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Keynote Paper

STATE OF ART ON 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, it is found that the running and starting characteristics of the impulse type turbines could be superior to those of the Wells turbine under irregular wave conditions.

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

1. INTRODUCTION

Several of the wave energy devices being studied under any wave energy program 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 Falcao, 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. On the other hand, 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, 1999; Takao et al., 1997]. However, the comparison of characteristics of all these has not been made so far.

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, which could be used for wave power conversion in the near future, 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

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has been suggested.

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 drives 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, 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.

A reciprocating airflow may be rectified in order to produce unidirectional flow, 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 OWC wave energy converter. However, the 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

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

Fig. 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" supported from the JOULE II [Falcao et al., 1993], where the turbine has actively pitch-controlled blades.

The project using the biplane Wells turbine (Fig. 4) is making progress in Islay, U.K. [Falcao 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. However, detailed information of the turbine characteristics has not been clarified.

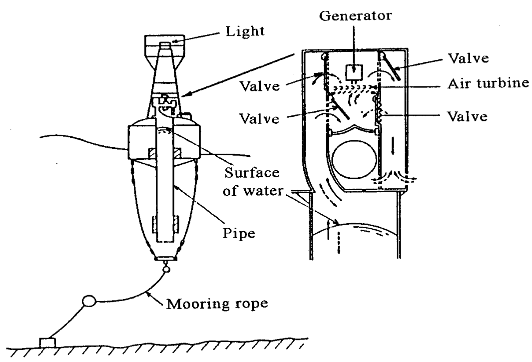


Fig. 1 Outline of navigation buoy

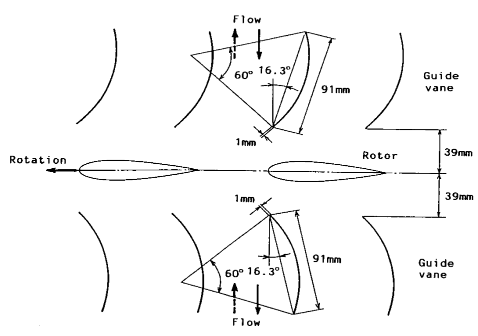


Fig.2 Wells turbine with guide vanes: WTGV

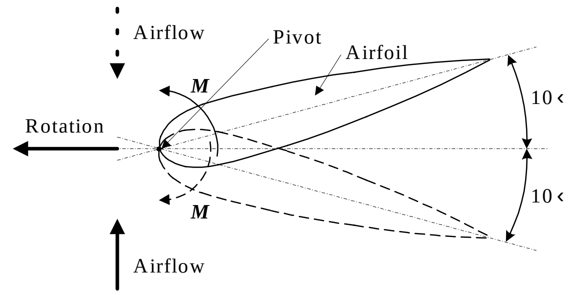


Fig. 3 Turbine using self-pitch-controlled blades: TSCB

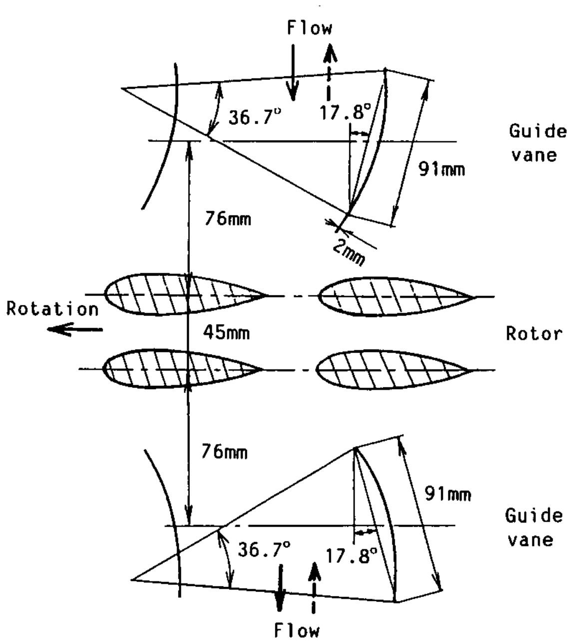


Fig. 4 Biplane Wells turbine with guide vanes: BWGV

3.2 Impulse Turbine

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

Fig. 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 hub and 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 oscillating airflow. This turbine (turbine diameter of 1.0m) has been constructed by NIOT, India [Santhakumar, 1996; Santhakumar et al., 1998].

