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Study of turbine with self-pitch-controlled blades for wave energy conversion

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Abstract

The objective of this paper is to clarify the performance of a Wells air turbine using self-pitch-controlled blades under the real sea conditions and to obtain the useful information about the optimum setting angle. Experimental investigations were performed by model testing of a rotor with fixed blades under steady flow conditions. Then, the running and starting characteristics under sinusoidally oscillating flow conditions were obtained by a computer simulation using a quasi-steady analysis. As a result, the performances of the air turbine using self-pitch-controlled blades under the real sea conditions were clarified, and a suitable choice of design factor has been suggested for the setting angle of the rotor.

Keywords: Wave energy conversion; Ocean energy; Wells turbine; Variable pitch blade; Axial flow turbine; Self-rectifying turbine; Oscillating air flow

1. Introduction

One of the so-called renewable energy sources which attracts attention especially in Europe and Asia is energy contained in ocean waves. There are various techniques for extracting energy from waves. Some are based on a power train system of hydro-pneumatic-mechanical-electrical energy conversion in which an air turbine is an essential element. A typical example of these techniques is wave energy conversion using an oscillating water column (O.W.C.).

In 1976, Dr. A.A. Wells proposed a form of self-rectifying, axial-flow air turbine, the so-called Wells turbine, as a device suitable for wave energy conversion in OWC. A self-rectifying turbine provides uni-directional rotational for a reciprocating air flow through it. So far, the performance of the Wells turbine has been studied in the United Kingdom, Japan, Ireland and other countries [1–5]. According to these results, the Wells turbine has inherent disadvantages: lower efficiency, poorer starting, higher noise level and higher axial thrust in comparison with conventional turbines.

On the other hand, in order to develop a practical wave power generator system, several types of variable pitch air turbines for wave energy conversion have been presented so far [6,7]. The authors have also proposed an air turbine with self-pitch-controlled blades [8–11]. The air turbine consists of several symmetrical blades that change the pitch angle by aerodynamic force so as to obtain higher torque and efficiency in a reciprocating airflow. This turbine is simpler geometrically and inexpensive to manufacture in comparison with an air turbine using active-pitch-controlled blades. According to previous studies, our turbine is advantageous from viewpoints of reduced speed of the rotor, as well as better running and starting characteristics in comparison with the Wells turbine. Furthermore, it was found by a computer simulation that the optimum setting angle of the rotor blade is about 6° [8,11]. In this simulation, however, it is assumed that the axial flow velocity does not depend on flow direction, although, in general, the maximum value of the airflow velocity during exhalation (i.e., from air chamber to atmosphere) is higher than that during inhale (i.e., from atmosphere to air chamber) according to observations in some wave energy plants [12,13]. Therefore, the flow conditions which were used in arriving at this optimum angle are not what wave seen in practical situations.

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Nomenclature

b	blade height	m
C_A	input coefficient (Eq. (2))	
C_T	torque coefficient (Eq. (1))	
f	frequency of wave $= 1/T$	s^{-1}
I	moment of inertia of rotor	$kg \cdot m^2$
l	chord length	m
Q	flow rate	$m^3 \cdot s^{-1}$
r_R	mean radius	m
S	dimensionless frequency $= f r_R / V$	
t	time	s
t^*	dimensionless time $= tf$	
T	period of wave $= 1/f$	s
T_0	output torque	$kg \cdot m^2 \cdot s^{-2}$
T_L	loading torque	$kg \cdot m^2 \cdot s^{-2}$
U_R	blade speed at $r_R = r_R \omega$	$m \cdot s^{-1}$
v	mean axial velocity under steady flow conditions	$m \cdot s^{-1}$
v_i	mean axial velocity during exhale	$m \cdot s^{-1}$
v_0	mean axial velocity during inhale	$m \cdot s^{-1}$
V	reference velocity $= (V_0 + V_i)/2$	$m \cdot s^{-1}$
V_i	maximum value of v_i	$m \cdot s^{-1}$
V_0	maximum value of v_0	$m \cdot s^{-1}$
X_I	dimensionless moment of inertia $= I/(\pi \rho r_R^5)$	
X_L	dimensionless loading torque $= T_L/(\pi \rho r_R^3 V^2)$	
z	number of rotor blades	

