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Study of an Impulse Turbine for Wave Power Conversion: Effects of Reynolds Number and Hub-to-Tip Ratio on Performance

The objective of this paper is to report the effects of Reynolds number and hub-to-tip ratio on the performance of the impulse turbine for wave energy conversion. The turbine was investigated experimentally under steady and sinusoidally oscillating flow conditions by model testing. As a result, it was found that the critical value of Reynolds number and the optimum hub-to-tip ratio are approximately 4.0×10^4 and 0.7, respectively. Furthermore, their effect on starting characteristics have been clarified. [DOI: 10.1115/1.1710868]

Introduction

Several of the wave energy devices being studied under many wave energy programs in the United Kingdom, Japan, Portugal, India and other countries make use of the principle of an oscillating water column (OWC) [1–5]. In such wave energy devices, a water column which oscillates due to wave motion is used to drive an oscillating air column which is converted into mechanical energy. The energy conversion from the oscillating air column can be achieved by using a self-rectifying air turbine such as a Wells turbine which rotates in a single direction in oscillating airflow and therefore does not require a system of non-return valves. Many reports describe the performance of the Wells turbine both at running and starting conditions [6–9]. However, according to these studies, the Wells turbine has inherent disadvantages: lower efficiency, poorer starting characteristics and higher noise level in comparison with conventional unidirectional turbines.

On the other hand, in order to develop a high performance self-rectifying air turbine for wave energy conversion, some authors have proposed the impulse turbine with self-pitch-controlled guide vanes and have clarified that the turbine can be operated with higher turbine efficiency and lower rotational speed than the Wells turbine [10,11]. This type of impulse turbine, however, has a disadvantage of maintenance of pivots on which the guide vanes are rotated automatically in a bi-directional airflow. In order to overcome this drawback, an impulse turbine with fixed guide vanes has been also proposed by the authors [12–14]. There are many reports which describe the performance of the impulse turbine both at starting and running conditions. However, the effect of Reynolds number *Re* on the performance of the impulse turbine has not been investigated so far. Furthermore, that of hub-to-tip ratio, v, has not yet been clarified for $v \leqslant 0.7$ [13].

The objective of this study is to present the effects of Reynolds number and hub-to-tip ratio on the performance of the impulse turbine for wave energy conversion and, in particular, for small scale turbines which are suitable for power generation in a navigation buoy. The turbine was investigated experimentally under steady and sinusoidally oscillating flow conditions by model testing.

Experimental Apparatus and Procedure

A schematic view of the test rig is shown in Fig. 1. The test rig consists of a large piston-cylinder (diameter: 1.4 m, length: 1.7 m), a settling chamber and a 300-mm-diameter test section with the inlet and outlet bell-mouth. The turbine rotor is placed at the center of the test section and tested under steady and sinusoidally oscillating flow conditions. In testing the turbine characteristics, the variations of turbine performance with flow coefficient were examined by changing the rotational speed of the rotor step by step from a low to high speed, so as to cover the effective operating range of the turbine. The overall performance was evaluated in terms of the turbine angular velocity ω, the output torque T_o, the flow rate Q and the total pressure drop Δp between the settling chamber and the atmosphere. Tests were performed in the range of ω up to 370 rad/s and Q up to 0.422 m^3/s. The accuracy of the measured turbine efficiency is $\pm1\%$.

As shown in Fig. 2, the turbine configuration employed is the impulse turbine having fixed guide vanes both upstream and downstream, and these geometries are symmetrical with respect to the rotor centerline.

The specifications of the impulse turbine rotor adopted in the experiments are as follows. The blade profile consists of a circular arc on the pressure side and part of an ellipse on the suction side. The ellipse has semi-major axis of 125.8 mm and semi-minor axis