Fig. 7 shows the impulse turbine with fixed guide

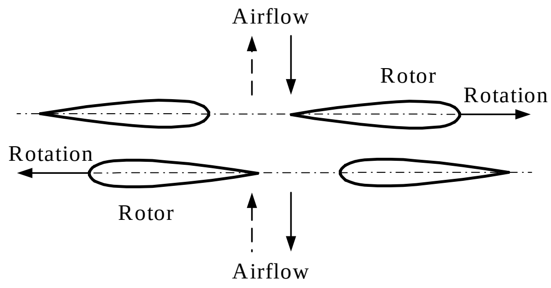


Fig. 5 Contra-rotating Wells turbine

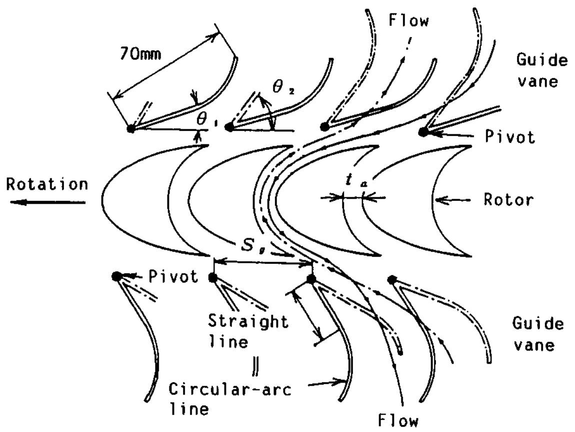


Fig. 6 Impulse turbine with self-pitch-controlled guide vanes connected by link motion: ISGV

