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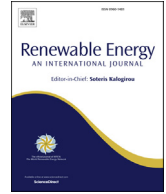
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Application of integrated wave-to-wire modelling for the preliminary design of oscillating water column systems for installations in moderate wave climates

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ABSTRACT

Oscillating water column (OWC) devices are the technical solution more favourably considered for a significant future exploitation of the energy potential of sea waves. While theoretically quite simple, these systems are indeed rather complex in terms of correct tuning between the caisson and turbine. Recent studies showed that only a holistic approach, also including turbine control, is the way of effectively performing their precise design.

To this end, the research presents the application of an analytical wave-to-wire model of an OWC device for the preliminary optimisation of these systems for a small installation in a moderate wave climate in the Mediterranean Sea. More specifically, scatter matrices were determined for imposing the wave states of a selected location positioned in Tuscany (Italy). Regarding previous similar studies, a novel joint optimisation of the caisson and controlled turbine is proposed for the specific site. Either monoplane isolated Wells turbines and axial impulse turbines were analysed. The operating curves, performance parameters, and attended annual energy harvesting were calculated, providing an interesting overview of the functioning of the system. The optimised devices with the Wells and impulse turbomachines convert 36.71 and 42.17 MWh/year operating with a global efficiency of 6.82% and 7.84%, respectively.

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1. Introduction

The sea is commonly recognised as having enormous energy potential, which has not been appropriately exploited. This energy comes in different forms, including waves, currents, tides, and thermal gradients. Among others, waves probably represent the most promising resource as they provide the highest combination of energy density and availability. As a general reference, the global power conveyed by waves to the coastline is expected to be between 0.5 and 2.2 TW [1,2].

During the last decades, the attention toward sea energy exploitation has been increasing, and several prototypes and pilot plants of wave energy converters (WECs) have been proposed [3,4]. Especially in the European Atlantic Arc, the combination of sea

energy potential, energy policy, market conditions, and new patents have fostered the development of these devices [5]. This area is currently considered one of the most promising in the short to medium term [6]. In particular, most WEC systems were deployed in the United Kingdom and Portugal. In recent years, important progress has been achieved even in America, Asia, and Oceania. Moreover, some developing countries with a favourable wave climate are promoting research on these devices to support their significant energy requirements [7,8]. This may be achieved by relying on the knowledge gained by European countries, the definition of clear and specific targets and a priority position of this technology in the energy industry, the realisation of efficient and stable regulations, and the establishment of capital grants and facilities [5].

Oscillating water column devices (OWCs) are considered the most credited solution for wave energy conversion. Compared to other technologies proposed over the years for wave energy harvesting as oscillating body devices (OBDS) and overtopping devices

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(OTDs), they have important advantages [9–11]. Indeed, OWCs combine a decent energy conversion efficiency, a simple structure, an easy installation, and a good reliability thanks to the fact that all mechanical components are installed above the still water level (SWL) for not being directly exposed to the harsh marine environment.

An OWC system is able to convert the kinetic energy of incoming waves into electrical energy. This transformation is obtained as a result of three main different steps. First, waves motion is converted by a chamber (primary converter) into the kinetic energy of an airflow, oscillating within a fixed duct. This is obtained as the incident waves determine a heave oscillating movement of the seawater column inside the chamber, which alternatively compresses and expands the air volume contained above the free water level. Then, this kinetic energy is transformed into a mechanical one by means of an air turbine (secondary converter). Finally, an electrical generator (tertiary converter) further converts this mechanical energy into electrical.

Despite their potential, OWC systems are still mainly in a demonstration phase, with some prototypes also tested at the real scale in the sea [12–16]. However, they have not turned into a consolidated industrial technology yet, primarily due to a levelised cost of energy (LCOE) that is very high. Indeed, it ranges between 90 and 101 €/MWh for onshore systems, and it is in the interval from 180 to 490 €/MWh for offshore systems [17,18]. The possibility of integrating wave energy converters with complementary solutions for renewable energy harvesting as wind and solar devices and the use of energy storage systems may contribute to reducing the cost of energy [19]. In this view, additional research is required to foster their installation worldwide. Moreover, the realisation of apposite national policies is beneficial to support the industrial development of OWC systems.

Optimising these systems is unfortunately a complex process. In particular, there is a mutual influence between the aero-servo-elastic behaviour of the turbine and the fluid-dynamic behaviour of the oscillating air delivered by the chamber. Both the effects determine the global efficiency of energy conversion. While most of the studies so far focused on the individual optimisation of one of the two components [20–24], the present research aims at providing a comprehensive analysis of the entire system for a distinctive wave climate. Indeed, recent findings showed that integrating dedicated simulation models for the caisson and the controlled turbine into a common framework is the only way to converge effectively towards optimal design [25–27]. Moreover, the system performance varies significantly with different wave climates, characterised by precise wave periods and heights, and during the seasons and from one year to another [27–29]. For example, Lopez et al. proposed a numerical and physical methodology to assess the energy extraction of an OWC device for a particular wave energy site. The method was initially studied for a single year [28] and then improved by taking into account possible fluctuations in energy availability between various years [29]. The case study revealed that the intra-annual variability in energy harvesting is remarkable, up to more than 20%.

