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A 3-Dimensional Numerical Simulation of Impulse Turbine for Wave Energy Conversion

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Key words : Impulse turbine, wave energy conversion, sliding mesh, Computational Fluid Dynamics

Abstract

This paper describes numerical analysis of the impulse turbine with fixed guide vanes, a high performance bi-directional air turbine having simple structure for wave energy conversion. A 3-dimensional incompressible viscous flow numerical analysis based on the full Reynolds-averaged Navier-Stokes equations was made to investigate the internal flow behavior. Numerical results are compared with experimental data. As a result, a suitable choice for the one of design factors has been clarified.

1. Introduction

For wave energy conversion, the air turbine, which is called Wells turbine, has been investigated for many years. It has been widely applied for ocean-wave energy conversion, mainly due to its simple structure at present. The Wells turbine has however some inherent shortcomings: relatively lower efficiency, maintenance, higher axial thrust and noise in the case of high power specification because the circumferential speed of the rotor is essentially quite high.[1,2]

The impulse turbine with self-pitch-controlled guide vanes has recently been proposed to overcome these drawbacks. It was clarified that the turbine can be operated with higher turbine efficiency and lower rotational speed than those of the Wells turbine.[5]

The impulse turbine with self-pitch-controlled guide vanes has a disadvantage of maintenance of the pivots on which the guide vanes are rotated automatically in a bi-directional air flow. Therefore, the authors studied the impulse turbine with fixed guide vanes to obtain higher efficiency than that of

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Wells turbine in a lower rotational speed.

A numerical simulation was carried out to grasp a relationship between the characteristics of the impulse turbine and its internal flow structures on a condition that the flow is under steady.

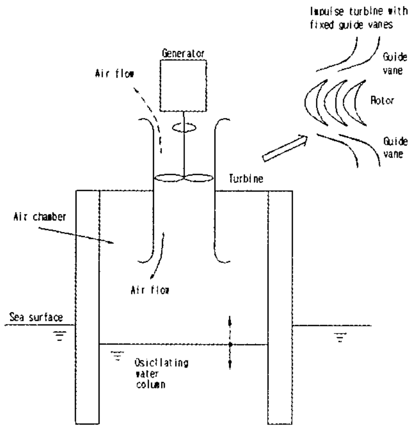


Fig. 1 Schematic of wave energy converter.

2. Numerical Analysis

2.1 Governing Equation

Three-dimensional analysis was carried out for the internal flow of the impulse turbine with the fixed guide vanes using the full Reynolds Averaged Navier-Stokes equation given as the below.

Continuity equation :

$$\frac{\partial(\rho U_i)}{\partial X_i}=0 \tag{1}$$

RANS equation:

$$\frac{\partial(\rho U_i U_j)}{\partial X_i}=-\frac{\partial P}{\partial X_i}+\frac{\partial}{\partial X_j}\left[\mu\left(\frac{\partial U_i}{\partial X_j}+\frac{\partial U_j}{\partial X_i}-\frac{2}{3}\delta_{ij}\frac{\partial U_i}{\partial X_i}\right)\right]+\frac{\partial}{\partial X_j}(-\rho\,\overline{U_i U_j}) \tag{2}$$

Where ρ is the air density, U_i the mean velocity in the X_i directions, p the pressure. Reynolds Stress Terms, $\rho\,\overline{U_i U_j}$ were closed by the two-equation turbulence model[3].

2.2 Grid system and boundary conditions

Periodic and zonal interfacing conditions were adopted to the boundary between the guide vane and the rotor. Absolute values were given as the inflow velocity conditions of the guide vanes and relative values for the rotor. A commercial code FLUENT 5.4 was used for the simulation for its versatility on complex geometry. The governing equations were solved based on the Finite Volume Method. QUICK scheme was used to calculate the convection terms. SIMPLE algorithm was used for the continuity equation correcting the pressure and the velocity components. The numerical results of the flow simulation were influenced by the grid parameters, i.e. grid structure, grid distribution, number of grid. The impulse turbine with fixed guide vanes and rotor blades are complex geometry. The total grid numbers are 360,000 and non-staggered grid system was adapted as shown in Fig. 2.