Greek symbols

γ	setting angle	
γ_i	γ during inhale	
γ_0	γ during exhale	
Δp	total pressure drop between before and after turbine	$kg \cdot m^{-1} \cdot s^{-2}$
η	turbine efficiency $= T_0 \omega / (\Delta p Q) = C_T/(C_A \phi)$	
$\bar{\eta}$	mean efficiency (Eq. (5))	
ρ	density of air	$kg \cdot m^{-3}$
σ_R	solidity of rotor at $r_R = zl/(2\pi r_R)$	m
ϕ	flow coefficient (Eq. (3))	
Φ	flow coefficient $= V/U_R$	
ω	angular velocity of rotor	
ω^*	dimensionless angular velocity $= \omega/f$	

The objective of this paper is to clarify the performance of the air turbine using self-pitch-controlled blades under the real sea conditions and to obtain information about the optimum blade setting angle. The experimental investigations were performed by model testing of the rotor with fixed blades under steady flow conditions. Then, the running and starting characteristics under sinusoidally oscillating flow conditions have been clarified by a computer simulation using a quasi-steady analysis.

2. Working principle

The working principle of the turbine using self-pitch-controlled blades is illustrated in Fig. 1. 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 of $\pm\gamma$. As an airfoil set at a certain angle of incidence experiences a pitching moment M about the pivot, the turbine blades can flip by themselves between $+\gamma$ and $-\gamma$ according to the flow direction, as shown in Fig. 1. Therefore, higher torque and efficiency are obtained with a lower rotating speed than that of the Wells turbine.

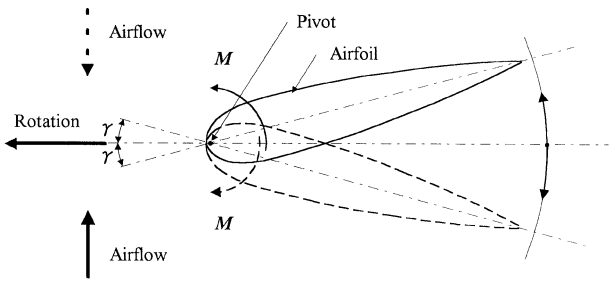


Fig. 1. Air turbine using self-pitch-controlled blades for wave energy conversion.

3. Test apparatus and procedures

The experiments were conducted by model tests of turbine rotors with fixed staggered blades under steady flow conditions. The performance test of the turbine was carried out in a wind tunnel which facilitated the production of both steady and oscillating air flows by controlling the motion of a 1.4 m dia. piston in a cylinder by a computer. The turbine was located at the exit of the wind tunnel. The test rig is shown in Fig. 2. In the test of turbine characteristics, the variation of turbine performance with relative inlet angle of $\tan^{-1}(v/U_R)$ was investigated by changing the rotational speed of the rotor step by step from a low speed to a high speed covering an effective operating range of the turbine. During the test, the turbine output torque, the airflow rate, and the total pressure drop across the rotor were also measured, and the signals were logged and analyzed with the online computer. The uncertainty of turbine efficiency is about $\pm 1\%$.

The details of turbine rotor used in this study are as follows. The rotor blade profile: NACA0021 with chord length $l = 75$ mm, number of blades: $z = 8$, solidity at mean radius: $\sigma_R = 0.75$, hub-to-tip ratio: 0.7, tip diameter:

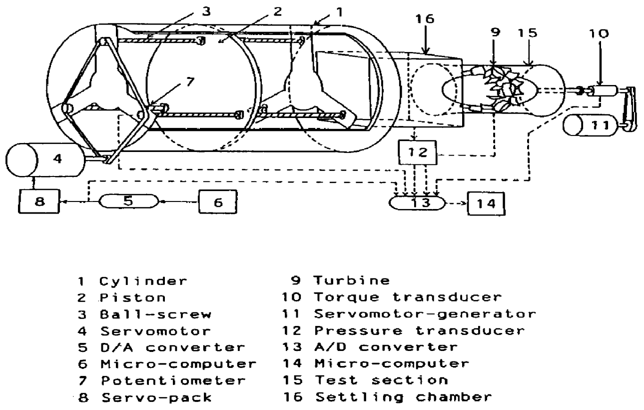


Fig. 2. Test rig.