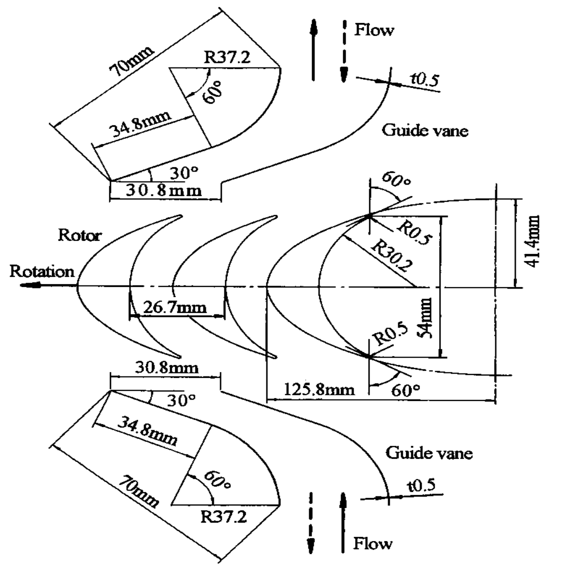


Fig. 7 Impulse turbine with fixed guide vanes: IFGV

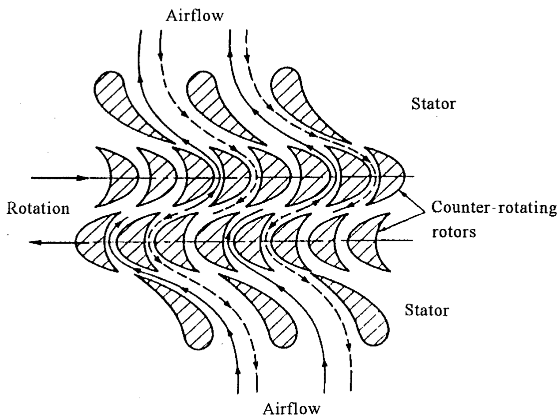


Fig. 8 McCormick counter-rotating turbine

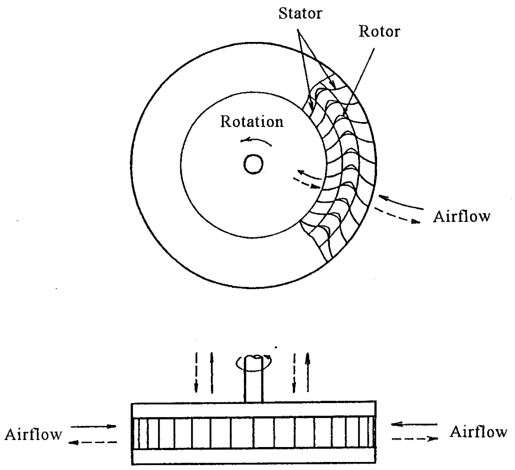


Fig. 9 Radial turbine

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

Fig. 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. On the other hand, according to a recent research, the efficiency is not so good [Takao 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 [Setoguchi et al., 1998]. (b) Turbine with self-pitch-controlled blades named TSCB, Fig. 3 [Inoue et al., 1989; 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. 6 [Setoguchi, et al., 1993, 1996]. (e) Impulse turbine with fixed guide vane named IFGV, Fig. 7 [Setoguchi et al., 2000]. Here the contra-rotating Wells turbine was not adopted in the study 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: $\nu=$ 0.7, aspect ratio: $AR=0.5$, chord length of rotor: $l_r=90$ mm, rotor solidity at mean radius: $\sigma_{rR}=0.67$ and guide vane solidity at mean radius: $\sigma_{gR}=1.25$. (b) TSCB (Fig. 3); NACA0021, $D=$ 298mm, $\nu=$ 0.7, $AR=$ 0.6, $l_r=75$ mm, $\sigma_{rR}=0.75$ and preset angle of 10°. (c) BWGV (Fig. 4); NACA0020, $D=298$mm, $\nu=0.7$, $AR=0.5$, $l_r=90$ mm, $\sigma_{rR}=0.45$ and $\sigma_{gR}=1.25$. (d) ISGV (Fig. 6); $D=298$mm, $\nu=0.7$, $t_a/S_r=0.4$ (see Fig. 6), l_r =54 mm, inlet (or outlet) angle of rotor: $\gamma=$ 60°, $\sigma_{rR}=$ 2.02, $\sigma_{gR}=$ 2.27, setting angle of upstream guide vane: $\theta_1=$ 17°, setting angle of downstream guide vane: $\theta_2=$ 72.5° and sweep angle of rotor: $\lambda=-7.5°$. (e) IFGV (Fig. 7); $D=298$mm,

$\nu = 0.7$, $t_a / S_r = 0.4$, l_r =54 mm, $\gamma = 60°$, $\sigma_{rR} = 2.02$, $\sigma_{gR} = 2.27$, guide vane setting angle of 30° and $\lambda = -7.5°$. 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, 1999]. Furthermore, all of them can start [Inoue et al., 1986a] by themselves.

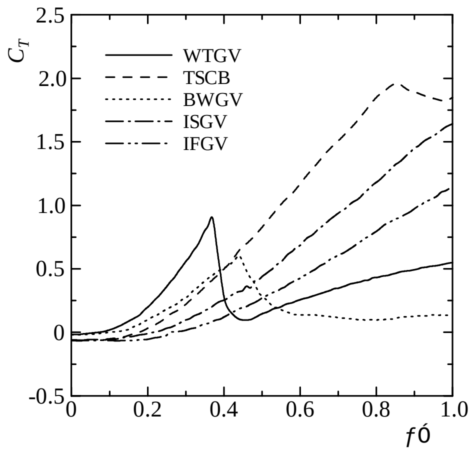
4.2 Turbine Characteristics under Steady Flow Conditions

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:

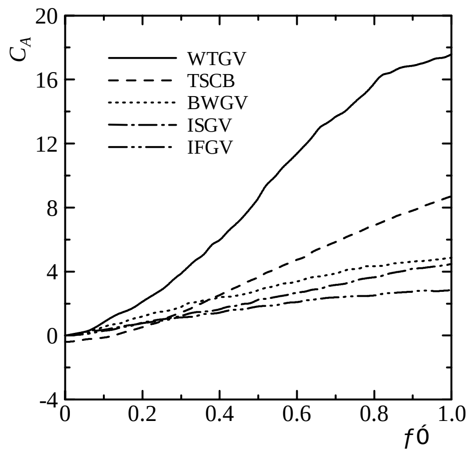
$$C_T = T_o / \{\rho_a (v_a^2 + U_R^2) b l_r z r_R / 2\} \quad (1)$$

