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Aero-economical optimization of Wells turbine rotor geometry

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ABSTRACT

Wave plants are currently extracting energy from sea waves at higher levelized cost of electricity (LCOE) compared to other renewable energy resources. The present research aims at reducing the LCOE of wave plants that use the axial flow Wells turbines as a power take-off system by optimizing the turbine rotor geometry. A novel rotor geometry was proposed, numerically investigated and optimized. This geometry was obtained by varying and optimizing the radial solidity distribution of the traditional Wells turbine rotors. Up to 15% saving of the Wells turbine LCOE was achieved by optimizing the rotor geometry. This cost saving is mainly due to the increase of the turbine output power where the change of the blade manufacturing cost is negligible. The present work highlights the significance of the plenum chamber-turbine coupling for every turbine design. This is because the numerical results showed an increase of the damping coefficient for the turbine with the optimized rotor geometry. Therefore, it was necessary to reduce the plenum chamber volume in order to maintain an optimum turbine-chamber coupling. This increases the cost saving to 20.6% at the turbine design point and reduces the plant construction time.

1. Introduction

Wave energy is getting more attention as a feasible renewable energy resource due to its availability in many countries. The axial flow Wells turbine is one of the most practical air turbines that can be used with Oscillating Water Columns OWC [1]. Fig. 1 shows a schematic representation of a Wells turbine with its plenum chamber and ducting. The free water surface oscillates inside the plenum chamber as a result of the incident waves. The plenum chamber converts the wave power into pneumatic power that drives the turbine. Both the turbine and the plenum chamber are affecting each other. Therefore, perfect turbine-chamber coupling is crucial for every wave power plant as discussed later. The oscillating wave motion inside the plenum chamber causes a bidirectional air flow through the turbine rotor. The symmetrical airfoil profile of the Wells turbine rotor blades maintains the direction of the tangential force F_u during the bidirectional air flow, Fig. 2. Consequently, Wells turbines rotate in the same direction despite the bidirectional air flow.

Authors e.g., [1–4] have extensively reviewed the performance of Wells turbines under different geometric and operating conditions. They also introduced and discussed the advantages and disadvantages of Wells turbines. The axial flow Wells turbines have the advantages of the simplicity of design and manufacturing as well as the compact size. However, they suffer low aerodynamic efficiency, narrow operating range, poor self-starting characteristics, high axial force coefficient and low tangential force coefficient. Despite these disadvantages, many commercial and pilot wave plants are now producing power from ocean waves (e.g., the Islay LIMPET and the Mutriku Breakwater Wave Plant). The cost of producing electricity from different energy resources is expressed in terms of the levelized cost of electricity (LCOE). The LCOE of a given technology is defined as the ratio of lifetime costs to lifetime electricity generation [5]. The current estimates for levelized cost of electricity (LCOE) of wave energy technologies in 10 MW demonstration projects is in the range of 330–630 EUR/MW h [6] compared to about 70–360 EUR/MW h for solar photovoltaic, 80–180 EUR/MW h for offshore wind energy and 26–145 EUR/MW h for onshore wind energy [5]. A detailed analysis of the LCOE for different renewable energy technologies can be found in [5,6]. The LCOE for wave power plants is considerably higher than that of the other renewable energy converters because wave energy harvesting technologies are still in the early stages of development. Therefore, the present work considers the optimization of Wells turbines from both the aerodynamic and the economical points of view.

Many attempts have been made to overcome the known Wells turbines' disadvantages. Curran and Gato [7] and Kim et al. [8] experimentally investigated the performance of Wells turbines with different designs. Setoguchi et al. [9,10] studied the perfor-

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Nomenclature

A_A	area at the annular turbine duct (m^2)
A_C	area at the water column surface (m^2)
B^*	turbine damping coefficient (N s/m)
C_T	torque coefficient (–)
c	chord length (m)
D	drag force (N)
Δp_o	total pressure drop (Pa)
Δp_o^*	pressure drop coefficient (–)
F_u	tangential force (N)
F_x	axial force (N)
L	lift force (N)
LCOE	levelized cost of electricity (EUR/MW h)
LE	leading edge
LT	expected life time (h)
MC	manufacturing cost (EUR)
N	number of blades (–)
Q	volume flow rate (m^3/s)
R	rotor tip radius (m)
R^*	normalized radius (–)
r	radius (m)
T	torque (N m)
t	time
TE	trailing edge
U	tangential velocity at rotor tip (m/s)
$\vec{v}$	absolute velocity (m/s)
v_a	axial velocity (m/s)
$\vec{v}_r$	velocity viewed from a moving frame (m/s)
$\vec{v}_t$	transitional velocity (m/s)
W	relative velocity (m/s)
$\dot{W}_T$	turbine output power (MW)

