Proceedings of the
International Conference on Mechanical Engineering and Renewable Energy 2011
(ICMERE2011) 22- 24 December 2011, Chittagong, Bangladesh

ICMERE2011-PI-007

A STUDY ON THE PERFORMANCE OF TWIN IMPULSE TURBINE TOPOLOGY FOR WAVE ENERGY CONVERSION

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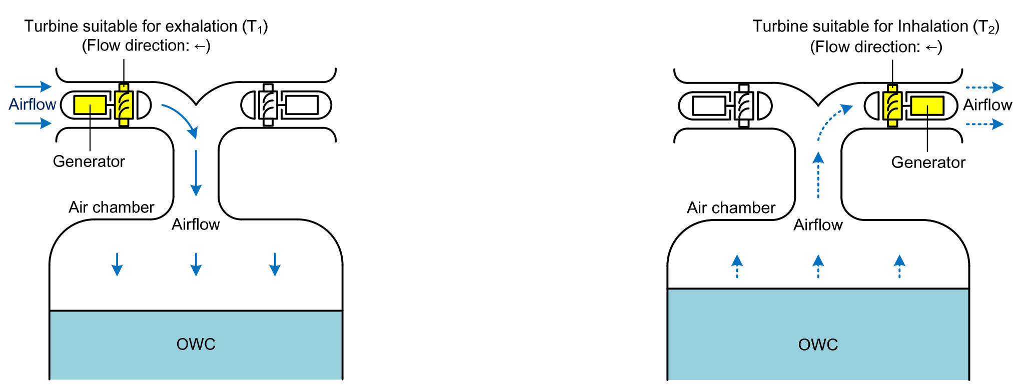
Abstract*-A novel twin unidirectional impulse turbine has been introduced in order to enhance the performance of OWC based wave energy plant. This new topology of turbine system uses two unidirectional impulse turbines and their flow direction is different each other. However, the turbine characteristics have not been clarified to date. The performances of a unidirectional impulse turbine under steady flow conditions were investigated experimentally by using a wind tunnel with large piston/cylinder in this study. Guide vanes with two different setting angles, θ = 14º and 20º were employed in order to determine the effects of setting angles on the proposed unidirectional impulse turbine. Furthermore, efficiency of the turbine has been estimated by a quasi-steady analysis using experimental results.*

Keywords: Fluid machinery, Impulse turbine, Wave energy conversion, Ocean energy

1. INTRODUCTION

The tremendous potential of converting and utilizing energy from ocean has fascinated people for more than hundred years. One of the to day's methods for wave energy conversion utilizes an Oscillating Water Column (OWC) invented by a Japanese Navy officer, Yoshio Masuda. In recent years, the power take-off mechanism in the OWC based wave energy plant has generally comprised of a bidirectional turbine, such as Wells turbine. An alternative to the use of a bidirectional turbine is an approach utilizing a unidirectional turbine with two or four valves that act to force air into the turbine in the same direction during exhalation or inhalation from air chamber. Although it has had success in navigational buoys for 30 years, the principle was not adopted in larger plants.

However, a unidirectional impulse turbine exhibited good efficiency over a broad range [1]. A new topology with twin impulse turbine has been proposed, which has a promise of yielding 50% wave to wire efficiency in an OWC configuration [2]. A laboratory model of the concept was built at IIT Madras, India. The basic arrangement of the twin unidirectional topology is shown in Fig. 1. The power take-off mechanism utilizes two unidirectional turbines, T_1 and T_2 coupled to electrical generators. It was conjectured that during intake, the air would flow into the Oscillating Water Column via the turbine T_1 and during exhaust the air would flow out from the Oscillating Water Column through T_2. The use of twin unidirectional turbines ensures that, even without the use of valves, each turbine in effect works for half a cycle.



(a) Inhalation (b) Exhalation

Fig.1: Basic arrangement of twin unidirectional Impulse turbine topology

As the first step of the study on the proposed twin turbine topology, the performances of unidirectional impulse turbine under steady flow conditions have been investigated experimentally in this study. Further, efficiency of the twin impulse turbine has been estimated by a quasi-steady analysis using experimental results.

2. EXPERIMENTAL SETUP AND PROCEDURE

The experimental setup and procedure are the same to Refs. [1, 3]. A schematic view of the test rig is shown in Fig.2. The test rig consists of a large piston-cylinder (diameter: 1.4m, length: 1.7m), a settling chamber and a 300-mm-diameter test section with the inlet and outlet bell-mouth. The turbine rotor was placed at the center of the test section and tested at a constant rotational speed under steady and sinusoidal flow conditions. In the steady flow condition, the flow rate through the turbine was measured by the Pitot tube. The overall performance of unidirectional impulse turbine was evaluated by torque T_o, flow rate q, rotor angular velocity ω, and total pressure drop through the turbine Δp. The tests were performed for the total pressure drops up to 4.0kPa, the flow rates up to 0.320m^3/s, and the angular velocities up to 370rad/s. The Reynolds number based on rotor blade chord was approximately equal to 8×10^4 at the peak efficiency.