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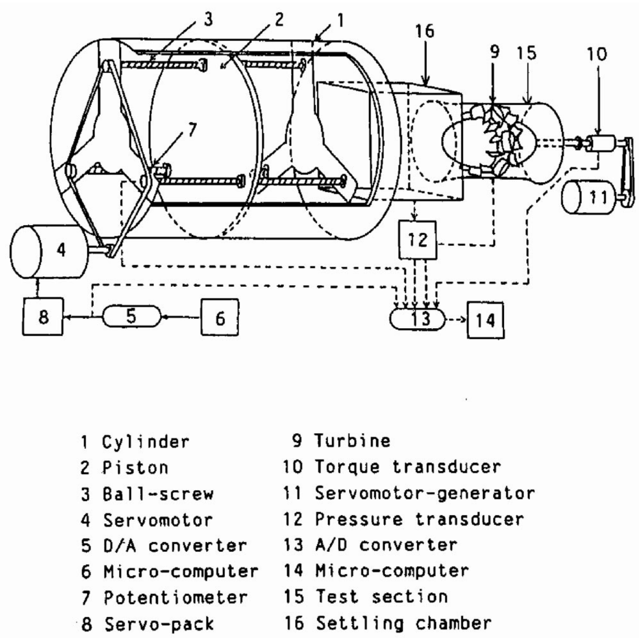

Fig. 1 Test apparatus

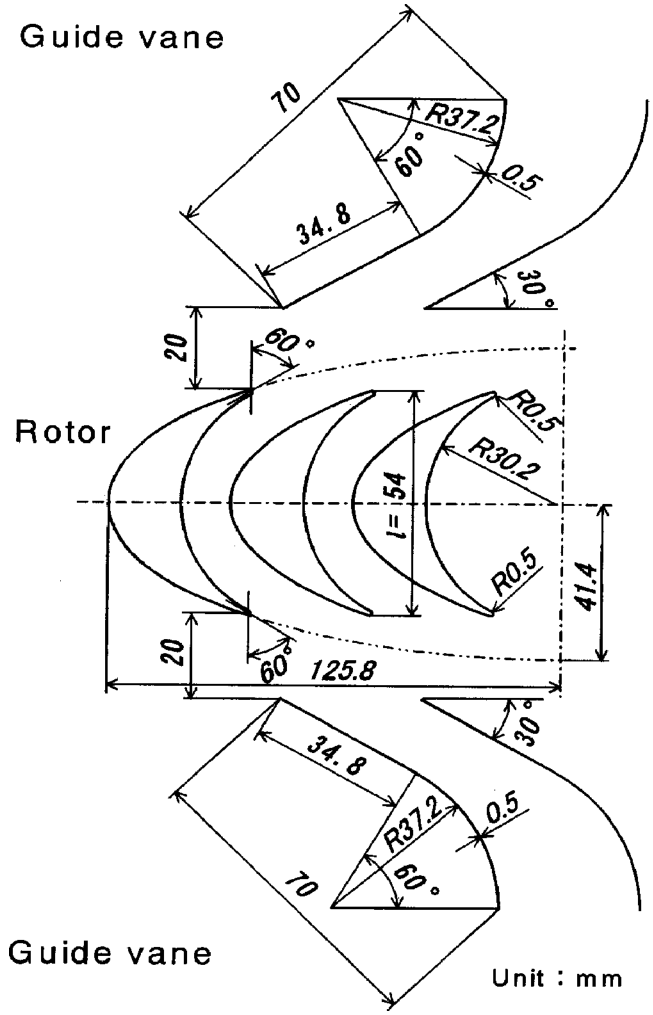

Fig. 2 Turbine configuration

of 41.4 mm. Rotor geometry is given by blade chord length of $l = 54$ mm; tip diameter of 298 mm; blade inlet (or outlet) angle of 60°; thickness ratio of 0.3; and tip clearance of 1 mm. The guide vanes with the chord length of 70 mm are symmetrically installed at a distance of 20 mm downstream and upstream of the rotor. The camber line of guide vane consists of a straight line and a circular arc. Detailed information about the guide vane is as follows: thickness ratio of 0.007; camber angle of 60°; gap to chord ratio of 0.37; setting angle of 30°. The detailed information for the rotor blade and guide vane is shown in Table 1.

It may be noted that this turbine is considered the most promising one in the previous studies [12,13].

The values of hub-to-tip ratio υ adopted in the present study are 0.6, 0.65 and 0.7 because the impulse turbine has already been tested for $\upsilon \geqslant 0.7$ in the previous study [13]. The effect of *Re*, which is defined by Eq. (5), was tested for ranges of $1.72\times10^4 - 4.49\times10^4$ for $\upsilon = 0.6$ and $2.17\times10^4 - 5.14\times10^4$ for $\upsilon = 0.7$, respectively, by changing the flow rate Q.

