al., 1996; Takao et al., 1997].

In the calculations, the values of C_T and C_A shown in Fig. 10 can be used here. For simplifying the numerical simulation of ISGV and TSCB, pitch angle of guide vanes and rotor blades are assumed to change at the same time when axial velocity changes from positive to negative (or from negative to positive). The validity of this assumption was shown by previous studies [Setoguchi et al., 1996; Takao et al., 1997], in which the calculation of mean efficiency and starting characteristics under unsteady flow condition agreed with the experimental data from a viewpoint of engineering use.

Figure 12 shows the comparison of conversion efficiency of wave energy. For the impulse type turbines, conversion efficiency is quite high at region of large $1/(K\bar{\omega}^*)$ {$=V/U_R$, K: non-dimensional period, V: reference velocity, and $\bar{\omega}^*$: non-dimensional angular velocity under irregular flow condition ($=\omega\bar{T}$, ω: angular velocity of rotor)} compared with the Wells type turbines. Especially, the value of maximum efficiency for ISGV is about 47 % and over 15 % larger than that of WTGV, which is now mainly used for wave power conversion. Since rotor stall does not occur for the impulse type turbine as shown in Fig. 10(a), torque can be obtained within a comprehensive region of flow coefficient. Moreover, the difference of the efficiency of the air chamber is considered to be one of the causes that the energy conversion efficiency of wave power conversion using the impulse type turbine is higher than that efficiency using the Wells type turbine. This is because the maximum value of OWC height for IFGV is larger than that for WTGV due to a smaller pressure in the chamber as shown in Fig. 10(b). Although ISGV has a disadvantage of maintenance of pivots, even for IFGV, where guide vanes are fixed for simple configuration, the maximum efficiency of IFGV is larger than that of WTGV by about 6 %. On the other hand, the efficiency of TSCB is also considerably higher than that of WTGV. However, it should be noted that maintenance of TSCB is more difficult than that of ISGV because TSCB is using the pitch-controlled rotor blades which is rotating around the shaft at high speed, though ISGV has the pitch-controlled guide vanes. Therefore, there is no doubt that the impulse type turbine has better running characteristics than the Wells type turbine. Here note that this tendency is almost the same for any $H_{1/3}$.

The starting characteristics for five turbines are shown in Fig. 13, where S: non-dimensional frequency, X_I: non-dimensional moment of inertia, X_L: non-dimensional loading torque. The impulse type turbine can start in a very short time. This fact means that a generating time of a generator with the impulse turbine is longer than that of the Wells turbine. Furthermore, the rotational speed at operation is much smaller than those of the Wells type turbines. These are because the torque coefficient C_T of the impulse type turbine is higher than that of the Wells turbine, and the flow coefficient at load-free condition for the impulse turbine is larger than the Wells type turbines as shown in Fig. 10(a). Therefore, when considering the impulse type turbines, it is possible to design an excellent turbine with low operational speed, which is desirable from the viewpoints of noise reduction and mechanical advantage.

5. REMARKS FOR TURBINE CHARACTERISTICS

5.1 Wells Turbine

The performance of the Wells turbine has a hysteretic loop in which the values of C_A and C_T in the accelerating flow are smaller than in the decelerating flow. Figure 14 shows the hysteretic characteristics of the input efficient C_A under sinusoidal flow condition, where α_R is the angle of attack at mean radius. This behavior cannot be explained by the dynamic stall of an airfoil because the hysteretic loop is opposite to that of the dynamic stall. Recently, the mechanism of hysteretic behavior has been elucidated by an unsteady 3-dimensional Navier-Stokes numerical simulation [Setoguchi et al., 2003].

Figure 15 shows an illustration of the flow structure obtained by the present numerical simulation. At high angle of attack a separation vortex appears on the blade suction surface in the hub side to reduce the blade circulation, because of the excessive angle of attack near the hub. A strong downward flow is induced by the separation vortex near the trailing edge. It brings about the clockwise vortical wake flow, which affects the separation on the adjacent blade suction surface. The intensity of the vortical flow varies in the accelerating and the decelerating flow process for the following reason.