Concerning the tested rotor profile, the chord length of rotor blade: 54mm; height: 44mm; tip diameter: 298mm; tip clearance: 1mm; solidity at mean radius: 2.02; radius of leading edge: 1.5mm; radius of trailing edge: 0.5mm; hub to tip ratio: 0.7. The profile is shown in Fig. 3. The guide vane profile used in this study consists of a straight line and a circular arc as shown in Fig. 4. The chord length of guide vanes: 70mm; thickness: 0.5mm; solidity at mean radius: 1.74. The setting angles of guide vanes are θ = 14º and 20º, in order to investigate the effect of guide vane setting angle on the performance. The specifications of guide vanes are shown in Table 1.

Table 1: Specifications of guide vanes

θº	R mm	δº
14	32	76
20	33.5	70

3. EXPERIMENTAL RESULTS

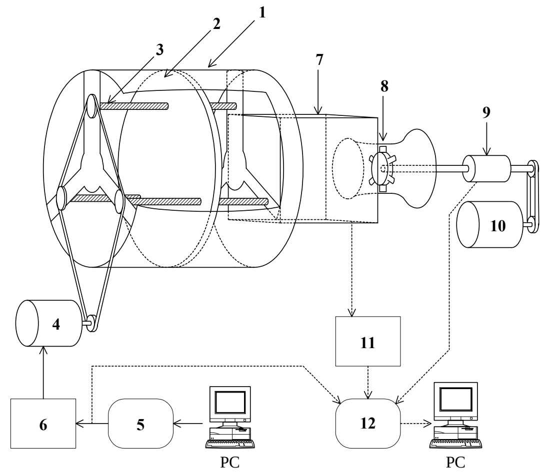
The turbine performance under steady flow conditions is evaluated by torque coefficient C_T, input coefficient C_A and turbine efficiency η against flow coefficient ϕ. The definitions of these parameters are as follows:

$$C_T = T_o/\{\rho(v^2+u^2)Ar/2\} \tag{1}$$

$$C_A = \Delta pQ/\{\rho(v^2+u^2)Av/2\} \tag{2}$$

$$\eta = T\omega/(\Delta pQ) = C_T/(C_A\phi) \tag{3}$$

$$\phi = v/u = (q/A)/(r\omega) \tag{4}$$



1 Wind tunnel
2 Piston
3 Ball-screw
4 Servomotor
5 D/A converter
6 Servo-pack
7 Settling chamber
8 Turbine
9 Torque transducer
10 Servomotor-generator
11 Pressure transducer
12 A/D converter

Fig.2: Experimental setup

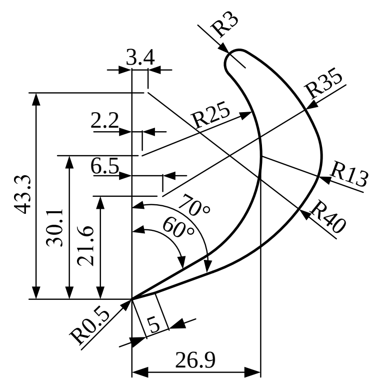


Fig.3: Rotor blade profile

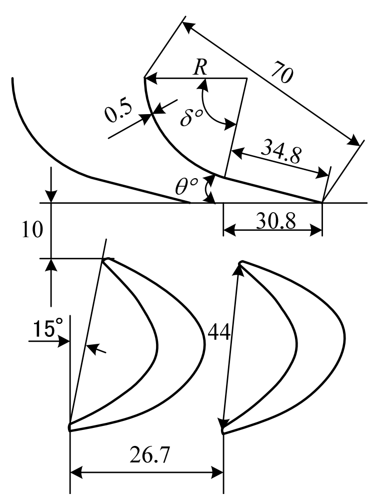


Fig.4: Turbine configuration

where A, u, v, z and ρ denote flow passage area, circumferential velocity at mean radius r, axial flow velocity, number of rotor blades and density of air, respectively.