298 mm, tip clearance: 1 mm: aspect ratio: 0.6, setting angle: $0° \leqslant \gamma \leqslant 12°$. Similar configuration is used for the Wells turbine rotor ($\gamma = 0°$) in the "Mighty Whale" project organized by JMSTEC, Japan [14,15].

4. Experimental results and discussions

4.1. Turbine characteristics under steady flow conditions

Turbine characteristics under steady flow conditions were evaluated in terms of variation of the torque coefficient C_T and the input power coefficient C_A against the flow coefficient ϕ, which are defined as:

$$C_T = T_0 / \{\rho(v^2 + U_R^2) blzr_R/2\} \quad (1)$$

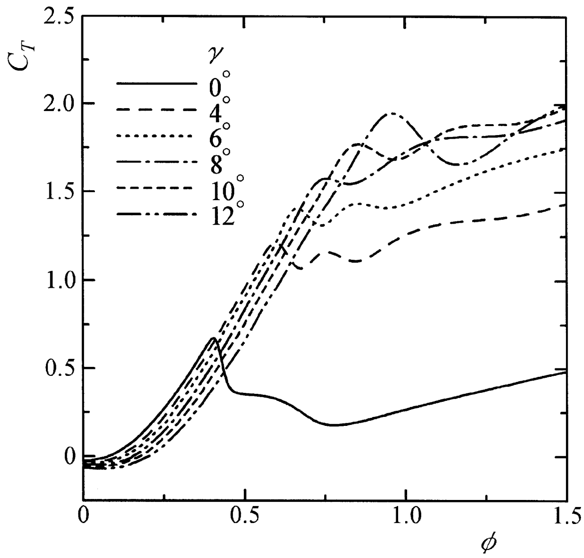
$$C_A = \Delta p Q / \{\rho(v^2 + U_R^2) blzv/2\} \quad (2)$$

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

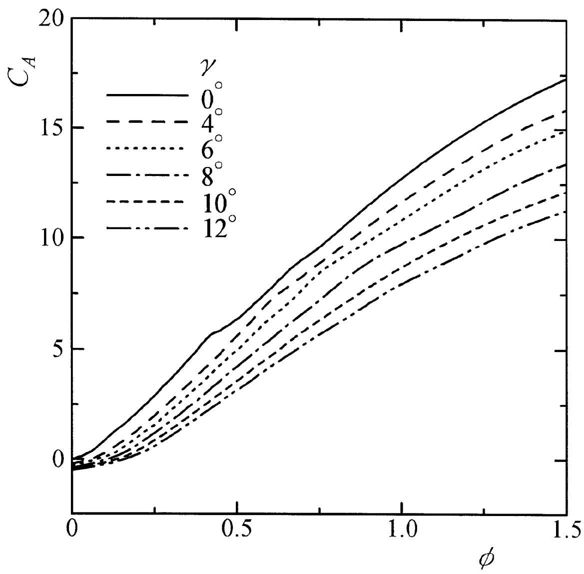
where ρ: density of air, v: mean axial flow velocity, U_R: circumferential velocity at mean radius r_R, b: rotor blade height, l: chord length.

Fig. 3(a) shows C_T–ϕ characteristics for the turbines with various blade setting angles. In this figure, a solid line ($\gamma = 0°$) represents the C_T for the Wells turbine. As is evident from the figure, the value of C_T decreases with increasing γ in the stall-free region. On the other hand, C_T at higher flow coefficients increases with γ. This means that the larger the γ is, the better the starting characteristics become. Furthermore, the flow coefficient at no-load condition (i.e., $C_T = 0$) and the stall point increase with γ. This is because the angle of attack is defined by $\{\tan^{-1}(v/U_R) - \gamma\}$.

Fig. 3(b) shows C_A–ϕ characteristics for the turbine. The figure indicates that the input coefficient C_A curve in the case of $\gamma = 0°$ is higher than those of $\gamma \neq 0°$ for all ϕ. This is due to the rotor geometry. That is, a rotor with a higher setting angle corresponds to a lower degree of reaction. Moreover, it should be noted that the value of C_A at small flow coefficient is negative in the case of $\gamma \neq 0°$ as shown in Fig. 3(a). This means that the rotor works as a fan at small relative inlet angles. Therefore, the averaged



(a)



(b)

Fig. 3. Turbine characteristics under steady flow conditions: (a) Torque coefficient; (b) Input coefficient.

performance of the rotor under an oscillating flow condition would deteriorate, if the setting angle was fixed. However, it is possible to overcome this shortcoming in the performance because the torque at a small relative inlet angle becomes zero by using self-pitch-controlled blades.

