



Transient simulation of OWC impulse turbine based on fully passive flow-driving model



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ABSTRACT

The rapid development of air turbines for oscillating-water-column wave energy conversion proposes a crucial demand for deep understanding of its unsteady performance in the reciprocating air flow condition. A 3D unsteady numerical model is presented in this study, which is established on the platform of commercial computational fluid dynamics (CFD) software ANSYS-Fluent 12.0. The passive rotation of turbine rotor is fully driven by the bi-directional air-flows following the Newton's second law. The numerical model is validated by experimental results within the grid independency and time-step sensitivity studies. The 3D overall pneumatic characteristics in a stable wave cycle are presented to demonstrate the unsteady performance of the turbine rotor. Effects of air-flow velocity amplitude and rotor's moment of inertia on the flow field and pressure distribution on the blades, pneumatic torque and rotation speed, and the transient input and torque coefficients are studied.

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1. Introduction

Ocean waves contain a huge amount of unstable energy, which can be used for electricity generation and sea water desalination. Oscillating water column (OWC) is the most widely used technology for the onshore wave energy plant because of its high reliability. A land-fixed OWC plant contains a partially submerged air chamber with an underwater opening on the skirt wall. The incident waves force the water column in the chamber to oscillate in heave motion. Consequently, the oscillating water column compresses and decompresses the air volume trapped upon the free surface, and generates a reciprocating air flow in the duct connecting to the atmosphere. A self-rectifying air turbine, capable of rotating in one direction under the bi-directional flow conditions and not requiring a rectifying valve system, can be employed to convert the low pressure pneumatic power to the mechanical

energy [1].

The axial-flow Wells turbine, invented in 1970s, was the most popular self-rectifying turbine used for the OWC plants. The innovation and optimization of Wells turbine's configuration have been conducted until today. Experimental results indicate that a proper design of the 3D blade profile can improve the peak efficiency and still cannot prevent stalling [2]. The entropy generation due to viscous dissipation around NACA 0015 blade was found to be less by approximately 12% than that around NACA 0012, 0020, 0021 airfoils [3,4]. It was concluded that a higher cascade solidity and the extension of stall-free operating range have the potential for reducing the equivalent sound power and increasing the output energy [5,6]. The presence of a curved duct was reported to increase the tip loading and cause an early flow separation of the Wells turbine [7]. The penetrating blade tip treatment was found to be effective to the enhancement of the turbine performance by dealing with the tip vortex [8–10]. The cross-section profiles and variable chord-lengths of blades were claimed to increase the efficiency of turbine by 3.6% and 4.7% [11,12], respectively. A passive flow control method was proposed to delay the stall occurrence and increase the blade torque coefficient by more than 40% [13].

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Comparing to the Wells turbine, the axial-flow self-rectifying impulse turbine has a wider operating range, a better self-starting capability and no stall point [14]. Effects of some shape parameters including the tip clearance [15], rotor solidity [16] and solidity ratio [17,18] between the rotor blades and guide vanes on the turbine performance were studied numerically by using the CFD technique. Considering the asymmetric pattern of bidirectional air flows, some asymmetrical designs of the rotor blade were proposed to improve the overall power output during the exhalation and inhalation [19–21].

A novel radial self-rectifying impulse turbine was proposed, which is symmetrical with respect to a plane perpendicular to its axis of rotation. The rotor blades are surrounded by a pair of radial-flow guide vane rows [22]. The peak efficiency was attained by over 80%, and a time-averaged efficiency in a bidirectional air flows induced by random waves is higher than 70% [23,24]. A twin-rotor radial air turbine, containing a pair of conventional radial-inflow rotors mounted on the same shaft, presented a peak value of about 86% for the overall efficiency [25].

From the literature review, it could be found that no matter experimental or numerical studies only could gain the steady state performance of self-rectifying air turbines for OWC wave energy conversion. On the other hand, the air flows in the duct driven by the oscillating water column are obviously unsteady. Although the quasi-steady method could predict some unsteady performance, it cannot tell us all the transient characteristics of air turbines in the unsteady flows. With a rapid development of the computer and CFD techniques, it has become possible to conduct a calculation and attain more unsteady performance of OWC turbines. Compared to the steady calculations, the fully passive model could achieve a numerical simulation more close to the real mechanism of the impulse turbine under the reciprocating air-flow conditions.