Figure 5 shows the experimental results. The torque coefficient C_T and the efficiency η in the case of reverse flow are quite low in comparison with the case of forward flow, especially the peak efficiency is approximately 3% only in the case of reverse flow in spite of different setting angle of the guide vane. The peak efficiency in the case of $\theta = 20^{\circ}$ is higher than that of $\theta = 14^{\circ}$ and its peak value is 0.72. However, it is important to compare the pressure drop across the turbine, because of the flow rate in reverse flow through the turbine. The reverse flow should be minimized in order to obtain higher power output by another turbine.

The input coefficient C_A of the turbine under investigation is shown in Fig.5(b). In case of reverse flow the input coefficient is quite higher than that of forward flow. The coefficient C_A in the case of guide vane setting angle $\theta = 14^{\circ}$ is higher than the case of setting angle $\theta = 20^{\circ}$ in spite of same (forward) flow direction.

3.1 Effect of Setting Angle of Guide Vane on turbine Characteristics under Sinusoidal Flow Conditions

Since the airflow into the turbine is generated by the OWC, it is very important to demonstrate the turbine characteristics under oscillating flow conditions. Here let us simulate the characteristics under sinusoidal flow conditions (Fig.6) in order to clarify the effect of guide vane setting angle on the turbine characteristics.

The steady flow characteristics of the turbine as shown in Fig. 5 are assumed to be valid for computing performance under unsteady flow conditions. Such a quasi-steady analysis has been validated by previous studies [3].

In the calculation, flow rates through the two turbines are obtained by using the steady flow characteristics and solving these simultaneous equations.

$$q = q_1 + q_2 \tag{5}$$

$$\Delta p_1 = \Delta p_2 = \Delta p \tag{6}$$

$$\phi_1 = v_1/u_1 = (q_1/A)/(r\omega) \tag{7}$$

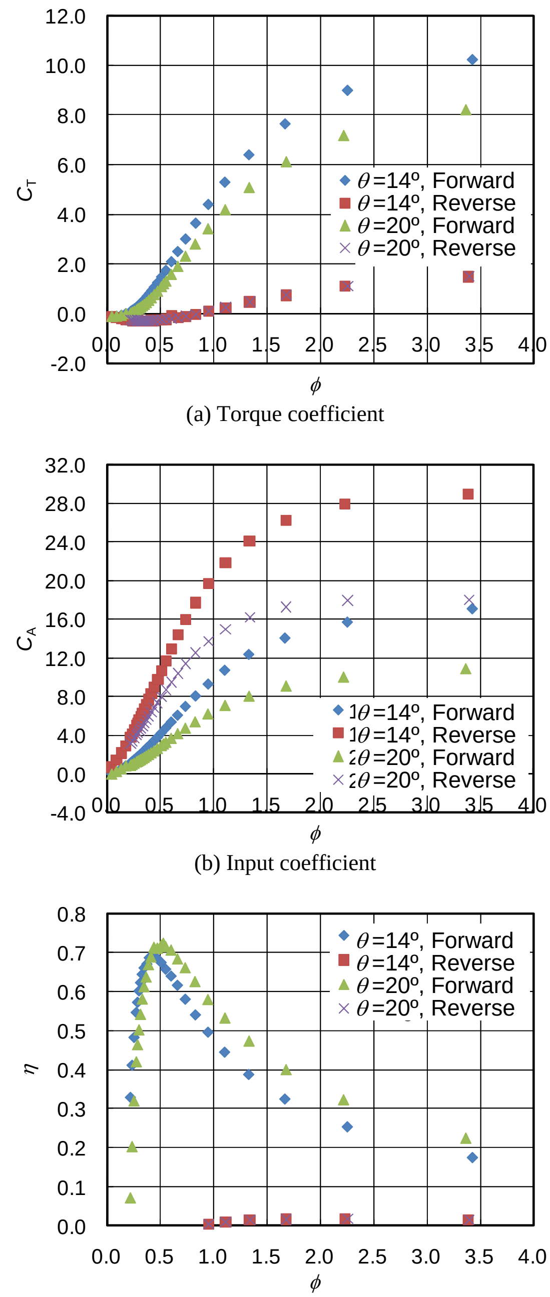
$$\phi_2 = v_2/u_2 = (q_2/A)/(r\omega) \tag{8}$$

where q denote flow rate through the turbine and subscripts 1 and 2 mean turbines T_1 and T_2, respectively (see Fig.1). Further, flow rate and rotor angular velocity are assumed as follows:

$$q = Q_0 \sin(2\pi t/T) \tag{9}$$

$$\omega_1 = \omega_2 = \omega = \text{const.} \tag{10}$$

where Q_0, t and T are maximum flow rate, time and period of wave.



