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One-dimensional Analysis of Impulse Turbine with Self-pitch-controlled Guide Vanes for Wave Power Conversion*

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A classical one-dimensional analysis in turbomachinery was presented to estimate aerodynamic characteristics of an impulse turbine with self-pitch-controlled guide vanes which is proposed by the authors for ocean wave power conversion. With some simplified assumptions, the efficiency vs/flow-rate coefficient curves were calculated and compared with the experimental results both in a unidirectional steady flow condition and a sinusoidally oscillating flow condition. The estimated results reveal a behavior of the actual characteristics curve of the turbine. Possibility of further improvement in efficiency was discussed from a viewpoint of specific speed and specific diameter.

Keywords: Fluid machinery, Ocean wave energy, Self-rectifying-turbine, Impulse turbine, Guide vane, Wave energy conversion

INTRODUCTION

In order to overcome the global environmental problem, it is important to develop an effective technique of utilization of natural energy which is inherently low density but has no harmful effects on the earth. Various investigations have been carried out on wave energy conversion in the past. Among them Wells turbine combined with oscillating water column (OWC) is assumed to be a most promising device (Inoue *et al.*, 1986; 1988). Wells turbine is a self-rectifying-turbine and it needs no rectifying valve system. The most favorable character of the Wells turbine is relatively higher tangential blade velocity compared with axial air velocity, resulting in a compact generator size. However for higher turbine loading, for example when input of Wells turbine increases by a energy enhancing technique such as wave focusing submerged plate, the above mentioned merit becomes weak point from the viewpoints of blade stress, maintenance of mechanical parts and noise generation. Furthermore,

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efficiency of the Wells turbine is relatively lower than the ordinary turbines.

For higher wave energy density, the use of impulse turbine is more reasonable because it operates with higher pressure drop at high efficiency. The authors had proposed an impulse turbine with self-pitch-controlled guide vanes, which were set at both sides of the rotor blade. The guide vanes were forced to oscillate automatically by the reciprocating air flow to operate as nozzle and diffuser. The model tests of the impulse turbine have been carried out to develop an efficient low speed air turbine for wave power generator (Kim *et al.*, 1988; Setoguchi *et al.*, 1993; 1994; Maeda *et al.*, 1995). In this paper a simple one-dimensional theory is proposed to estimate the performance of the impulse turbine, and possibility of an improvement of the turbine performance is discussed.

BACKGROUND

In the first proposed air turbine with self-pitch-controlled guide vane (Kim *et al.*, 1988), it was found by the model test that the diffuser efficiency was low due to improper movement of the downstream guide vanes (Setoguchi *et al.*, 1993). Accordingly improved version was proposed introducing splitter vanes into the guide vanes as shown in Fig. 1.

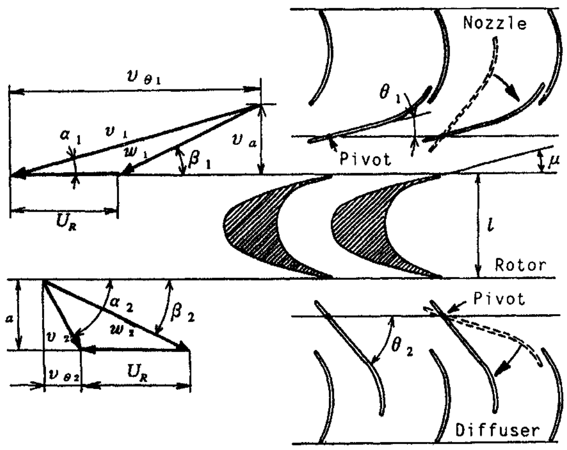

FIGURE 1 Blade geometry and velocity diagram.