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

$$\phi = v_a / U_R \quad (3)$$



(a) Torque coefficient



(b) Input coefficient

Fig. 10 Turbine characteristics under steady flow conditions

where ρ_a: density of air, b: rotor blade height, l_r: chord length of rotor. 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 0.4×10^5 for ISGV and IFGV (impulse type turbines).

Fig. 10(a) shows $C_T - \phi$ 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.

Fig. 10(b) shows $C_A - \phi$ 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 air chamber is small when impulse type turbines are adopted for wave power generator device.

Concerning the turbine efficiency η under steady flow conditions, we can easily take $\eta - \phi$ 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 characteristics depend on the efficiency of air chamber, i.e., the ratio of power of OWC and incident wave power.

4.3 Simulation of Turbine Characteristics under Irregular Flow Conditions

Since sea waves are irregular, and the airflow generated by the oscillating water column is also thus irregular, it is very important to clarify the turbine characteristics in connection with OWC under irregular flow conditions. Here let us simulate the characteristics in order to clarify the turbine for wave energy conversion.

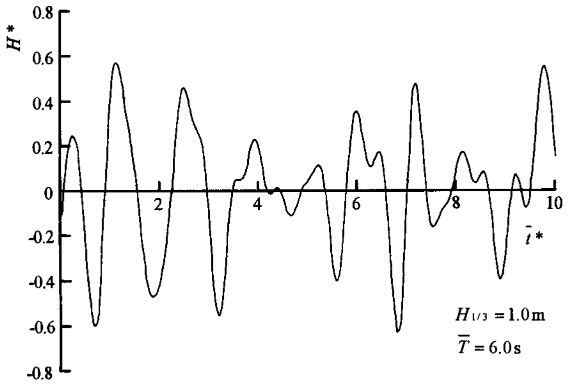


Fig. 11 Test irregular wave

The test irregular wave used in this study is based on the ISSC (International Ship Structure Congress) spectrum which is typical in the field of ocean engineering [Hineno and Yamauchi, 1980]. The spectrum is given as:

$$S^*(f^*) = 0.11 f^{*-5} \exp(-0.44 f^{*-4}) . \qquad (4)$$

The incident wave height H is given as a function of time by such a spectrum. A typical example of wave height in dimensionless form $H^* = H / H_{1/3}$ is shown in Fig. 11, where the significant wave height $H_{1/3}$, the wave mean frequency $\overline{f}$ and the area ratio m are 1.0 m, 0.167 Hz and 0.0234, respectively.

On the other hand, for a wave energy device as shown in Fig. 12, the relationship between the incident wave height and the wave height within air chamber [Setoguchi et al., 1999] is given as:

$$\frac{d}{dt}\left(\rho_s h A_c \frac{dh}{dt}\right) = \{\rho_s g(H-h) - \Delta p\} A_c . \qquad (5)$$

where ρ_s : density of seawater, A_c : air chamber cross-sectional area, g : gravity. Since $v_a = \frac{1}{m}\frac{dh}{dt}$, Δp is a function of $\frac{dh}{dt}$ if the rotational speed U_R is given. This is approximate equation because the equation of motion about OWC is generally expressed by using linear water wave theory [Chatry et al., 1999]. As the relationship between Δp and $\frac{dh}{dt}$ is obtained from $C_A - \phi$ characteristics, here let it put as $\frac{\Delta p}{\rho_s} \equiv F\left(\frac{dh}{dt}\right)$, Eq. (5) is rewritten as follows:

$$\rho_s A_c \left\{ \left(\frac{dh}{dt}\right)^2 + h\frac{d^2h}{dt^2} \right\} = A_c\{\rho_s g(H-h) - \Delta p\}$$