Some of the authors recently provided a detailed overview of integrated OWC simulation systems [30]. To these references is added the work of Scialò et al. that in 2021 investigated the design of a power take-off for a possible installation in a breakwater in Calabria, in Italy. The operation of a monoplane isolated Wells turbomachine and a biradial impulse turbomachine coupled with an OWC system of U-type and equipped with a relief air valve was analytically investigated through a wave-to-wire code. The turbine diameter and power output were optimised for both the machines, and the effects of the interconnection of air chambers sharing a single turbine were assessed [31].

Overall, the analysis of state of the art shows that only limited contributions have been provided up to now to the evolution of wave-to-wire codes. They mainly apply simple methods for modelling turbomachine aerodynamics, reducing the accuracy of evaluating the airflow conditions imposed by the coupling with the caisson. These models often do not allow the designer for a specific optimisation of both the caisson and turbomachine for a distinctive application site and, thus, for the attended working conditions. In addition, some recent applications of experimental testing of the OWC system at a laboratory scale were proposed, including a small-scale prototype of the air turbine [32]. However, even this method seems not the best solution due to the significant scaling effects and the relevant costs and time of investigation [14,33]. All these aspects indicate that using analytical wave-to-wire modelling approaches directly solving the turbine aerodynamics is paramount.

Within this framework, the early versions of an improved analytical model for the design of devices for wave power conversion relying on the principle of the oscillating water column were developed [30,34–36]. The chamber, air turbine, and electric generator are integrated into a wave-to-wire perspective. Also, the code features aerodynamic models for both a Wells turbine with a single rotor cascade and an impulse turbine of axial type with one rotor cascade and two stator cascades. In the present study, the tool was further developed to perform the joint optimisation of the primary and secondary converters. The model was applied to a demonstrative installation in Tuscany (Italy) for evaluating the system potential in a moderate wave climate in the Mediterranean Sea. The novelty of the research concerning the literature is the site-specific optimisation of either caisson and turbines, which provides an unprecedented insight into the real potential of an OWC application in the Mediterranean environment. The main strength of the analyses is the joint solution of the three converters to determine power transformation from sea waves to the electric wire through a fast and reliable approach. Thus, the proposed model is especially recommended in the early design stages to determine the application site and the main geometric and operating parameters of the device maximising energy harvesting.

The present article proceeds from the research presented at the 16th Conference on Sustainable Development of Energy, Water and Environment Systems (SDEWES).

2. Methodology

The wave-to-wire model proposed in the present research integrates three analytical codes of the leading converters of an OWC system. The model structure is depicted in the flow chart in Fig. 1. The wave-to-wire analysis starts with the determination of the sea wave states defining the wave spectrum of the application site. Afterwards, an iterative procedure is applied to jointly solve the hydrodynamics, aerodynamics, and thermodynamics of the chamber with the aerodynamics and dynamics of the turbine and the dynamics of the electrical generator. The turbine operating curves are utilised for iteratively matching the airflow conditions provided by the chamber and the damping imposed by the turbomachine. The control strategy regulates the coupling of the turbine and electric generator regarding the torque and rotational velocity.

2.1. Primary converter modelling

The caisson code relies on the potential flow theory, which applies the hypotheses of inviscid, incompressible, and irrotational flow. In particular, the rigid piston schematisation is implemented to determine the water column movement within the structure. This approach represents the main widespread practice of the

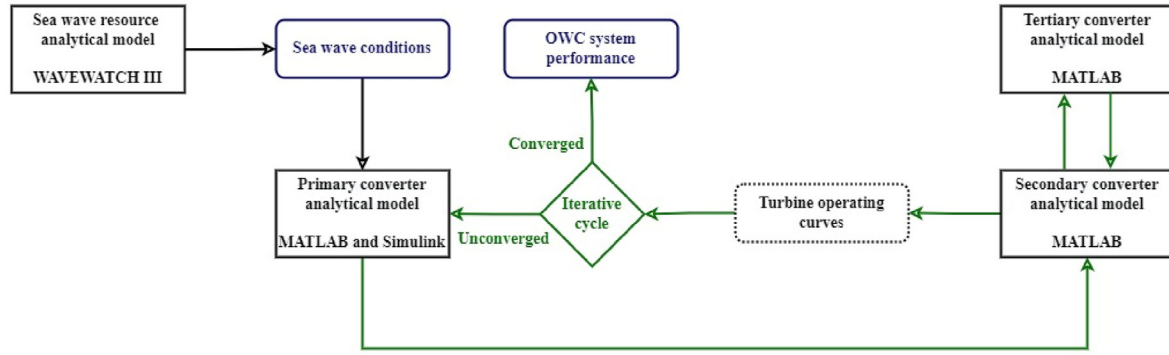


Fig. 1. Flow chart of the structure of the wave-to-wire model.

literature due to the good compromise between accuracy and computational requirements [37–40]. The schematisation is appropriate for the current study cases as the chamber size is small compared with the wavelength of the incoming wave.