In this case only the splitter vanes make an oscillating motion, upstream splitter vanes playing as nozzle and downstream ones as diffuser (Setoguchi *et al.*, 1994). As a result of the model test of this turbine in a sinusoidally oscillating flow, higher turbine efficiency was obtained than the most efficient Wells turbine (Wells turbine with guide vanes) tested in the same test rig with the same size of rotor as the impulse turbine. At the same time, it was pointed out that maximum efficiency was obtained at a higher flow coefficient (the ratio of axial flow velocity to tangential velocity of rotor blade at mid span). It means that the turbine operates at about a half-rotation speed compared with the Wells turbine. This is advantageous from the viewpoint of mechanical stress and noise generation. Therefore this turbine seemed to be worth while for practical use although it had moving parts in the guide vane system. Then systematic experimental investigations had been carried out to improve the performance, in which rotor geometry, number of blades, setting angle of the splitter vane were varied (Maeda *et al.*, 1995). As a result, turbine efficiency was not so improved as expected from ordinary impulse turbine. In order to investigate the reason for this and to discuss the possibility of improvement of turbine efficiency, the turbine performance is analyzed theoretically in the following section.

ONE-DIMENSIONAL ANALYSIS

Performance under Unidirectional Steady Flow

Although the upstream and downstream guide vanes of the present turbine make an oscillating pitching motion to operate as nozzle and diffuser, the motion occurs very quickly when flow direction changes, then the pitch angle is fixed to a certain setting angle for the most part of the period. Therefore, similar to a simple theory for ordinary turbomachine, the following assumptions can be given to analyze the turbine performance.

(1) Turbine performance is estimated from the condition at mean radius.

(2) Absolute nozzle exit flow angle α_1 is constant.
(3) Relative rotor exit flow angle β_2 is constant.

Following these assumptions and velocity diagram shown in Fig. 1, theoretical enthalpy drop of the turbine I_{th} is written as:

$$I_{th} = U_R(v_{\theta 1} + v_{\theta 2}) = U_R^2(K\phi - 1), \quad (1)$$

where U_R is the tangential blade velocity at mean radius, $v_{\theta 1}$ the swirl component of inlet absolute velocity, $v_{\theta 2}$ the swirl component of exit absolute velocity, v_a the axial velocity, $K = \cot\alpha_1 + \cot\beta_2$ and $\phi = v_a/U_R$. Loss coefficient is defined as $\zeta = \Delta I_l/(v_a^2/2)$ for enthalpy loss ΔI_l through whole turbine including rotor, upstream and downstream guide vanes. Actual enthalpy drop through the turbine I is written as:

$$I = I_{th} + \Delta I_l = U_R^2(K\phi - 1 + \tfrac{1}{2}\zeta\phi^2) \quad (2)$$

and turbine efficiency η is written as follows:

$$\eta = \frac{I_{th}}{I} = \frac{K\phi - 1}{K\phi - 1 + \zeta\phi^2/2}. \quad (3)$$

The values of K and ζ introduced above are estimated from torque coefficient C_T and input coefficient C_A obtained by performance test under steady flow condition. C_T and C_A are defined as follows, respectively:

$$C_T = T/(\rho(v_a^2 + U_R^2)blzr_R/2); \quad (4)$$

$$C_A = pQ/(\rho(v_a^2 + U_R^2)blzv_a/2), \quad (5)$$

where T is the turbine shaft torque, ρ the density of air, b the span length of rotor blade, l the chord length of rotor blade, z the number of rotor blade, r_R the mean radius, p the total pressure drop through turbine and Q the flow rate. Output and input powers are given as follows, respectively:

$$\rho v_a A I_{th} = T(U_R/r_R); \quad (6)$$

$$\rho v_a A I = pQ, \quad (7)$$

where A is the cross sectional area of flow passage and $\rho v_a A$ the mass flow rate. From Eqs. (1), (2), (4)–(7) and defining solidity as $\sigma = blz/A = l/(2\pi r_R/z)$, the equations for K and ζ are obtained:

$$K = \frac{1}{2}\sigma C_T\left(1 + \frac{1}{\phi^2}\right) + \frac{1}{\phi}; \quad (8)$$

$$\zeta = \sigma\left(C_A - \frac{C_T}{\phi}\right)\left(1 + \frac{1}{\phi^2}\right). \quad (9)$$

With Eqs. (8) and (9) into Eq. (3), turbine efficiency is obtained as follows:

$$\eta = \frac{C_T}{C_A\phi} \quad (10)$$

which is the same as the ratio of Eq. (6) to Eq. (7).