Then,

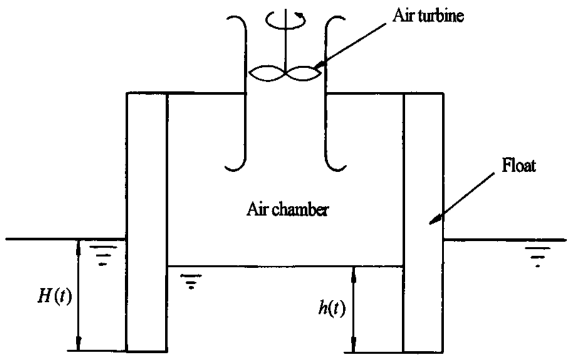


Fig. 12 Schematic layout of OWC-air turbine type wave power generator system

$$h\frac{d^2h}{dt^2} + \left(\frac{dh}{dt}\right)^2 + F\left(\frac{dh}{dt}\right) - g(H-h) = 0 . \qquad (6)$$

The above equation can be solved by using Runge-Kutta-Gill method, and then the wave height within air chamber is obtained. The incident wave power $\overline{W_i}$ and power of OWC $\overline{W_O}$ are defined as follows:

$$\overline{W_i} = \sum_{i=1}^{N} \frac{1}{32\pi} \rho_s g^2 H_i^2 T_i^2 / \sum_{i=1}^{N} T_i \qquad (7)$$

$$\overline{W_O} = \sum_{i=1}^{N} \frac{1}{32\pi} \rho_s g^2 h_i^2 T_i^2 / \sum_{i=1}^{N} T_i . \qquad (8)$$

Then, the efficiency of air chamber is

$$\tilde{\eta}_c = \overline{W_O} / \overline{W_i} . \qquad (9)$$

Note here that strictly speaking, pressure and flow rate of air should be taken into consideration for an evaluation of the efficiency of air chamber. However, the objective of this study is just to compare the performances of the turbines relatively. In this case, it is considered that above method is suitable to enough to evaluate the efficiency of the chamber. Therefore, a ratio of the power of OWC to the incident wave power has been adopted as the efficiency.

Assuming incompressible flow, the axial flow velocity is directly proportional to a variation of the wave height. The non-dimensional axial flow velocity through the turbine v_a^* is written as:

$$v_a^* = \frac{d(h/H_{1/3})}{d(t/\overline{T})} = \frac{dh^*}{d\bar{t}^*} . \qquad (10)$$

The running and starting characteristics of the turbine in irregular flow were calculated by numerical simulation. The steady flow characteristics of the turbines are assumed to be valid for computing performance under unsteady flow conditions. Such a quasi-steady analysis has been validated by the previous studies for both Wells turbine [Inoue et al., 1986a] and the impulse turbine [Setoguchi et al., 1993].

The equation of motion for a rotating system of the turbine in irregular flow can be described in dimensionless form as:

$$K^2 X_I \frac{\mathrm{d}\overline{\omega}^*}{\mathrm{d}\bar{t}^*} + X_L = C_T(\phi)\frac{(K\overline{\omega}^*)^2 + v_a{}^{*2}}{2}\sigma_{rR}\frac{4(1-\nu)}{1+\nu} \tag{11}$$

where, $\phi = v_a{}^*/(K\overline{\omega}^*)$, $K\overline{\omega}^* = \omega m r_R \overline{T}/H_{1/3}$ and $v_a{}^* = m\overline{T}v_a/H_{1/3}$. The first and second terms on the left side of Eq. (11) are inertia and loading terms respectively, and the right hand side represents a torque generated by a turbine. It is clear from Eq. (11) that the behavior of the turbine (starting characteristics) can be calculated as a function of $K\overline{\omega}^*$ and $v_a{}^*$, when loading characteristics $X_L(\overline{\omega}^*)$, torque coefficient $C_T(\phi)$ and rotor geometrical parameters such as X_I , σ_{rR} and ν are specified.