The chamber typology taken into account is fixed and detached regarding the seabed. The marine depth of the reference site is hypothesised to be fixed, and the presence of unidirectional waves is imposed. In these circumstances, the only possible movement of the water column is the heave oscillation, which is schematised as a rigid piston. Therefore, its equation of motion is obtained with the application of Newton's second law in the vertical direction, which assigns through Eq. (1) the balance of the forces acting on the rigid piston.

$$m_w \ddot{z} + B \dot{z} + Cz = f_{exc} + f_{add} + f_{pto} \quad (1)$$

where m_w is the water column mass, z is the free surface position regarding the still water level, positive when above it, B is the radiation damping coefficient, C is the hydrostatic restoring coefficient, f_{exc} is the excitation force exerted on the bottom of the water column by the hydrodynamic pressure of the incident waves, f_{add} is the force due to the added mass, and f_{pto} is the force determined by the air pressure of the power take-off (PTO) in the top of the caisson.

The seawater mass oscillating within the structure is provided by Eq. (2) as the product of the seawater density ρ_w , the horizontal section of the structure A_c , and the sum of the length of the seaward wall draught D and the free surface level.

$$m_w = \rho_w A_c (D + z) \quad (2)$$

The hydrostatic restoring coefficient of Eq. (3) depends upon the seawater density, the gravitational acceleration g , and the chamber radius R determining its horizontal section.

$$C = \rho_w g \pi R^2 \quad (3)$$

The radiation damping coefficient is defined relying on the hydrostatic restoring coefficient and the water total mass, given by the sum of the water column mass and the added mass [39]. The added mass m_{add} of the cylindrical-shaped and axisymmetric chamber was analytically determined through Eq. (4). It is considered the same as a hemisphere with radius equivalent to that of the chamber [39]. The radiation damping coefficient is obtained through Eq. (5) combining the restoring coefficient and the inertia of the seawater [37,40].

$$m_{add} = \frac{2}{3} \rho_w \pi R^3 \quad (4)$$

$$B = 0.2 \sqrt{C(m_w + m_{add})} \quad (5)$$

As stated by the linear wave theory, the vertical component of the water surface position η_w of an incident sea wave with height H is defined through Eq. (6) as a cosinusoidal relation.

$$\eta_w = \frac{H}{2} \cos(kx - \omega t) \quad (6)$$

where k is the wavenumber, x is the horizontal component of the wave surface position, ω is the angular frequency, and t is the time. The wavenumber is determined by the dispersion formulation of Eq. (7) for the peculiar seawater profundity h and wave period. The result was obtained by applying the Newton-Raphson scheme.

$$\omega^2 = kg \tanh(kh) \quad (7)$$

The dynamic pressure of the water inside the chamber p_w varies over time according to Eq. (8) based on the instant values of the water depth, rigid piston position, and wave displacement.

$$p_w = \eta_w \rho_w g \frac{\cosh(k(h+z))}{\cosh(kh)} \quad (8)$$

Since the caisson dimensions are significantly low compared with the wavelength, the diffraction of sea waves may be neglected. Therefore, the exciting force acting in correspondence with the front wall draught D is provided by Eq. (9).

$$f_{exc} = \eta_w A_c \rho_w g \frac{\cosh(k(h-D))}{\cosh(kh)} \quad (9)$$

The added mass of the seawater oscillating inside the chamber determines the force f_{add} of Eq. (10) due to the relative vertical acceleration $\dot{v}_{z,w} - \ddot{z}$ of the water particles compared to the rigid piston.

$$f_{add} = m_{add} (\dot{v}_{z,w} - \ddot{z}) \quad (10)$$

The power take-off force f_{pto} is determined by Eq. (11) as a function of the chamber horizontal section and the relative air pressure p_c of the chamber regarding the atmosphere.

$$f_{pto} = -A_c p_c \quad (11)$$

The previous relations are included in Eq. (1) to reformulate Newton's second law of the rigid piston as in Eq. (12). This formula is applied to solve the hydrodynamics of the chamber.

$$m_w \ddot{z} + B \dot{z} + A_c \rho_w g z = \eta_w A_c \rho_w g \frac{\cosh(k(h-D))}{\cosh(kh)} + m_{add}(\dot{v}_{z,w} - \ddot{\eta}_w) - A_c p_c \quad (12)$$