Experimental Results and Discussions

Effect of Hub-to-Tip Ratio. The turbine performance under steady flow conditions is evaluated by turbine efficiency η, torque coefficient C_T and input coefficient C_A against flow coefficient ϕ. The definitions of these parameters are as follows:

$$\eta = T_o\omega/(\Delta p Q) = C_T/(C_A\phi) \tag{1}$$

$$C_T = T_o/\{\rho(\nu_a^2 + U_R^2) blz r_R/2\} \tag{2}$$

$$C_A = \Delta p Q/\{\rho(\nu_a^2 + U_R^2) blz \nu_a/2\} \tag{3}$$

$$\phi = \nu_a/U_R \tag{4}$$

where ρ, z, b, υ_a and U_R denote the density of air, number of rotor blades, blade height, axial flow velocity and circumferential velocity at r_R, respectively.

Figure 3 shows the effect of hub-to-tip ratio υ on turbine characteristics under steady flow conditions for a Reynolds number of 3.82×10^4. In this study, the Reynolds number is defined as follows:

$$Re = wl/(\mu/\rho) \tag{5}$$

where w and μ are the relative inflow velocity at peak efficiency point and the viscosity of air, respectively.

It can be observed from Fig. 3(*a*) that the turbine efficiency η increases with υ, though the trend of all the tested hub-to-tip ratios is qualitatively the same. Now, the reason for this is discussed using both the $C_T - \phi$, $C_A - \phi$ characteristics (Figs. 3*b* and 3*c*) and the velocity diagram. Comparing the values of C_T and C_A for $\upsilon = 0.6$ with those for $\upsilon = 0.7$, it is seen that the rate of decrease in C_T is larger than that in C_A. This phenomenon could be due to the difference between the incidence angle at the tip and near the hub increasing with decreasing υ. That is, when the υ is small, the incidence angle at tip becomes large, and then boundary layer separation occurs on the pressure side at the leading edge and on the suction side at maximum thickness point of the rotor blade. Consequently, the output torque T_o deteriorates comparatively.

Table 1 Specifications of rotor blade and guide vane

Hub-to-tip ratio ν	Number of rotor blades z	Solidity of rotor blade at r_R	Number of guide vanes	Solidity of guide vane at r_R
0.7	30	2.02	26	2.27
0.65	29	2.01	25	2.25
0.6	28	2.01	24	2.23

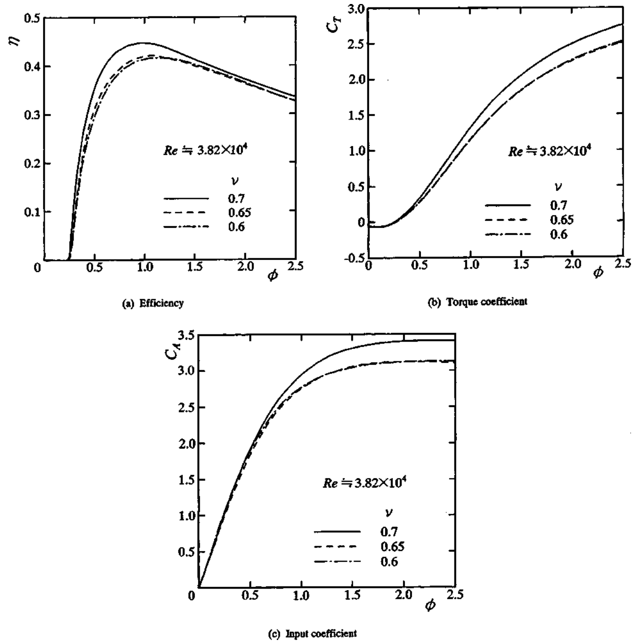

Fig. 3 Effect of hub-to-tip ratio on turbine characteristics under steady flow conditions (*a*) Efficiency (*b*) Torque coefficient (*c*) Input coefficient