4.2. Simulation of turbine characteristics under sinusoidally oscillating flow conditions

A number of wave energy plants have been constructed and tested in the United Kingdom, Japan, India and other countries [14–17]. On these test plants, the axial flow velocity during exhalation (i.e., from air chamber to atmosphere) is found to be higher than that during inhalation as shown in Fig. 4 [12,13]. If such is this case, we can see that the opti-

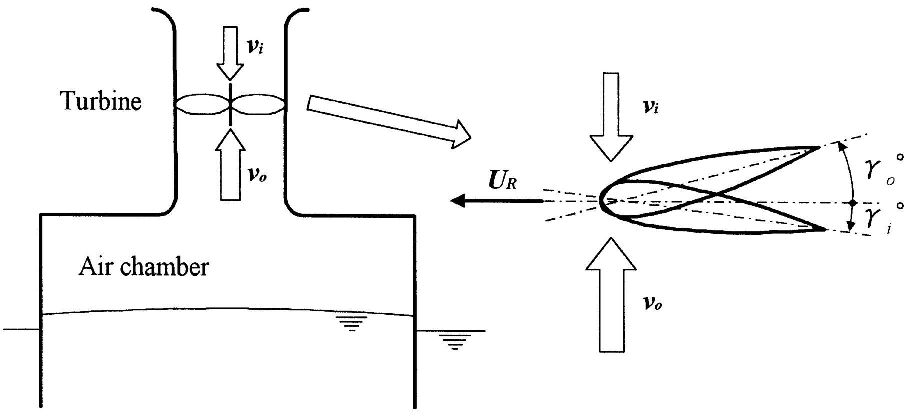


Fig. 4. Outline of wave energy plant with air turbine using self-pitch-controlled blades.

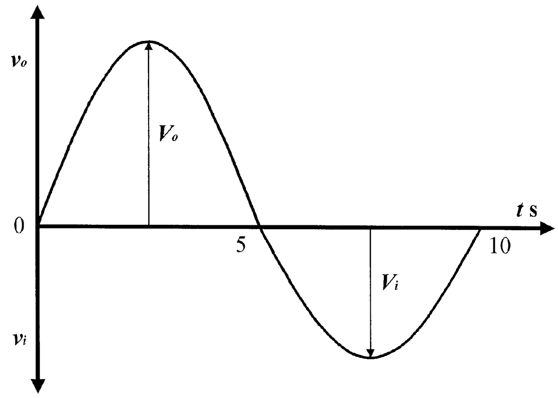


Fig. 5. Axial velocity.

mum setting angle will differ for the two different direction of the airflow. In this simulation, in order to clarify the optimum setting angle, the effect of setting angles on the turbine performance under the real sea conditions has been investigated.

The analytical procedure is the same as shown in Ref. [1]. The equation for a rotating system is written as follows:

$$I\frac{d\omega}{dt} + T_L = T_0 \tag{4}$$

where I: moment of inertia of rotor, t: time, and T_L: loading torque. Eq. (4) can be solved numerically as an initial problem when I, T_L, T_0 and a wave motion are given. This gives the starting characteristics of the turbine at the beginning and the running characteristics as the asymptotic condition. When the solution is in the asymptotic condition, the turbine performance can be obtained by a mean efficiency shown as follows:

$$\bar{\eta} = \left(\frac{1}{T}\int_0^T T_0\omega\,dt\right) \Big/ \left(\frac{1}{T}\int_0^T \Delta p Q\,dt\right) \tag{5}$$

where $\bar{\eta}$ is evaluated over one wave period the rotor rotating at constant speed.

In the calculations, the flow condition is assumed to be quasi-steady, and therefore, the values of C_T and C_A shown in Fig. 3 can be used here. For simplifying the numerical simulation, pitch angle of rotor blades is assumed to change at the same time when axial velocity changes from V_0 to V_i (or from V_i to V_0). The validity of this assumption was shown by a previous study [10], in which mean efficiency calculated for unsteady flow condition agreed with the experimental data sufficiently closely for engineering use.