The determination of the airflow conditions at the interface between the caisson and the turbine requires even the solution of the thermodynamics and the aerodynamics of the primary converter. The hypotheses of adiabatic and isentropic transformations are applied for the alternate air compression and expansion. Thus, Eq. (13) provides the mass airflow rate passing through the system based on the pressure variation inside the caisson [14]. For a significantly low value of A_{cz} relative to V_0 , the contribution of the former term is negligible. Thus, the mass flow rate is finally expressed as a linear relation [30,34–36,41].

$$\dot{m} = A_c \rho_a \dot{z} - V \frac{1}{s^2} \frac{dp_c}{dt} = \rho_a \dot{q} - (V_0 + A_{cz}) \frac{1}{s^2} \frac{dp_c}{dt} = \rho_a \dot{q} - V_0 \frac{1}{s^2} \frac{dp_c}{dt} \quad (13)$$

where ρ_a is the air density, s is the sound speed in the air, $\dot{q}$ is the volume airflow rate, and V and V_0 are the instant and unperturbed values of the air volume, respectively.

It is worth pointing out that a recent assessment of real gas effects on the performance of an OWC system proved that the vapour mixture of water and air undergoes a not entirely adiabatic process [42]. In particular, high values of moisture and high levels of pressure drop in the chamber may partially affect the estimated power output of the system. However, considering these effects is out of the scope of the present work.

2.2. Secondary converter modelling

The Wells and impulse turbine aerodynamics and dynamics were solved with analytical approaches. The models were developed to achieve a good compromise between accuracy and computational requirements for effective integration within the entire code. The blade element momentum (BEM) theory was implemented, assuming each blade of a turbine row as a set of airfoils stacked in the radial direction. Every couple of adjacent airfoils defines a single blade element.

2.2.1. Wells turbine

The aerodynamics of the Wells turbine was solved by applying an actuator disc model. This approach considers a cascade of a turbomachine as an infinitely thin disc simulating the effect of the former in terms of energy exchange with the flow. These models impose the conservation of mass and momentum to determine the energy extraction by flows that are not subject to high vorticity and energy losses [43,44]. The procedure has been widely used for many types of rotors, including wind turbines, and in the present study it was adapted to Wells turbines. In this latter case, a partial limitation to the model accuracy could be represented by the typically high solidity of these rotors, for which the validity of the actuator disc model approaches its limits. However, the turbine configurations for the study were selected to avoid this issue.

A monoplane isolate Wells turbine was considered for the operation in the wave energy converters investigated. A single rotor row constitutes this turbomachine with symmetrical blades staggered around the central hub (Fig. 2). The blades are installed symmetrically relative to the plane orthogonal to the rotation axis and, therefore, with an angle of 90° regarding the incoming flow [45,46].

The power P_i exchanged between the air and the rotor blades is provided by Eq. (14).

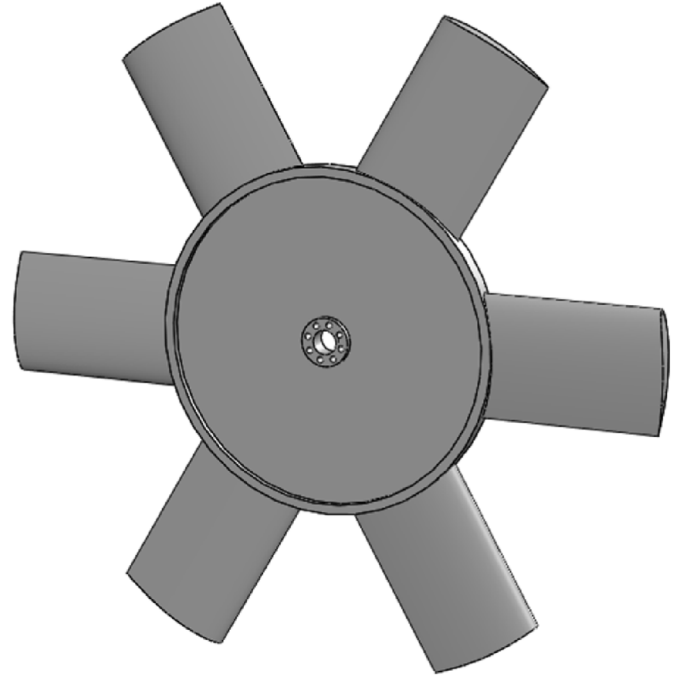


Fig. 2. Rendering of the monoplane isolated Wells turbine optimised for functioning in Tuscany.

$$P_i = \frac{1}{2} Z \rho_a l_i C_{\theta CIF,i} w_{1,i}^2 \Omega r_i dr \quad (14)$$

where Z is the blade number, l is the blade chord length, $C_{\theta CIF}$ is the circumferential force coefficient modified through the cascade interference factor (CIF) to consider the effects of mutual influence amongst rotor blades, w is the relative velocity of the air, Ω is the rotor angular speed, r is the radius, and subscripts 1 and i indicate, respectively, the inlet section of the rotor and the generic blade element.