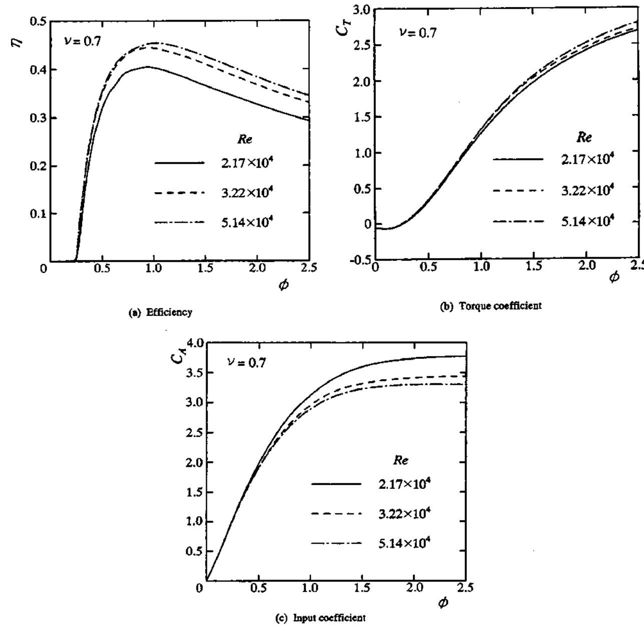

Fig. 5 Effect of Reynolds number on turbine characteristics under steady flow conditions (*a*) Efficiency (*b*) Torque coefficient (*c*) Input coefficient

The starting characteristics of the turbine are evaluated from the variation of rotational speed with time as it accelerates from rest. Figure 4 shows a comparison of starting characteristics under sinusoidal flow conditions (frequency of wave motion: $f = 0.1$ Hz) without any load on the turbine. The results are given in the form of non-dimensional angular velocity ω^* versus dimensionless time t^*. Here, S, X_I and X_L denote the non-dimensional frequency, non-dimensional moment of inertia of the rotor and non-dimensional loading torque, respectively. It is found that in all the cases the turbine started in a very short time and that the rotational speed at operation is almost the same. Consequently, it is considered that the starting characteristics of the turbine are independent of the hub-to-tip ratio.

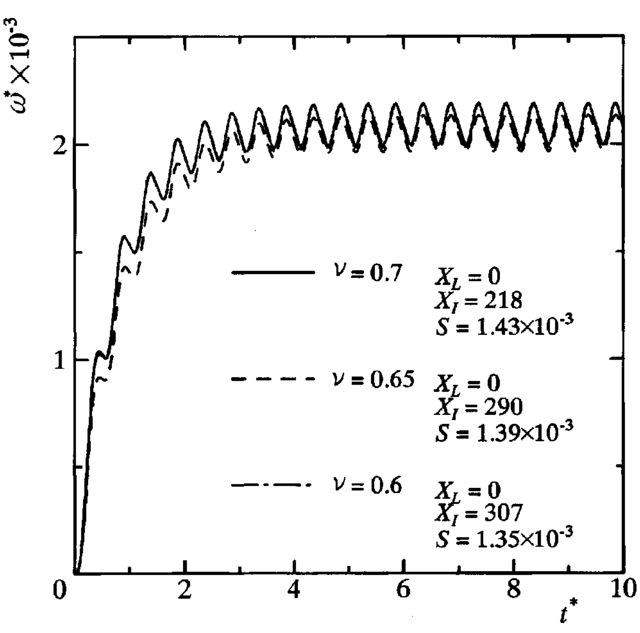

Fig. 4 Effect of hub-to-tip ratio on starting characteristics

According to the previous study [13], υ values equal to and greater than 0.7 were investigated and the optimum value of υ was found to be approximately 0.7. Therefore, it is concluded from the results obtained in this study and the previous study that the optimum value of υ of the impulse turbine with fixed guide vanes for wave power conversion is approximately 0.7.

Effect of Reynolds Number. Figure 5 shows the turbine characteristics under steady flow conditions for three Reynolds number, and for $\upsilon = 0.7$. The efficiency increases with Re in the entire range of ϕ (Fig. 5*a*). This is because the C_T slightly increases with Re, and the C_A decreases with Re (Figs. 5*b* and 5*c*). Therefore, as is evident from Eq. (1), η increases with Re.

Figure 6 shows the effect of Reynolds number Re on peak efficiency η_p for $\upsilon = 0.6$ and 0.7. The value of η_p increases gradually in the case of both the turbines and then remains almost constant for $Re \geqslant 4.0\times10^4$. Therefore, it was concluded that the critical value of Reynolds number of the impulse turbine seems to be approximately 4.0×10^4.