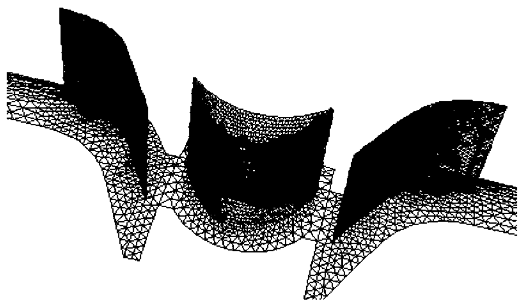


Fig. 2 Computational grid system

3. Results and discussion

The characteristics of the impulse turbine are obtained under steady state flow conditions. The results are expressed in the forms of torque coefficient C_T, input coefficient C_A, and turbine efficiency η against flow coefficient, Φ . The definitions are as follows:

$$C_T = T / \left\{ \rho (v_a^2 + U_R^2) b l z r_R / 2 \right\} \tag{3}$$

$$C_A = \Delta p Q / \left\{ \rho (v_a^2 + U_R^2) b l z v_a / 2 \right\} \tag{4}$$

$$\phi = v_a / U_R \tag{5}$$

$$\eta = T\omega / (\Delta p Q) = C_T / (C_A \phi) \tag{6}$$

Where T is the output torque, ρ the air density, v_a the mean axial velocity, U_R the circumferential velocity at r_R, b the blade height, l the chord length of rotor blade, z the number of rotor blades, r_R the mean radius of rotor, Δp the total pressure drop across the rotor, Q the volumetric flow rate.

3.1 2-D Analysis

Kim et al. (2000) investigated the flow characteristics of the internal flow field of the impulse turbine and reported an optimal installation angle of the guide vanes for higher efficiency through a test on five installation cases between 15° and 45°[6]. Figure 3 shows that the turbine has high efficiency near Φ = 0.25 in case of θ =15° and near Φ= 0.5 in case of θ =30° and 45°. When flow coefficient is greater than 0.5, the efficiency drops largely. We know from both numerical and experimental results that the optimum installation angle is θ =30°.

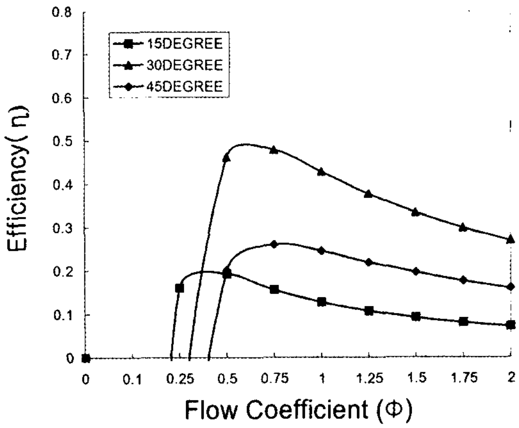
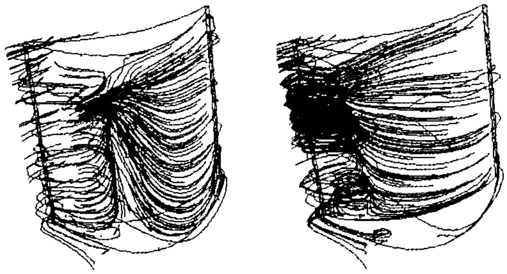
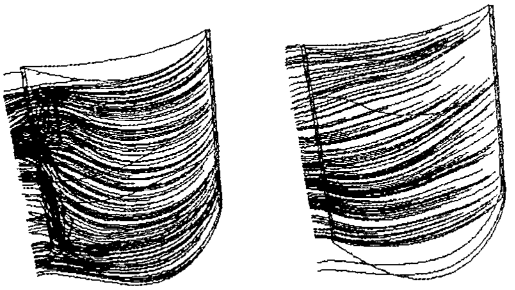


Fig. 3 Efficiency under steady flow condition

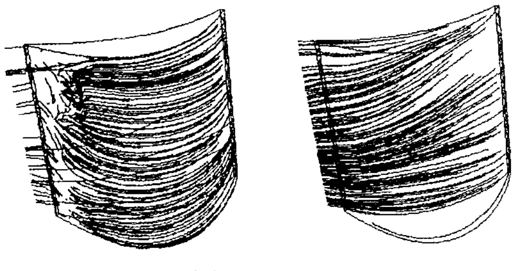
Figure 4 shows streamline distribution in the suction and the pressure side according to the change of flow field of the 3-D impulse turbine. In case of Φ =0.25 circumferential velocity is faster than axial velocity, and flow stream toward tip direction and flow acceleration and separation of turbines internal flow occur over the surface of the trailing edge of suction side through the effect of centrifugal force due to rotors rotation. Flow stream toward axial direction, because circumferential velocity is similar to axial velocity in case of Φ = 0.75, and although the axial velocity is more fast than circumferential velocity, small flow separations occur due to centrifugal force in the trailing edge of suction side in case of Φ =1.25. These affect the efficiency. We can observe that main flow is produced along the rotor surface as flow coefficient increases.