The variation of the axial flow velocity adopted in the simulation is a sinusoidally and periodically oscillating flow with a wave period $T = 10$ s. The axial velocity is shown in Fig. 5, where v_i: mean axial velocity during inhalation, v_0: mean axial velocity during exhalation, V_i: maximum value of v_i, and V_0: maximum value of v_0. According to Refs. [12,13], for instance, the maximum value of v_i is about 0.6 times of that of v_0 in the wave energy plants of NIOT, India [16], and "Mighty Whale" of JAMSTEC, Japan [14, 15]. Therefore, V_i/V_0 values are taken 1.0, 0.8 and 0.6.

Fig. 6 shows the effect of setting angle on the mean efficiency of the turbine in the case of $\gamma_0 = \gamma_i$, where the abscissa represents the flow coefficient Φ ($= V/U_R$, V: reference velocity $= (v_i + v_0)/2$). As is evident from these three figures, $\bar{\eta}$ increases with γ for any V_i/V_0, and the maximum value of the peak efficiency is obtained in the case of $\gamma_0 = \gamma_i = 10°$. Furthermore, $\bar{\eta}$ decreases with decreasing V_i/V_0 for any setting angle.

On the other hand, the effect of γ_i on the mean efficiency is shown in Fig. 7. In this calculation, γ_0 is kept at 10° for any V_i/V_0 because the turbine with this angle gave the best performance in Fig. 6. In the case of $V_i/V_0 = 1.0$ (Fig. 7(a)), the maximum efficiency is obtained in the case of $\gamma_i = 10°$ as expected. However, in the case of $V_i/V_0 = 0.6$ (Fig. 7(c)), the maximum efficiency is obtained by the turbine with $\gamma_i < 6°$, and the efficiency decreases with increasing γ_i. Moreover, the maximum efficiency in Fig. 7(c) is higher than that in Fig. 6(c). Effect of γ_i on mean efficiency for the case of $V_i/V_0 = 0.8$ as shown in Fig. 7(b). It is found from these fact that the performance of the wave energy plant with the air turbine using self-pitch-controlled blades can be improved by using smaller setting angle during inhalation.

The effect of γ_i on the starting characteristics are shown in Fig. 8, where γ_0 is 10° for any V_i/V_0. The abscissa