Based on the preliminary exploring studies [26], a transient fully flow-driving numerical model is proposed in this paper to study the transient performance of OWC impulse turbine during self-starting. ANSYS-Fluent 12.0 is employed as the basic platform and the Newton's second law is realized by User-Defined Function (UDF) to compute the unsteady rotation of rotor and update the driving torque by integrating the variations of pressures on the blades. The self-starting performance of an impulse turbine under the sinusoidal-profile air-flow velocity condition is investigated by the proposed numerical model hereby, which is validated by experimental results. The 3D overall pneumatic performance in a stable wave cycle will be presented. Effects of the sinusoidal inlet-velocity amplitude and rotor's moment of inertia on the flow and pressure fields, the pneumatic performance and transient coefficients are studied.

2. Impulse turbine for OWC wave energy conversion

An axial-flow impulse turbine with fixed guide vanes is employed in this study [14,27]. The 3D schematic and shape parameters of the impulse turbine are shown in Fig. 1. As illustrated in Fig. 1 (a), the turbine is divided into three parts: the rotor and upstream & downstream stators. The rotor contains thirty blades, which can make a unidirectional rotation in the reciprocating flows. Twenty-six guide vanes are fixed symmetrically on the two stators, respectively. As presented in Fig. 1 (b), the rotor blade profile is elliptic shaped on the suction side and circular arc shaped on the pressure side. The camber line of guide vanes consists of a straight line and a circular arc.

The rotor blade pitch S_r is 26.7 mm, and the chord length l_r is 54 mm. The width of flow path t_a is 10.6 mm. The rotor blade inlet angle $\gamma = 60^\circ$, and the circular arc radius on the pressure side is

30.2 mm. The semi-major and semi-minor axis lengths of the elliptic arc on the suction side of the rotor blade E_a and E_e are 125.8 mm and 41.4 mm, respectively. The radius of blade tip circle R_t is 0.5 mm.

The chord length of guide vanes l_g is 70 mm, and the length of the straight camber line l_s is 34.8 mm. The radius of circle arc part R_a is 37.2 mm. The camber angle δ and setting angle θ are 60° and 30° , respectively. The thickness of guide vanes t_g is 2 mm. The pitch of guide vanes S_g is 30.8 mm. The spacing between rotor blades and guide vanes G is 20 mm. The duct diameter is set at 300 mm, and the hub-to-tip ratio v is 0.7. Consequently, the hub diameter is 210 mm. The tip clearance of rotor blades here is 1.0 mm. The moment of inertia of the rotor varies from 10.0 to 40.0 g·m².

Referring to the previous work [14], we defined two new parameters to represent the transient performance of the impulse turbine operating in the reciprocating air flows: Transient input coefficient C_A^* , torque coefficient C_T^* . The definitions of above coefficients are given as follows:

$$C_A^* = \frac{2\Delta p_t Q_t}{\rho_a (v_{at}^2 + U_{Rt}^2) b l_r z v_{at}} \quad (1)$$

$$C_T^* = \frac{2T_D}{\rho_a (v_{at}^2 + U_{Rt}^2) b l_r z r_R} \quad (2)$$

where, ρ_a , r_R , b , z represent the air density, the mean radius of rotor blade, the blade height and number of blades; Δp_t and T_D are defined as the absolute value of pressure drop between two sides of rotor blades and the torque generated by all rotor blades at the time t , respectively. U_{Rt} , Q_t and v_{at} represent the circumferential velocity at r_R , the absolute value of air flow rate and corresponding mean axial flow velocity over the ring section at the time t , respectively.

In this preliminary study, the loading from the generator acting as a power take-off unit will not be employed in the calculations. Hence, there is no need for discussing the energy conversion in a conservation rotating system without any power output. Because of this, we will only validate the energy conservation instead of presenting any efficiency ratios.

3. Fully passive flow-driving numerical model

3.1. Set-up of numerical model

Different from traditional calculations of artificially forced rotations with a constant angular velocity, the turbine in the present model is fully driven by the air flows and rotates passively, which is governed by Newton's second law of motion. The governing equation of the turbine rotor rotating in one degree of freedom (DOF) can be written as the follow:

$$I \cdot \frac{d\omega}{dt} + T_L = T_D \quad (3)$$

where, I is the inertia moment of turbine rotor, ω is the rotational angular velocity. T_D is the resultant driving torque acting on the rotor. T_L is the loading torque, which is related to the reaction torque of the linked electricity generator, system frictions & resistances. In these unsteady calculations, the rotation of impulse turbine always starts from rest to be accelerated. The variation of rotating velocity is determined by the updated angular acceleration derived from the resultant torque. The angular velocity at the time t can be modified from Eqn. (3):

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