(a) Φ=0.25



(b) Φ =0.75



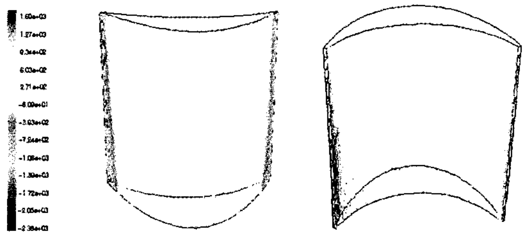
(c) Φ=1.25

(i) Suction side (ii) Pressure side

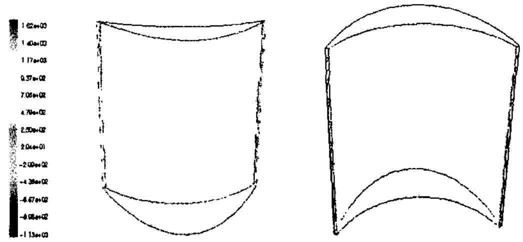
Fig. 4 Streamlines on the rotor blades

3.2 3-D Analysis

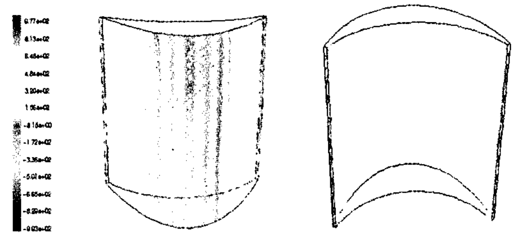
Figure5 shows the pressure distribution on the suction and the pressure side of the rotor. As flow coefficient increases, the pressure in the leading edge of the rotor is highest and there are more pressure changes in the suction side because of flow changes and rotor rotation. It can be said from the pressure distribution on the pressure side that flows in the both ends of the rotor are delayed and this makes less efficiency.



(a) Φ=0.25



(b) Φ=0.75



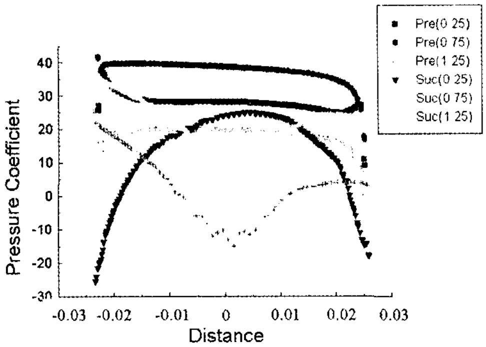
(c) Φ=1.25

(i) Suction side (ii) Pressure side

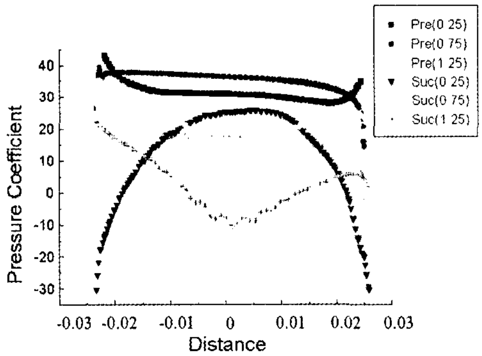
Fig. 5 Pressure distributions on the suction and pressure sides

Figure 6 shows the distributions of pressure coefficient on the suction and pressure sides at radius R=0.22m, R=0.24m and 0.25m. The fact that the pressure difference is small in the central part and large in the both ends in case of Φ = 0.25, makes less efficiency. It is similar to Φ = 0.75 generally in case of Φ

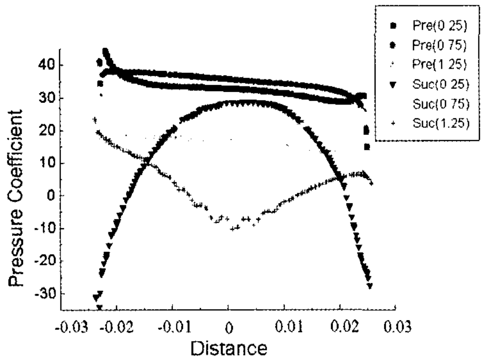
= 1.25, but the higher pressure in the suction side than in the pressure side near the leading and trailing edge disturbs either turbine rotation, so the efficiency results low.