The difference in static pressure between the sections upstream and downstream of the rotor is given by Eq. (15). The formula imposes the equilibrium of the pressure force agent on the airflow and the force exerted on the rotor blades in the axial direction [30,34–36,41].

$$p_{1,i} - p_{2,i} = \frac{\rho_a l_i Z C_{x,i} v_{x1,i}^2}{4\pi r_i \sin^2 \beta_{1,i}} \quad (15)$$

where p is the static air pressure, C_x is the axial force coefficient, v_x is the absolute axial velocity of the air, β is the relative flow angle, and subscript 2 refers to the outlet section of the rotor.

The actuator disc model applies an iterative procedure based on energy conservation for calculating the flow field across the turbine cascade [30,34–36,41]. The model was validated through the comparison with computational fluid dynamics investigations and literature experimental and analytical studies [47].

2.2.2. Impulse turbine

A typical axial impulse turbomachine for WECs is depicted in Fig. 3. It usually comprises one rotor row and two stator rows, symmetrically arranged with respect to their plane of symmetry and orthogonal to the rotation axis. Depending on the flow direction, the first static cascade acts as an inlet guide vane (IGV), accelerating the flow. The outlet guide vane (OGV) is a diffusive

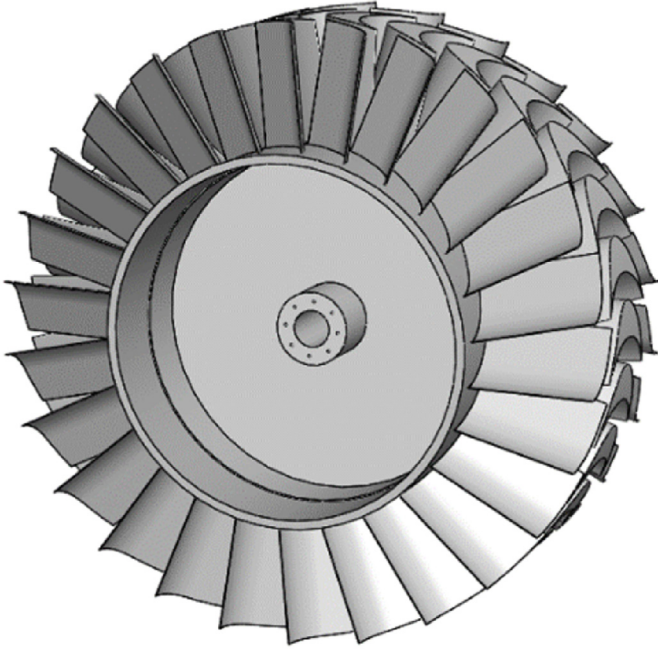


Fig. 3. Rendering of the stators and rotor of the axial impulse turbine optimised for functioning in Tuscany.

cascade that straightens and decelerates the airflow after the passage through the rotor. The stators regulate the incidence flow angle on impact with the rotor to minimise aerodynamic losses and reduce the risk of stall onset provoked by boundary layer separation.

The aerodynamics of impulse turbines is less prone to simplified modelling concerning Wells turbines. This study proposes an original analytical model relying on the blade momentum theory [30,34–36,41]. The flow is modelled through experimental correlations derived from axial turbomachines [44].

The analytical model applies the criterion of Zweifel, the criterion of Carter and Hughes, and the empirical rule formulated by Howell to determine the geometry of the rotor and stator cascades and the velocity triangles of the airflow [44].

The power P_i exchanged between the airflow and the rotor blades is provided by Eq. (16) through the application of Euler's theorem based on the variation of the velocity triangles of the airflow crossing the rotor [30,34–36,41].

$$P_i = \rho_{a,i} A_{T,i} v_{x,i} U_i (v_{\theta 1,i} + v_{\theta 2,i}) \quad (16)$$

where A_T is the area of the turbomachine annulus, U is the blade speed, v_θ is the tangential component of the absolute flow velocity, and subscripts 1 and 2 indicate the sections at the rotor entrance and exit, respectively.

The correlation of Ainley and Mathieson of Eq. (17) is imposed to assess the aerodynamic losses of the upstream stator and the rotor [44]. The global losses Y_{AM} are provided by combining the profile losses Y_{pr} , the secondary losses Y_{se} , the tip leakage losses Y_{tc} , and the trailing edge losses Y_{te} of the cascade, and the correction factor cf reliant on the Reynolds number [30,34–36,41].

$$Y_{AM} = [(Y_{pr} + Y_{se})cf + Y_{tc}]Y_{te} \quad (17)$$

The aerodynamic losses of the downstream stator are evaluated through the correlation of Howell [44]. The losses are calculated with Eq. (18) relying on the equivalent global drag coefficient $C_{D,HW}$

given by the sum of the profile, skin friction, and secondary drag coefficients $C_{D,pr}$, $C_{D,sf}$, and $C_{D,se}$, respectively [30,34–36,41].

$$C_{D,HW} = C_{D,pr} + C_{D,sf} + C_{D,se} \quad (18)$$

While quite simple and making use of methods adapted from compressors with design specifications partially different from those of the blades of an impulse turbine, the proposed approach showed to nicely fit the few experimental data available for validation [30]. This can be obtained at a very reduced computational cost compared to other techniques often proposed for these applications as computational fluid dynamics.