Greek

α	angle of attack (°)
ϕ	flow coefficient (–)
η	efficiency (–)
ρ	air density (kg/m^3)
σ	solidity (–)
σ^*	normalized solidity (–)
$\bar{\tau}$	viscous stress (N/m^2)
ω	angular velocity (rad/s)

Subscript

Hub	turbine hub
m	mid-span
optimum	performance with optimized rotor geometry
original	performance with original radial solidity distribution
r	radial
Tip	turbine tip
σ	performance with modified radial solidity distribution

mance of mono-plane and biplane Wells turbines with fixed inlet guide vanes as well as with self-pitch-controlled guide vanes. Kinoue et al. [11] and Thakker and Abdulhadi [12] experimentally investigated the performance of Wells turbine with different blade profiles. Kim et al. [13] conducted numerical simulations of the Wells turbine performance with different values of the blade sweep. Kim et al. [14] studied the effect of hub-to-tip and aspect ratios on the performance of Wells turbine. Authors have also investigated the effect of inlet guide vanes [15,16], blade setting angle [17], end plates [18] and tip clearance [19,20]. Shaaban and Abdel Hafiz [21] numerically investigated and optimized the effect of duct geometry on Wells turbine performance. Mohamed and Shaaban [22,23] studied and optimized the effect of blade pitch angle of a self-rectifying Wells turbine. Halder and Samad [24,25] numerically investigated the effect of casing treatment on Wells turbine performance and established a relationship for the optimum turbine speed under different wave conditions. They reported that this relation can be used to design turbines with higher performance.

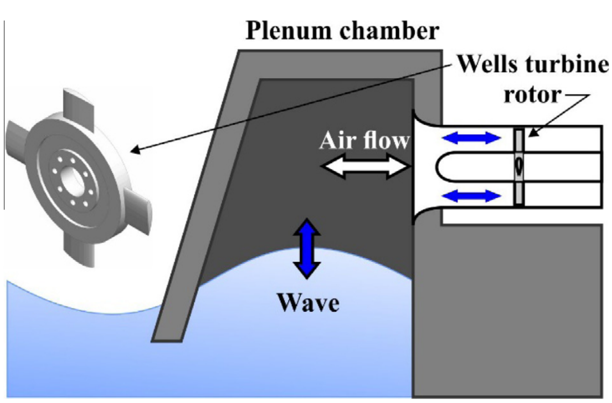

Fig. 1. Schematic cutaway of a Wells turbine.

Among the different design parameters, turbine solidity significantly influences the performance of Wells turbines. The turbine solidity σ_m is defined as

$$\sigma_m = \frac{Nc}{2\pi r_m} \tag{1}$$

where N the number of blades, c the chord length and r_m is the mid-span radius (geometric mean radius). Due to its significance, many authors have investigated the effect of solidity on Wells turbine performance. Thakker and Abdulhadi [12] investigated the Wells turbine performance at solidities $\sigma_m = 0.48$ and 0.64. They showed that the preferable rotor geometry is the one with solidity $\sigma_m = 0.64$. Gato et al. [26] and Torresi et al. [27] experimentally and numerically studied the performance of high solidity Wells turbines while Torresi et al. [28] analyzed the performance of a low solidity turbine. Raghunathan and Tan [29] considered both the starting and running performance of a Wells turbine and showed that a solidity of $\sigma_m = 0.6$ is the most favorable one. Raghunathan [1] reported that at small values of solidity the effect of solidity is small but significant reduction in efficiency occurs for $\sigma_m > 0.5$. He also showed that increasing the turbine solidity increases the

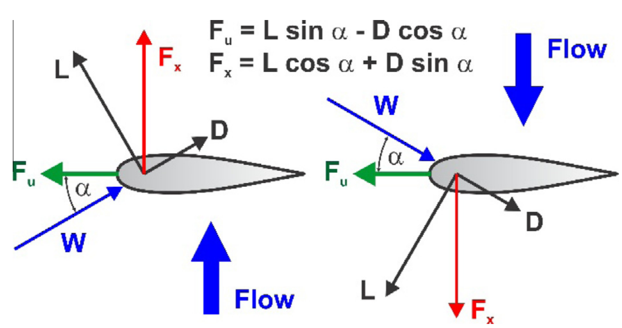

Fig. 2. Force analysis for a Wells turbine blade.