(a) R=0.22m



(b) R=0.24m

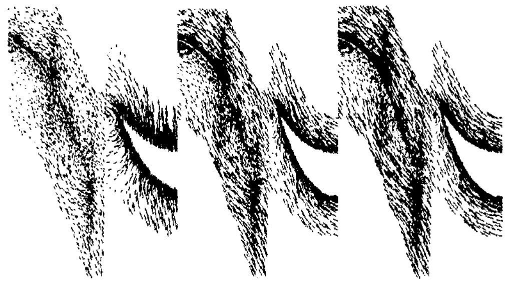


(a) R=0.25m

Fig. 6 Distributions of pressure coefficient.

(i) $\Phi=0.25$ (ii) $\Phi=0.75$ (iii) $\Phi=1.25$

(a) R=0.22m (near hub)



(i) $\Phi=0.25$ (ii) $\Phi=0.75$ (iii) $\Phi=1.25$

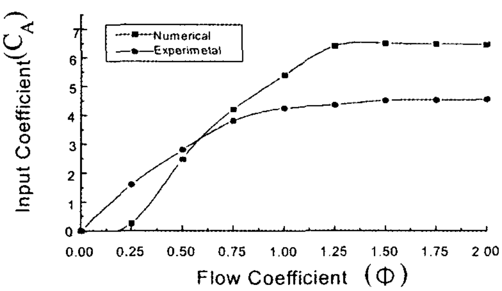
(b) **R=0.24m (near mid-radius)**

(i) $\Phi=0.25$ (ii) $\Phi=0.75$ (iii) $\Phi=1.25$

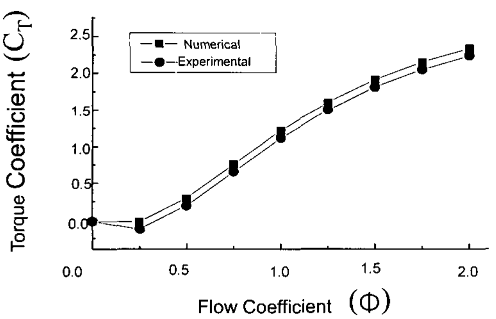
(c) R=0.25m (near tip)

Fig. 7 Relative velocity distributions.

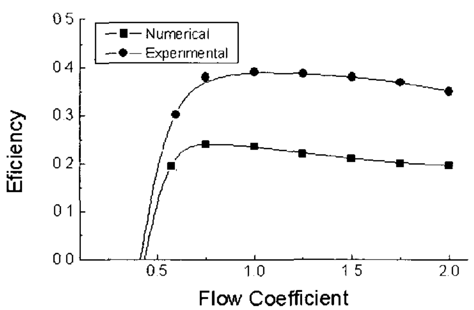
Figure 7 shows velocity distribution around the under guide vane and in the rotor part, in which flows change most rapidly. If we observe velocity distribution at smaller flow coefficient, it is estimated that negative torque will occur. And we can see that flows change rapidly in the suction side of the rotor and under the guide vane as flow coefficient increases and it goes toward the tip. It is also observed that small flow separations occur in the trailing edge of the suction side and vortex occurs in under the guide vane. And the range will be wide as flow coefficient increases.



(a) Input coefficient



(b) Torque coefficient



(c) Efficiency

Fig. 8 Turbine characteristics under steady flow condition

Figure 8(a) shows total pressure drop variations against the flow coefficients. It is seen that the input coefficient increases as the flow coefficient increases. Figure 8(b) shows that the torque coefficient becomes negative at low flow coefficient and it increases again with increase of the flow coefficient. Acceleration of the turbine is not attainable with high rotor speed, but it can be seen that the torque overcomes the negative ranges and reaches normal operating conditions due to the increase of turbines inertia. Figure 8(c) shows the efficiency curve. It is seen that the efficiency decreases regardless of high C_A and C_T at the region of $\phi > 0.75$, of which reasons are thought as the existence of separation and trailing vortex of the edges and the guide vane. The trend of the curve is similar to those of the experimental data though some discrepancy is still seen quantitatively[7].

4. Conclusions

A numerical analysis has been carried out for practical design of the impulse turbine through an investigation into the characteristics of the internal flows. It was cleared that the optimal angle of installation for the guide was 30° through a comparison of the data obtained by 2-D numerical analysis with those obtained by experimental data. Followings are the summaries not concerning quantitatively but concerning qualitatively between numerical results and experimental ones.

(1) The results of 3-D numerical analysis

showed good agreements qualitatively with experimental ones.

(2) In cases of ϕ=0.25 and 1.25, flow separations were seen in the trailing edge of the suction side, which will be eventually contribute to the decrease of the turbine efficiency.

(3) The efficiency is highest in case of flow coefficient Φ =0.5~0.75, which implies that the balanced pressure distribution plays a vital role in transferring torques to the rotor most effectively.

5. References

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