2.3. Tertiary converter modelling

The air turbines for oscillating water column systems are subject to changes in working conditions during their functioning. This is due to the sinusoidal behaviour of the airflow for a specific wave status and the variability of sea states. Therefore, effective energy conversion requires implementing a regulation method that controls the connection of the air turbine and the electricity generator. The algorithm balances the instant values of rotor torque and monitors the angular speed, keeping the system in controlled rotation in either design and off-design operation.

The hypothesis of negligible aerodynamic friction losses caused by windage is applied. Thus the function governing the angular acceleration $\dot{\Omega}$ of the turbine rotor depends on its rotational inertia J and torque τ_T and the torque of the electrical generator τ_g as expressed in Eq. (19).

$$\dot{\Omega} = \frac{\tau_T - \tau_g}{J} \quad (19)$$

The mechanical transmission of power between the turbine and electric generator depends upon the torque acting on the former. When its value is negative, the two components are disconnected, determining the decrease of angular velocity on the basis of the rotational inertia. Instead, for positive values, the control algorithm compares the actual level of the rotor angular velocity with its design level Ω_0 , as indicated by Eq. (20).

$$\begin{cases} \Omega(t) < \Omega_0 \rightarrow \tau_g = 0 \\ \Omega(t) \geq \Omega_0 \rightarrow \tau_g = \tau_T \end{cases} \quad (20)$$

When the instant rotor speed is lower than the design value, the mechanical torque is not transmitted by the turbine to the electric generator as they are disconnected. This implies that the turbomachine is free to accelerate for reaching its design speed. When the rotor velocity is equivalent or superior to the design value, the two components are promptly connected to transform the mechanical power into electric power [30,34–36,41].

The electric power is defined through Eq. (21) considering the organic efficiency ξ_o and the electrical efficiency ξ_e , and the power provided by the secondary converter, expressed by Eq. (14) or Eq. (16) for the Wells and impulse turbines, respectively. Both the efficiency values are assumed equal to 0.97 based on the characteristic intensity of the losses involved.

$$P_e = \xi_o \xi_e P_T \quad (21)$$

3. Results and discussion

The wave-to-wire code was utilised for the preliminary optimisation of both the chamber and turbomachine of an oscillating water column system. The application site was selected to be

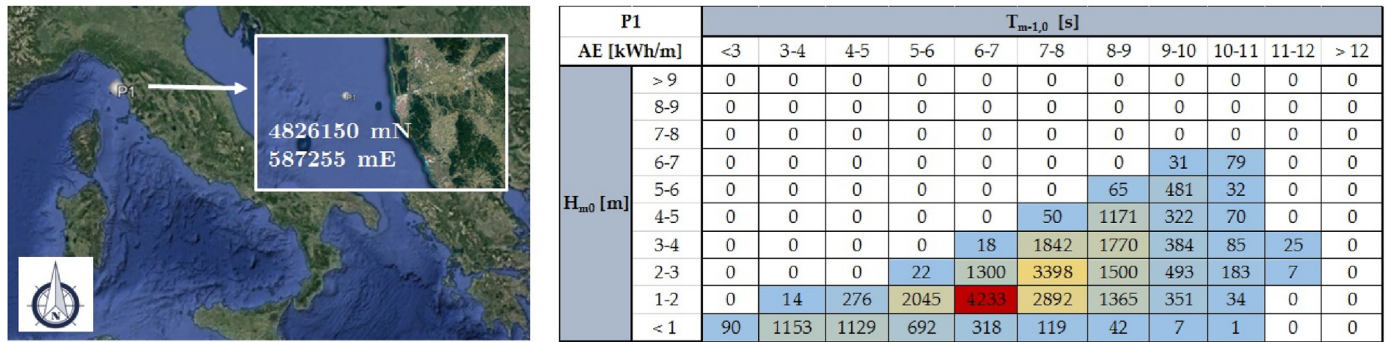


Fig. 4. (left) Geographic map and coordinates of the selected site located in Tuscany and (right) scatter matrix of its annual specific wave energy [35].

representative of the moderate wave conditions of the central Mediterranean area. Parametric analyses were conducted to designate the most suited geometric features of the fixed and detached chamber and the geometry and operating conditions of the Wells and impulse turbines by maximising the overall energy harvesting of the entire system. A control algorithm was implemented to manage the rotor speed and the mechanical coupling of the turbomachine and electricity generator.