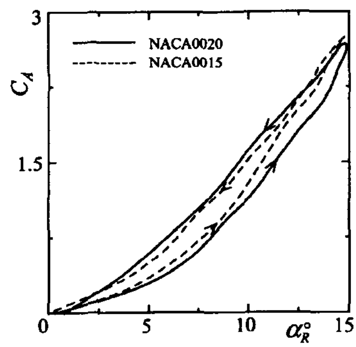


Fig. 14 Hysteretic characteristics of Wells turbine

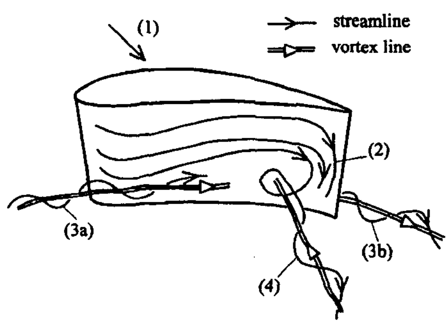


Fig. 15 Illustration of the flow structure in blade suction side

In the accelerating flow process, as the blade circulation increases, vortices opposite to the blade circulation are shed from the trailing edge according to Kelvin's theorem. The stronger vortices are shed at larger radii because the blade circulation increases more than at a smaller radius. Then, the clockwise trailing vortices are generated according to Helmholtz's theorem. Therefore, the clockwise vortical flow is intensified by these vortices. In the decelerating flow process where the blade circulation decreases, the shed vortices are in the same direction as the blade circulation. They form counter-clockwise trailing vortices which suppress the vortical wake flow. Since the stronger vortical wake flow enlarges the separation on the suction surface of the adjacent blade, the performance in the accelerating flow process becomes lower than in the decelerating flow process.

5.2 Impulse Turbine with Self-Pitch-Controlled Guide Vanes

A number of studies were conducted on impulse turbine with self-pitch-controlled guide vanes (Fig. 6) over a period of time. In previous type of turbine,

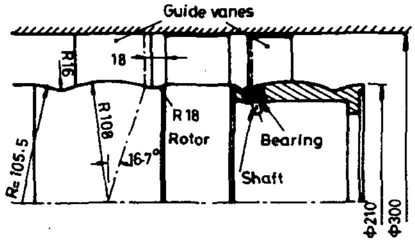


Fig. 16 Meridional section of impulse turbine without kinks

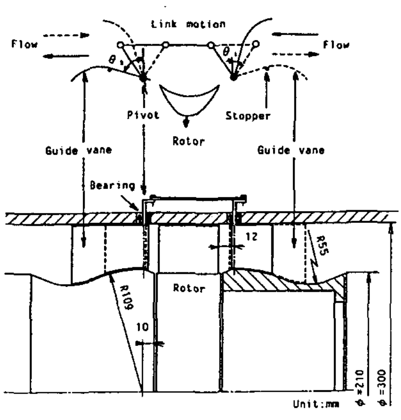
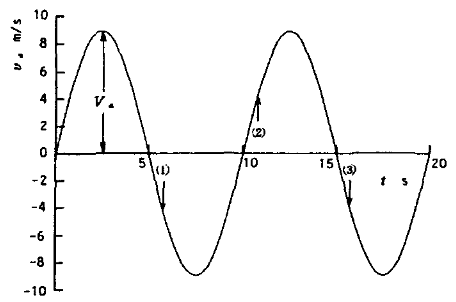


Fig. 17 Outline of impulse turbine with *self-pitch-controlled guide vanes* connected by links

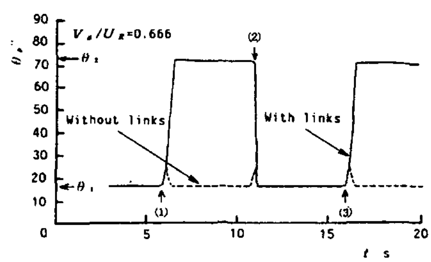
the guide vanes were set on a spherical hub so as to keep the hub clearance constant while the vane rotates, as shown in Fig. 16. However, the performance of this turbine was not superior to what we expected in a reciprocating flow. This is due to the deterioration of the diffuser guide vanes in the process of changing axial flow velocity [Setoguchi et al., 1993].

A new type of impulse turbine with self-pitching links guide vanes [Setoguchi et al., 1996] was proposed to overcome this difficulty. Figure 17 shows the meridional section of the new type of turbine, where θ_1 and θ_2 are nozzle and diffuser setting angles, respectively. The bearings of the vane are set in the outer casing. Every vane on one side of the rotor is connected by a link outside the casing to a vane on the other side of the rotor. That is, any pair of vanes on either side of the rotor is constructed to rotate together. This is because the aerodynamic moment generated by the upstream guide vanes in the process of nozzle action may be utilized to prevent the insufficient movement of downstream guide vanes at diffuser action.

The variation of axial flow velocity v_a with time is shown in Fig. 18(a). Under this condition, in Fig. 18(b), the time variation of pitch angle θ_p of the guide vane at $V_a / U_R = 0.666$ is shown for two types of guide vanes, that is, the cases with and without links. It is clearly seen that the use of links brings the vane to



(a) Axial velocity



(b) Pitch angle

Fig. 18 Behavior of guide vane showing comparison with and without links

diffuser setting angle and keeps it there and thus improves diffuser action as compared to vanes without links.

6. CONCLUSIONS

The characteristics of self-rectifying air turbines for wave power conversion proposed so far have been investigated by the model testing and numerical simulation under irregular flow conditions. It has been found that the impulse type turbines have the potential to be superior to the Wells type turbines in the overall performances under irregular flow conditions. Furthermore, the authors have explained the mechanism of hysteretic behavior of the Wells turbine and the necessity of links for the improvement of the performance of the impulse turbine with self-pitch-controlled guide vanes.

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