Similarly, the running characteristics are obtained by keeping rotational speed constant. In this case, the mean output $\overline{C_o}$ and input coefficient $\overline{C_i}$ from $\bar{t}^* = 0$ to 10 (see Fig. 11) are given respectively as:

$$\overline{C_o} = \frac{1}{\bar{t}^*}\int_0^{t^*} C_T(\phi)\frac{(K\overline{\omega}^*)^2 + v_a{}^{*2}}{2}\sigma_{rR}\frac{4(1-\nu)}{1+\nu}\overline{\omega}^*\mathrm{d}\bar{t}^* \tag{12}$$

$$\overline{C_i} = \frac{1}{\bar{t}^*}\int_0^{t^*} C_A(\phi)\frac{(K\overline{\omega}^*)^2 + v_a{}^{*2}}{2K}\sigma_{rR}\frac{4(1-\nu)}{1+\nu}v_a{}^*\mathrm{d}\bar{t}^* . \tag{13}$$

Then, mean turbine efficiency is

$$\tilde{\eta}_t = \overline{C_o}/\overline{C_i} . \tag{14}$$

Therefore, the conversion efficiency of the wave energy device is

$$\tilde{\eta} = \tilde{\eta}_c \cdot \tilde{\eta}_t . \tag{15}$$

In the calculations, the flow condition is assumed to be quasi-steady, therefore, 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.

Fig. 13 shows the comparison of conversion efficiency of wave energy. For the impulse type turbines, conversion efficiency is quite high at region of large $1/(K\overline{\omega}^*)$ 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 with comprehensive region of flow coefficient. 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, it 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 $H_{1/3}$.

Fig. 14 shows time variation of wave height in air chamber at condition showing maximum efficiency for WTGV and IFGV. The maximum value of h^* for IFGV is larger than that for WTGV. Since the value of C_A for WTGV is large as shown in Fig. 10(b), the pressure in air chamber becomes high and the motion of OWC is suppressed. For IFGV, however, the motion of OWC is more active because the value of C_A is lower and rotational speed of rotor is quite lower than that of the Wells turbine. Therefore, the difference of the

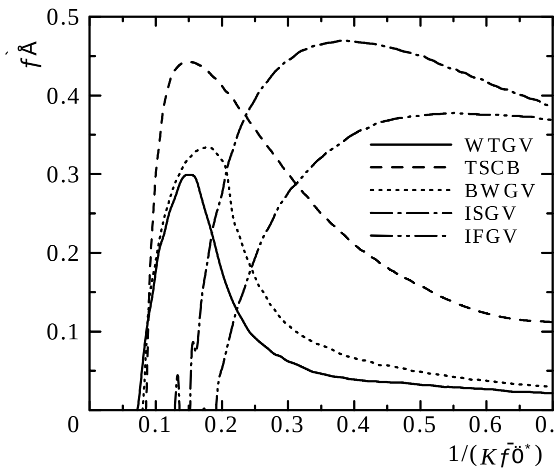


Fig. 13 Comparison of conversion efficiency of wave energy

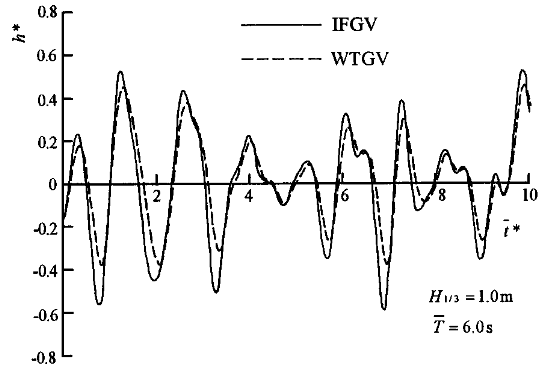


Fig. 14 Time variations of wave height in air chamber at condition showing maximum efficiency

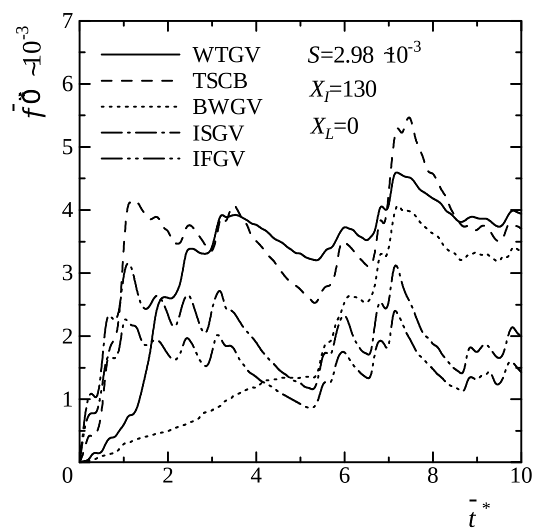


Fig. 15 Starting characteristics under irregular wave conditions

efficiency of air chamber is considered to be one of causes that energy conversion efficiency of wave power conversion using the impulse type turbine is higher than that efficiency using the Wells type turbine.