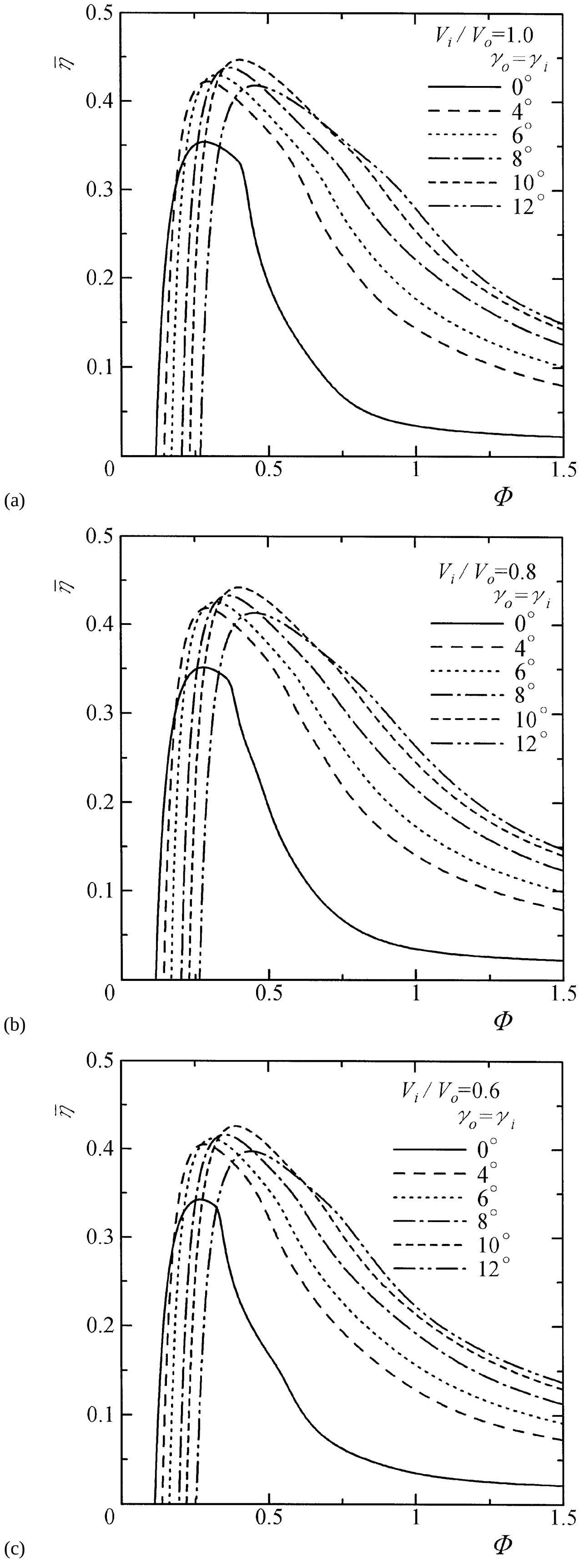


Fig. 6. Mean efficiency under sinusoidally oscillating flow conditions ($\gamma_0 = \gamma_i$): (a) $V_i / V_0 = 1.0$; (b) $V_i / V_0 = 0.8$; (c) $V_i / V_0 = 0.6$.

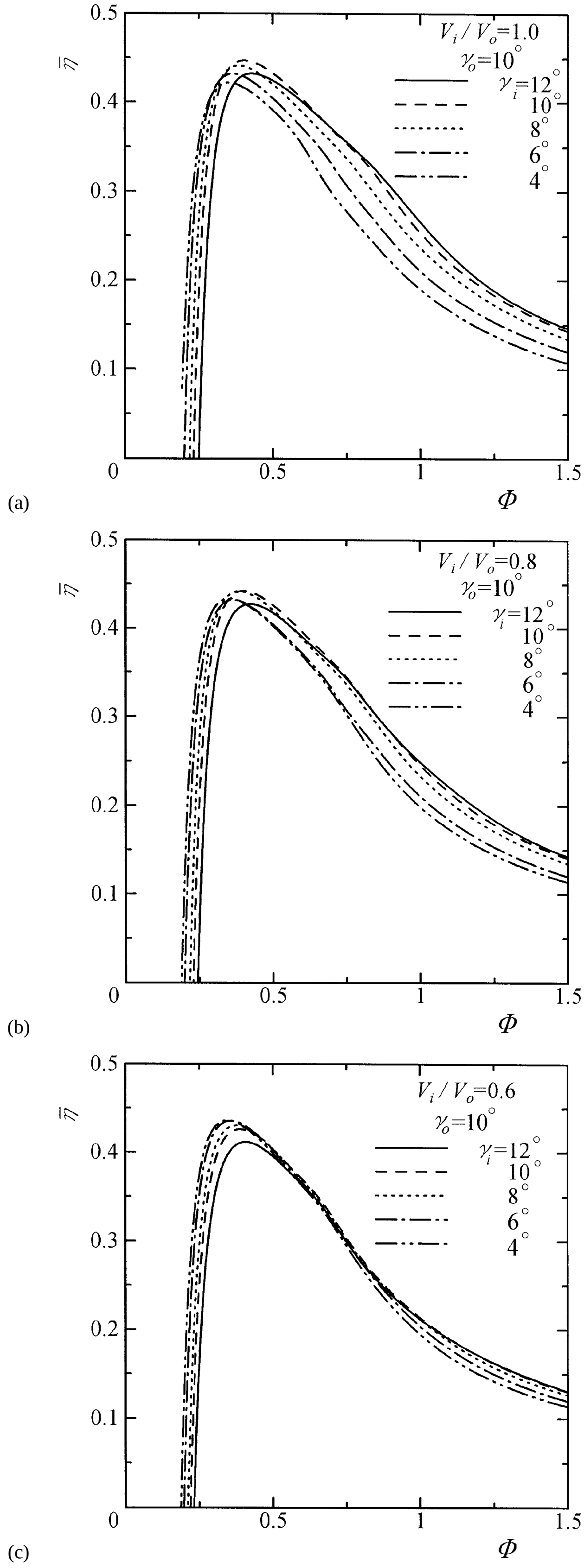


Fig. 7. Effect of setting angle γ_i on mean efficiency: (a) $V_i / V_0 = 1.0$; (b) $V_i / V_0 = 0.8$; (c) $V_i / V_0 = 0.6$.