According to our experiments, there was considerably a wide range of ϕ where K and ζ were nearly constant as will be shown in the next section. Therefore, for simplicity, K and ζ are assumed to be constant to estimate an approximate characteristics of this turbine. In this case, the maximum efficiency η_{max} is obtained at $\phi = 2/K$:

$$\eta_{max} = \frac{1}{1 + (2\zeta/K^2)}. \quad (11)$$

Performance under Reciprocating Flow

In this section the turbine performance under reciprocating flow is estimated by quasi-steady analysis with K and ζ obtained in steady flow experiment.

When axial velocity v_a changes sinusoidally with amplitude V_a and period T_w, ϕ is written as

$$\phi = \Phi \sin\varphi, \quad (12)$$

where $\Phi = V_a/U_R$, $\varphi = 2\pi t/T_w$. Then mean turbine efficiency $\tilde{\eta}$ is calculated from

$$\tilde{\eta} = \int_0^{2\pi} I_{th}|v_a|\,d\varphi \Big/ \int_0^{2\pi} I|v_a|\,d\varphi. \quad (13)$$

Introducing Eqs. (1), (2) and (12), and executing the integration assuming K and ζ to be constant again,

$$\tilde{\eta} = \frac{(\pi/4)K\Phi - 1}{(\pi/4)K\Phi - 1 + \zeta\Phi^2/3}. \tag{14}$$

The maximum efficiency is obtained at $\Phi = 8/(\pi K)$:

$$\tilde{\eta}_{\max} = \frac{1}{1 + 64\zeta/(3\pi^2 K^2)}. \tag{15}$$

COMPARISON WITH EXPERIMENTAL RESULTS

Table I shows the geometrical parameters of the test turbines investigated in the past (Setoguchi *et al.*, 1994). Among them relatively higher efficiency was obtained by the turbine with the following combination of parameter: $\mu = 30°$, $\theta_1 = 15°$, $\theta_2 = 50°$, $Z_g = 20$. On the other hand, efficiency was moderate with: $\mu = 20°, 30°$, $\theta_1 = 15°$, $\theta_2 = 50°$, $z_g = 18$. These two types were selected to discuss the validity of the present one-dimensional theory.

Figure 2(a) and (b) show plots of K and ζ against ϕ, respectively, where the values of K and ζ are calculated from torque coefficient C_T and input coefficient C_A obtained under steady flow turbine test. The points indicated by circle in the figure are the operating point for maximum efficiency. It is observed that the value of K is almost constant except for very low ϕ, and this means the validity of assumption of the present one-dimensional theory. On the other hand, the value of ζ is roughly constant at high flow-rate coefficients, but increases remarkably with ϕ decreasing at low flow range including maximum efficiency point. In order to discuss the reason for this unexpected high ζ in low flow range, inlet and exit velocity diagram of the rotor was investigated from experimental data at mean radius. As a result, it was found that exit

TABLE I Geometrical parameters of test turbine

Rotor blade (Impulse blade)	
Suction side	Elliptic-arc
Pressure side	Circular-arc
Chord length	$l = 52$ mm
Solidity	$\sigma = 2.0$
Blade angle	$\mu = 20°, 30°, 40°$ (see Fig. 1)
Guide vane (with splitter blade)	
Fixed vane:	
Blade section	Circular-arc
Camber angle	70°
Chord length	36.7 mm
Blade thickness	1.5 mm
Movable vane:	
Blade section	Circular-arc (camber angle of 45°) and straight
Chord length	50 mm
Blade thickness	1.5 mm
Number of blade	$z_g = 15 \sim 30$
Setting angle of nozzle	$\theta_1 = 10° \sim 20°$
Setting angle of diffuser	$\theta_2 = 15° \sim 70°$
Tip dia.	298 mm
Hub-to-tip ratio	0.7