The starting characteristics for five turbines are shown in Fig. 15. The impulse type turbine can start in 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 loading-free condition for the impulse turbine is larger than the Wells type turbines as shown in Fig. 10(a). Therefore, it is possible for the impulse type turbines to design an excellent turbine with low operational speed, which is desirable from the viewpoints of noise reduction and mechanical advantage.

5. 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. As a result, the impulse type turbines have the potential to be superior to the Wells type turbines in the overall performances under irregular flow conditions. This is because the impulse turbine has no rotor stall and the operational speed is very low.

NOMENCLATURE

A_c : air chamber cross-sectional area
AR : aspect ratio
A_t : turbine flow passage area
b : blade height
C_A : input coefficient {Eq. (2)}
$\overline{C_i}$: mean input coefficient {Eq. (13)}
$\overline{C_o}$: mean output coefficient {Eq. (12)}
C_T : torque coefficient {Eq. (1)}
D : turbine tip diameter
f : frequency of wave motion
$\overline{f}$: mean frequency of wave motion $=1/\overline{T}$
f^* : non-dimensional frequency = $f/\overline{f}$
h : wave height in air chamber
h^* : non-dimensional wave height in air chamber $=h/H_{1/3}$
H : incident wave height
$H_{1/3}$: significant wave height
H^* : non-dimensional incident wave = $H/H_{1/3}$
I : moment of inertia of rotor
K : non-dimensional period = $r_R m/H_{1/3}$
l : chord length
N : number of waves
m : area ratio = A_t/A_c
Q : flow rate
r_R : mean radius
S^* : non-dimensional spectrum of wave motion {Eq. (4)}
S_r : rotor blade space at r_R
t : time
$\bar{t}^*$: non-dimensional time in irregular flow $=t/\overline{T}$
t_a : width of flow path at mean radius
T : period of wave motion
T_o : output torque
T_L : loading torque
$\overline{T}$: mean period in irregular flow $=1/\overline{f}$
U_R : circumferential velocity at r_R
V : reference velocity = $H_{1/3}/(m\overline{T})$
v_a : mean axial flow velocity

v_a^* : non-dimensional axial flow velocity = v_a / V {Eq. (10)}
w : relative inflow velocity
$\overline{W_i}$: incident wave power {Eq. (7)}
$\overline{W_O}$: wave power of OWC {Eq. (8)}
X_I : non-dimensional moment of inertia = $I / (\pi \rho_a r_R^5)$
X_L : non-dimensional loading torque = $T_L / (\pi \rho_a V^2 r_R^3)$
z : number of rotor blades

Greek symbols
Δp : total pressure drop between settling chamber and atmosphere
γ : blade inlet (outlet) angle for impulse turbine rotor
δ : camber angle of guide vane
$\tilde{\eta}$: conversion efficiency under irregular flow condition {Eq. (15)}
$\tilde{\eta}_c$: efficiency of air chamber {Eq. (9)}
$\tilde{\eta}_t$: mean turbine efficiency under irregular flow condition {Eq. (14)}
θ : setting angle of guide vane
ν : hub-to-tip ratio
λ : sweep angle
ρ_a : density of air
ρ_s : density of seawater
σ : solidity at r_R $= lz / (2\pi r_R)$
ϕ : flow coefficient {Eq. (3)}
ω : angular velocity of rotor
$\overline{\omega}^*$: non-dimensional angular velocity under irregular flow condition = $\omega \overline{T}$

Subscripts
g : guide vane
r : rotor
R : mean radius
1 : nozzle
2 : diffuser

Superscript
* : non-dimensional value

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