(a) Torque coefficient

(b) Input coefficient

(c) Efficiency

Fig.5: Effects of flow direction and guide vane setting angle on turbine characteristics

3.2 Running Characteristics of turbine under Sinusoidal Flow Conditions

When the turbine is in the running conditions, the parameters such as T_o, ω, Δp and q vary periodically in a sinusoidal oscillating flow. In this case, the turbine performances should be represented by mean value such as mean efficiency. Assuming that only the turbine under forward flow condition operates, the running characteristics of the twin turbine under sinusoidal flow conditions are evaluated by mean efficiency η_m against the flow coefficient Φ, which are defined as follows:

$$\eta_m = \frac{\frac{1}{T}\left(\int_0^{T/2} T_{o1}\omega dt + \int_{T/2}^{T} T_{o2}\omega dt\right)}{\frac{1}{T}\int_0^T \Delta p(q_1 + q_2)dt} \tag{11}$$

$$\Phi = (Q_0/A)/u \tag{12}$$

Figure 7 shows the effect of guide vane setting angle on mean efficiency under sinusoidal flow condition. It is found from the figure that the efficiency for the turbine with $\theta = 20^o$ is superior to the case of $\theta = 14^o$ and its peak value is 0.45. However, it decreases by 0.27 in comparison with the peak efficiency under steady flow condition. This is because of some airflow passes through the turbine under reverse flow condition and the wave energy plant loses pneumatic energy which is converted from ocean wave.

4. CONCLUSIONS

As the first step of study on the proposed twin turbine topology, regarding a unidirectional impulse turbine, the effects of flow direction and guide vane setting angle on turbine characteristics were investigated experimentally in this study. Moreover, the performances of twin unidirectional impulse turbine have been estimated by a quasi-steady analysis using experimental results.

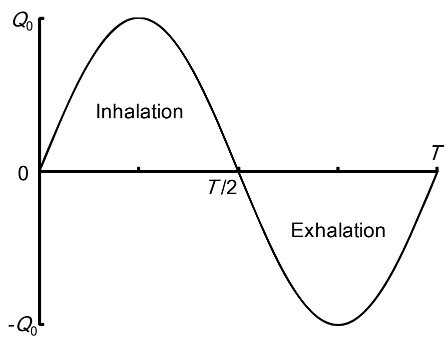


Fig.6: Sinusoidal flow

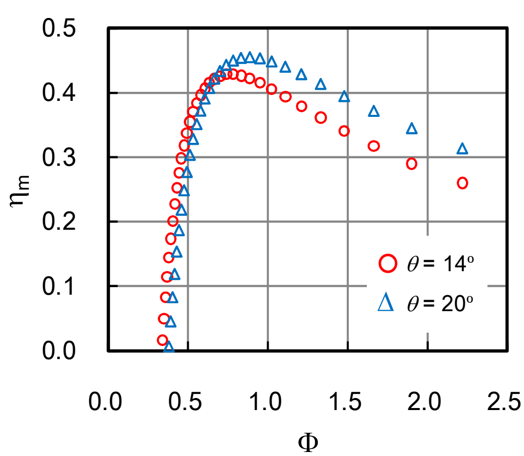


Fig.7: Effect of guide vane setting angle on mean efficiency under sinusoidal flow condition

The results are summarised as follows:

(1) The torque coefficient and the efficiency in the case of reverse flow are quite low compared with those of forward flow.

(2) The input coefficient in the case of reverse flow is quite higher than that of forward flow.

(3) The turbine with $\theta = 20^o$ seems to have better performance than that of the turbine with $\theta = 14^o$.

5. REFERENCES

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6. NOMENCLATURE

Symbol	Meaning	Unit
A	Flow passage area	(m^2)
C_A	Input coefficient defined by Eq.2	Dimentio-nless
C_T	Torque coefficient defined by Eq.1	Dimentio-nless
Δp	Total pressure drop through the turbine	(Pa)
q	Flow rate	(m^3/s)
Q_o	Maximum flow rate	(m^3/s)
R	Radius of curvature of guide vane	(m)
r	Mean radius	(m)
t	Time	(sec)
T	Period of wave	(sec)
T_o	Torque	(Nm)
u	Circumferential velocity at mean radius r	(m/s)
v	Axial flow velocity	(m/s)
δ	Subtended angle	(º)
θ	Guide vane setting angle	(º)
η	Turbine efficiency	Dimentio-nless
η_m	Mean turbine efficiency	Dimentio-nless
ϕ	Flow coefficient defined by Eq.4	Dimentio-nless
Φ	Flow coefficient defined by Eq.12	Dimentio-nless
ρ	Density of air	(kg/m^3)

ω	Rotor angular velocity	(rad/s)