Figure 7 shows the starting characteristics for the three Reynolds numbers under sinusoidal flow conditions. It is found that all the cases started in a very short time and the rotational speed at

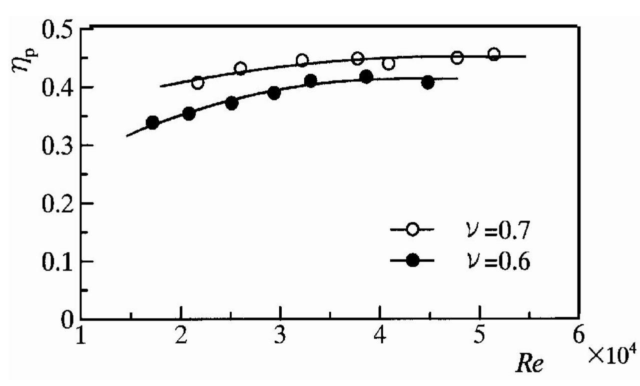

Fig. 6 Effect of Reynolds number on peak efficiency

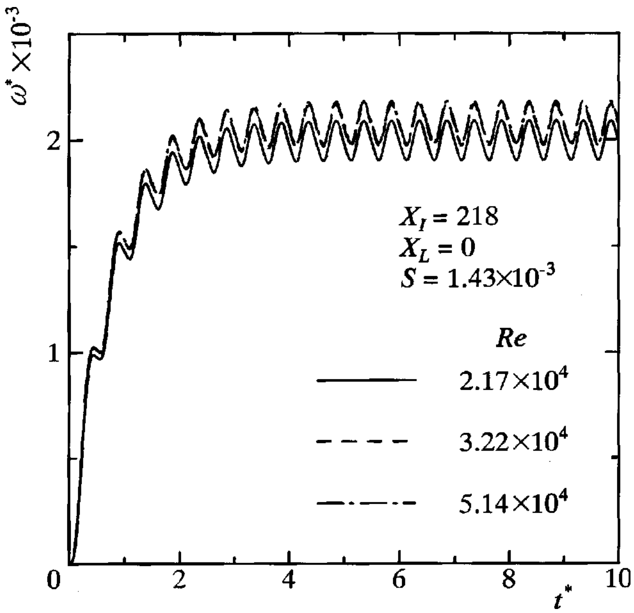

Fig. 7 Effect of Reynolds number on starting characteristics

operation is almost the same. Therefore, the starting characteristics of the turbine are independent of the Reynolds number.

Conclusions

The effects of Reynolds number and hub-to-tip ratio on the performance of the impulse turbine for wave energy conversion have been investigated experimentally by model testing under steady and sinusoidally oscillating flow conditions. As a result, it was found that the critical value of Reynolds number and the optimum hub-to-tip ratio are approximately 4.0×10^4 and 0.7, respectively. Furthermore, the starting characteristics of the impulse turbine are independent of the Reynolds number and the hub-to-tip ratio.

Nomenclature

b = blade height, m
C_A = input coefficient {Eq. (3)}
C_T = torque coefficient {Eq. (2)}
f = frequency of wave motion $=1/T$, Hz
I = moment of inertia of rotor, kg m^2
l = chord length of rotor blade, m
Q = flow rate, m^3/s
r_R = mean radius, m
S = non-dimensional frequency of wave motion $= f r_R / V_a$
t = time, s
t^* = non-dimensional time in sinusoidally oscillating flow $= t/T$
Re = Reynolds number {Eq. (5)}
T = period of wave motion $=1/f$, s
T_o = output torque, N m
T_L = loading torque, N m
U_R = circumferential velocity at r_R, m
v_a = mean axial flow velocity, m/s
V_a = reference velocity of v_a, m/s
w = relative inflow velocity, m/s
X_1 = non-dimensional moment of inertia $= I/(\rho \pi r_R^5)$
X_L = non-dimensional loading torque $= T_L/(\rho \pi V_a^2 r_R^3)$
z = number of rotor blades
Δp = total pressure drop between settling chamber and atmosphere, Pa
η = turbine efficiency under steady flow conditions {Eq. (1)}
η_p = peak value of η
ν = hub-to-tip ratio
μ = viscosity of air, Pa s
ρ = density of air, kg/m^3
ϕ = flow coefficient {Eq. (4)}
ω = angular velocity of rotor, rad/s
ω^* = non-dimensional angular velocity under sinusoidally oscillating flow conditions $= \omega/T$

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