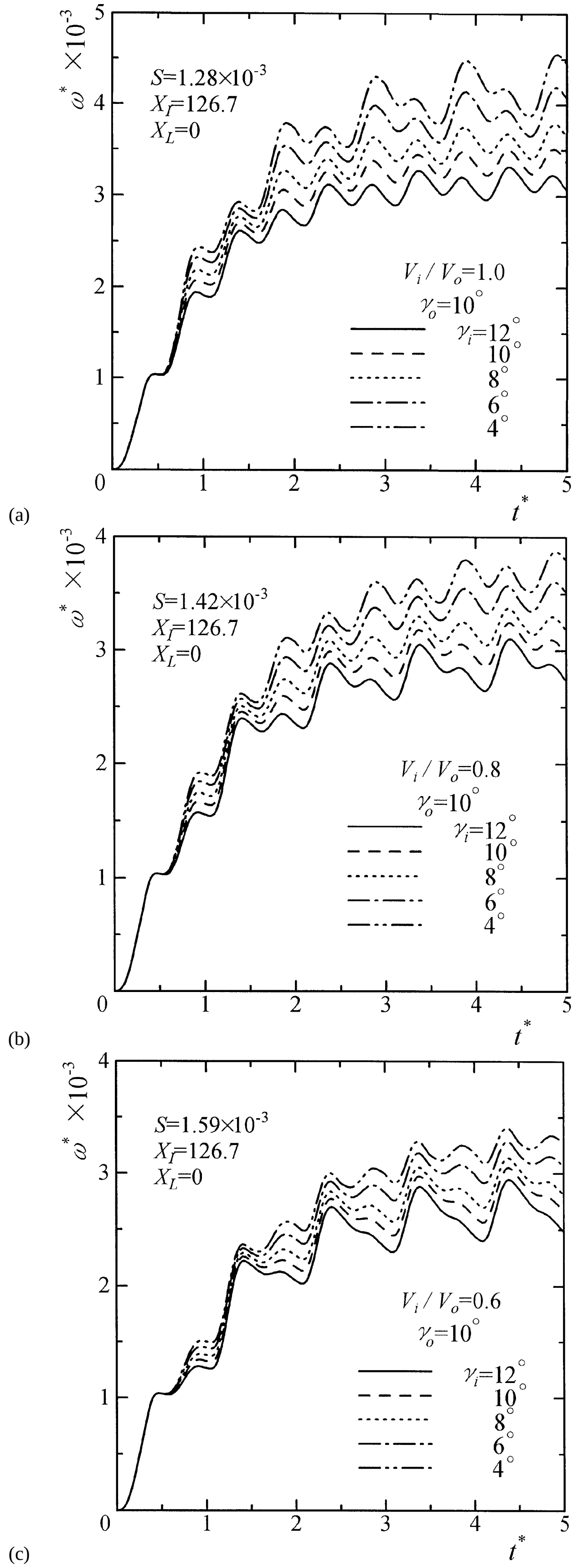


Fig. 8. Starting characteristics under sinusoidally oscillating flow conditions: (a) $V_i/V_0 = 1.0$; (b) $V_i/V_0 = 0.8$; (c) $V_i/V_0 = 0.6$.

and the ordinate represent the dimensionless time $\bar{t}^*$ and the dimensionless angular velocity $\overline{\omega}^*$, respectively. S, X_I and X_L in the figure denote the dimensionless frequency, the dimensionless moment of inertia and the dimensionless loading torque, respectively. It is clear from the figure that all the turbine can start in a short time for any V_i/V_0. Moreover, the larger the γ_i is, the lower the rotational speed in the running condition. This is because the flow coefficient at loading-free condition increases with γ as shown in Fig. 3(a). Therefore, the use of larger blade setting angle enables us to design an excellent turbine with low operational rotational speed, which is desirable because low rotation speed reduces noise and is advantageous in driving low rpm loads.

From above results, it is considered that the optimum values of γ_0 and γ_i under the real sea conditions are about 10° and 6°, respectively.

5. Conclusions

The performance of an air turbine using self-pitch-controlled blades under the real sea conditions has been investigated by the model testing and the numerical simulation. As a result, the running and starting characteristics of the turbine under the real sea conditions were clarified, and a suitable choice of design factor has been suggested for the setting angles of the rotor.

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