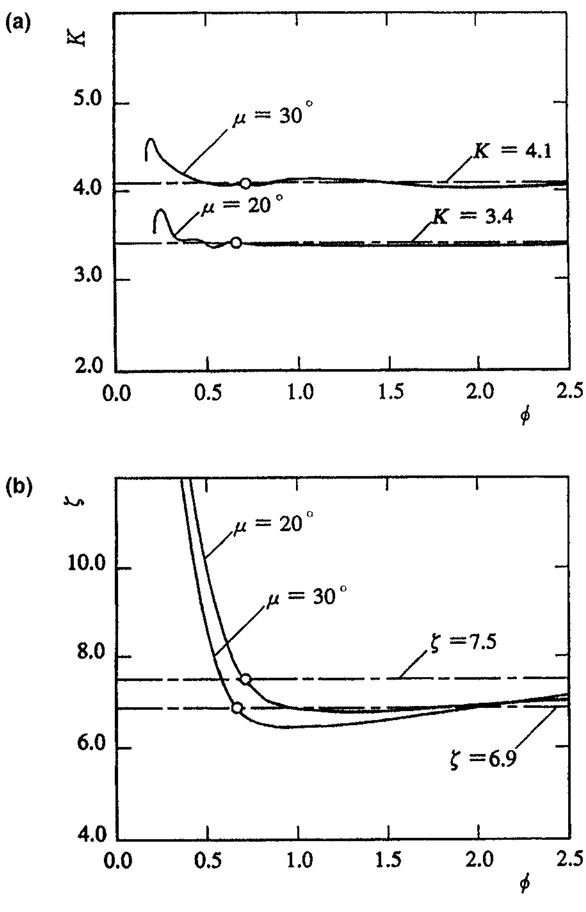

FIGURE 2 (a) $K-\phi$ characteristics for steady flow. (b) $\zeta-\phi$ characteristics for steady flow.

absolute flow angle α_2 was higher than 90° near maximum efficiency point. This led to ill matching condition with the downstream guide vane, and diffuser efficiency decreased. The lower the flow coefficient is, the higher α_2 becomes, and the lower efficiency becomes.

According to the assumption of the present theory, turbine efficiency can be estimated by Eq. (3) if the values of K and ζ were given. Figure 3 shows the theoretical efficiency comparing with experiment. In this case the values of K and ζ were kept constant to the values at the maximum efficiency points (dot dashed line in Fig. 2), respectively. It is observed from this figure that the present simplified theory can predict the behavior of efficiency-flow rate characteristics quite well. Discrepancy between theory and experiment becomes noticeable in low flow range due to higher loss coefficient in that region.

From the velocity diagram discussed above, it is expected that exit absolute flow angle α_2 may fit with the downstream diffuser angle θ_2 for much higher flow coefficient than that for maximum efficiency point. But experimental and theoretical values of efficiency at such a high flow coefficient are low. The reason for this is concerned with the values of specific speed n_s and specific diameter d_s for axial impulse turbine. According to Balje (1962), high efficiency can be realized only within a very limited area on $n_s - d_s$ diagram for axial impulse turbine. Figure 4 shows the $n_s - d_s$ diagram by Balje. The relation between the flow coefficient defined in the present theory and the values of n_s and d_s by Balje is written as follows:

$$\phi = 16/(\pi(1+\nu)^2(1-\nu)n_s d_s^3), \qquad (16)$$

where ν is hub ratio. The dot dashed line in the figure indicates the relation for $\nu = 0.7$, that is the value for the test turbine. From Eq. (16) it is observed that high efficiency can be obtained for $\phi = 0.2 \sim 0.8$ and efficiency drops noticeably for $\phi > 1.5$. Therefore it is important to select optimum values of n_s and d_s for the design of the present impulse turbine, and it is also noted that one cannot discuss performance of the present turbine only from a compatibility of the velocity diagram.