3.1. Application site definition

The reference site is positioned in front of the coast of Tuscany (Italy) and has a latitude of 4826150 m N and a longitude of 587255 m E as shown in Fig. 4. It is situated on a seawater depth of 50 m and has a mean annual power associated with the wave motion of about 3.3 kW/m [30,35,48]. It is worth noticing that most of energy transport is determined by waves with significant height between 1 m and 3 m and energy period between 5 s and 9 s. Therefore, the site is representative of moderate wave conditions.

The energy potential of the sea waves was determined from the hindcasting database for the period from 2000 to 2019 calculated using the spectral wave model WAVEWATCH III® [49,50]. The sea states of the location are defined by the scatter matrix of the bivariate distributions of significant wave height H_{m0} and energy period $T_{m-1,0}$ indicating each wave condition. The corresponding specific energy AE conveyed by a wave during one year is calculated based on Eq. (22), approximating the deep-water power P_w for a meter of wavefront length.

Table 1

Scatter matrix of specific wave energy AE during the year for the location selected in Tuscany [30,35,36].

AE, kWh/m		$T_{m-1,0}$, s										
		2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5
H_{m0} , m	4.5	0	0	0	0	0	50	1171	322	70	0	0
	3.5	0	0	0	0	18	1842	1770	384	85	25	0
	2.5	0	0	0	22	1300	3398	1500	493	183	7	0
	1.5	0	14	276	2045	4233	2892	1365	351	34	0	0
	0.5	90	1153	1129	692	318	119	42	7	1	0	0

Table 2

Scatter matrix of the relative frequency rf of the wave conditions during the year for the location selected in Tuscany [30,35,36].

rf , %		$T_{m-1,0}$, s										
		2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5
H_{m0} , m	4.5	0	0	0	0	0	0.01	0.16	0.04	0.01	0	0
	3.5	0	0	0	0	0.01	0.48	0.41	0.08	0.02	0	0
	2.5	0	0	0	0.02	0.76	1.73	0.67	0.20	0.07	0	0
	1.5	0	0.04	0.65	3.94	6.90	4.09	1.70	0.39	0.03	0	0
	0.5	3.44	31.44	23.94	12.00	4.67	1.51	0.47	0.07	0.01	0	0

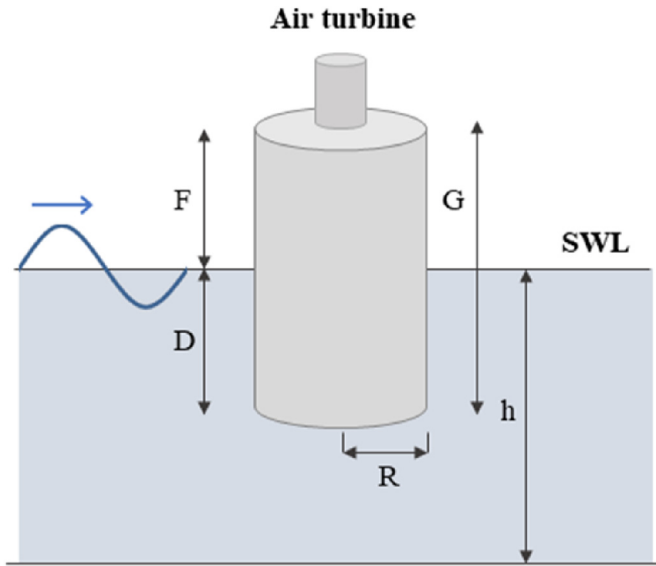


Fig. 5. Scheme of the structure of the primary converter analysed with the optimisation procedure.

Table 3

Test plan of the geometric parameters of the caisson assigned in the optimisation procedures.

Parameter	OWC chamber		
	Minimum	Maximum	Interval
Radius, m	5.00	10.00	2.50
Draught, m	3.00	9.00	1.50
Height, m	9.00	15.00	1.50

air intake from the submerged lip of the caisson during the passage of wave troughs, while the highest value was set to 9 m for constraining the dimensions and the decrease of the exciting force. A fixed length of the freeboard wall equal to 6 m was imposed as it represents a possible reference value for preventing seawater from reaching the turbine in the foreseen wave conditions. Therefore, the design space of Table 3 was applied in the optimisation analyses.

3.3. Turbine selection

The optimisation of the monoplane isolated Wells turbine and the axial impulse turbine was performed for the wave conditions of the scatter matrices of Tables 1 and 2 considering the geometric parameters of Table 3 for the caisson. The parametric investigation was conducted by varying the turbine geometry, as reported in Table 4. The ranges were designated relying on the main prototypes and experimental devices of OWC systems tested in the last decades [14,25,51,52]. The tip clearance gap was kept to 1% of the rotor blade height. The rotor speed in design conditions was set to 900 rpm and 300 rpm for the Wells and impulse turbines, respectively, based on previous optimisation analyses [30,34–36].