In Fig. 5, the mean turbine efficiency $\bar{\eta}$ from Eq. (14) for oscillating flow is compared with experiment. It is shown that the efficiency characteristics can be well estimated qualitatively. Drop in turbine efficiency for oscillating flow is only about 2% compared with steady flow. This is a preferable character of the present impulse turbine.

Finally a possibility of improving efficiency of the present turbine is discussed. In Fig. 6 the value of maximum efficiency is plotted from Eq. (11). To get higher efficiency higher value of K and lower value of ζ are desirable. The value of K depends on nozzle exit angle θ_1 and rotor angle μ. Experimentally the highest value of $K(=4.1)$ was obtained at $\theta_1 = 15°$, $\mu = 30°$. It may be possible to increase K to some degree if exit deviation angle could be reduced by optimum rotor design. On the other hand it is most important to reduce loss coefficient at maximum efficiency point, $\phi = 2/K$. However the relations $1/\phi = \cot\alpha_1 - \cot\beta_1 = \cot\beta_2 - \cot\alpha_2$, $K = \cot\alpha_1 + \cot\beta_2$ must hold, and then $\cot\alpha_2 \approx -\cot\beta_1$ must be satisfied near maximum efficiency point. This means that it is necessary to design a diffuser with angle θ_2 which fits for absolute flow angle $\alpha_2 > 90°$. This may be difficult from structural standpoint for the present turbine in which guide vanes make a pitching motion automatically.

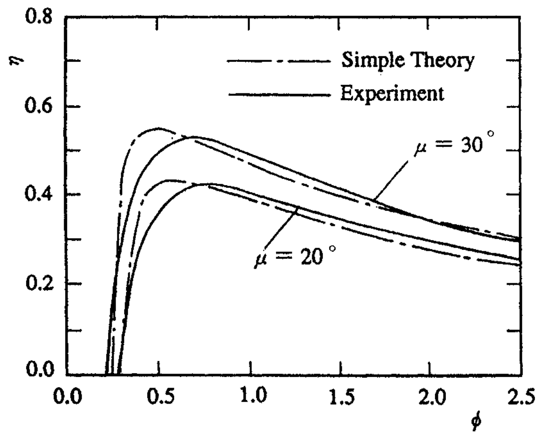

FIGURE 3 Efficiency characteristics for steady flow.

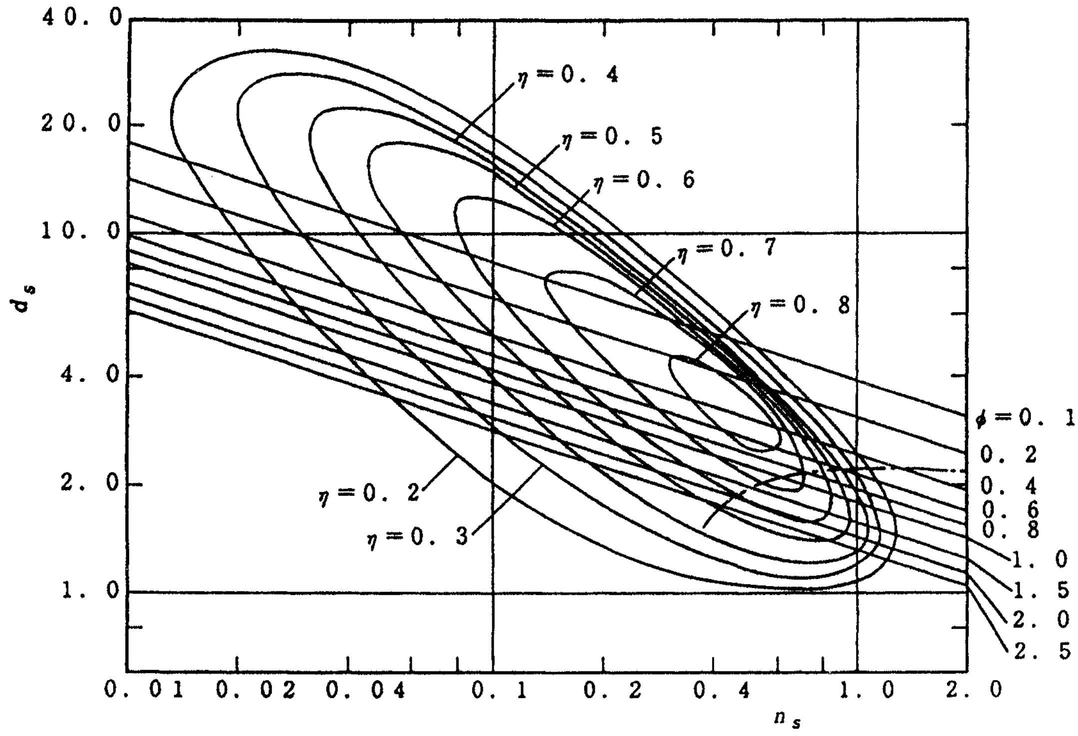

FIGURE 4 $n_s - d_s$ diagram for impulse turbine.

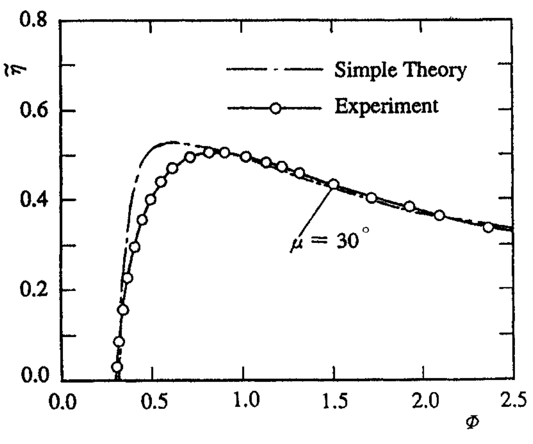

FIGURE 5 Efficiency characteristics for oscillating flow.

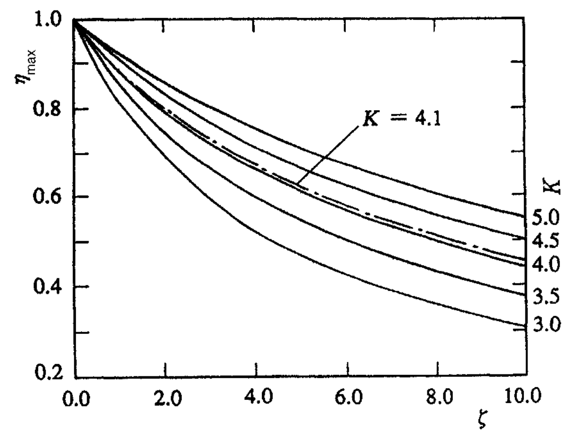

FIGURE 6 Relation between maximum efficiency and loss coefficient.

SUMMARY

In this paper a simplified one-dimensional theory was proposed to estimate the performance of the impulse turbine for wave energy generator based on a classical turbine theory. The present theory can explain the efficiency characteristics quite well. The theoretical value was compared with experimental value and some measures for improving the turbine efficiency were discussed. Main points mentioned are optimum selection of specific speed and specific diameter, rotor design with lower deviation angle and improvement of inlet condition to downstream guide vane.

Acknowledgements

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