Afterwards, the rotational speed analysis was refined for the most promising turbine configurations to increase the annual energy conversion further. Intervals of 400 rpm centred around the values formerly determined were considered and variations of 25 rpm were imposed.

3.4. System operating maps

The operating maps of the configurations of the OWC systems functioning with the Wells or impulse turbines were determined

for evaluating the effects on energy conversion of the most important design parameters of either caisson and turbine. For each turbine typology, the annual energy harvesting ratio $\Sigma_{d,y}$ is provided by Eq. (23) as the ratio of the energy converted by the system design considered and the optimal system configuration.

$$\Sigma_{d,y} = \frac{E_{d,y}}{E_{max,y}} \quad (23)$$

where E is the energy harnessed, subscript d indicates the specific arrangement evaluated, subscript max refers to the optimal configuration maximising energy conversion, and subscript y is the year.

In Figs. 6 and 7, the energy harvesting ratio of the systems operating with the Wells and impulse turbines is presented by comparing four chamber configurations with a radius of 7.5 m or 10 m and a chamber draught of 3 m or 4.5 m. These layouts were chosen to assess turbine performance using the same caisson parameters that provide high levels of primary energy conversion. For every chamber, the energy harvesting ratio is shown as a function of the hub-to-tip ratio and casing radius of the turbomachine, which rotates at 900 rpm and 300 rpm for the Wells and impulse turbine, respectively.

For the Wells turbine, the optimal chamber configuration has a radius of 7.5 m and a draught of 3.0 m. Varying radius decreases energy harnessing as it influences the hydrodynamic coefficients and the seawater inertia. Reducing draught increases performance due to the enhancement of the exciting force exerted by the incident wave corresponding to the front wall draught. Regarding turbomachine, enlarging turbine diameter allows for greater energy extraction due to increased flow rate. In particular, a turbine diameter of 2 m maximises the performance of the analyses conducted. The hub-to-tip ratio significantly influences $\Sigma_{d,y}$ and reducing this geometrical parameter to 0.5 significantly increases energy harvesting, as it determines the rise of flow rate. Moreover, low values of the hub-to-tip ratio require lower levels of rotor solidity to ensure the ability of the turbine to run-up independently to a certain rotational speed. In turn, limiting the solidity reduces the blockage of the flow across the rotor cascade and, thus, the requirement of primary energy availability. For the most-suited configurations, the rotor tip solidity is between 0.30 and 0.45.

The comparison of the caisson configurations in Fig. 7 indicates that the optimal geometry for the impulse turbine has a radius of 10 m and a draught of 3.0 m. In this case, energy conversion is similar for the values of the caisson radius of 7.5 m and 10 m as it slightly increases through the rise of this dimension. Conversely, energy harvesting reduces drastically for a radius of 5 m. Regarding the chamber draught, the rise of this parameter determines the reduction of system performance in analogy with the other turbine typology. Concerning turbine specifications, casing diameter has relevant effects, and the increase of this parameter enhances system outcomes. Especially, a value of 2 m maximises resource exploitation for the test plan. The hub-to-tip ratio significantly influences system operation and a level of 0.6 characterises the most-suited configurations.

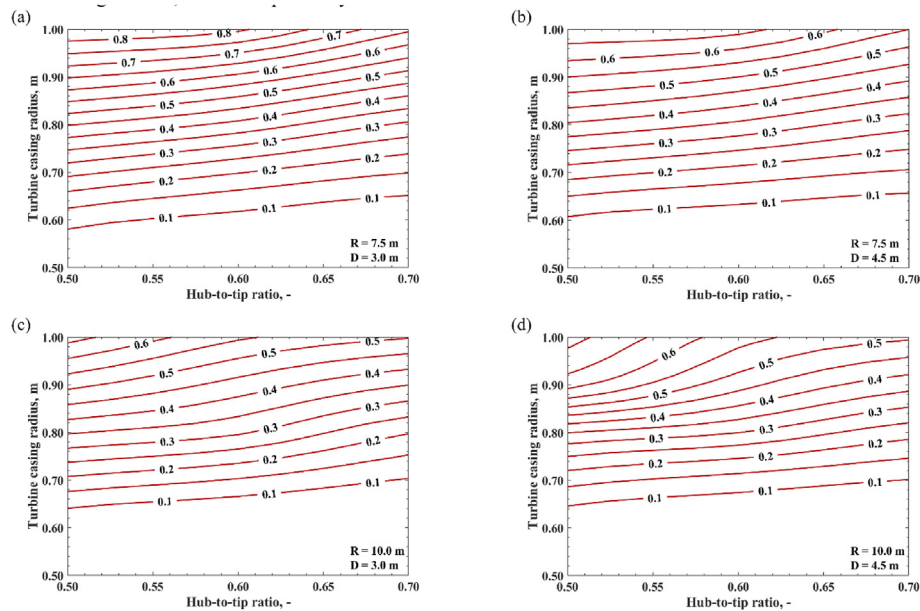
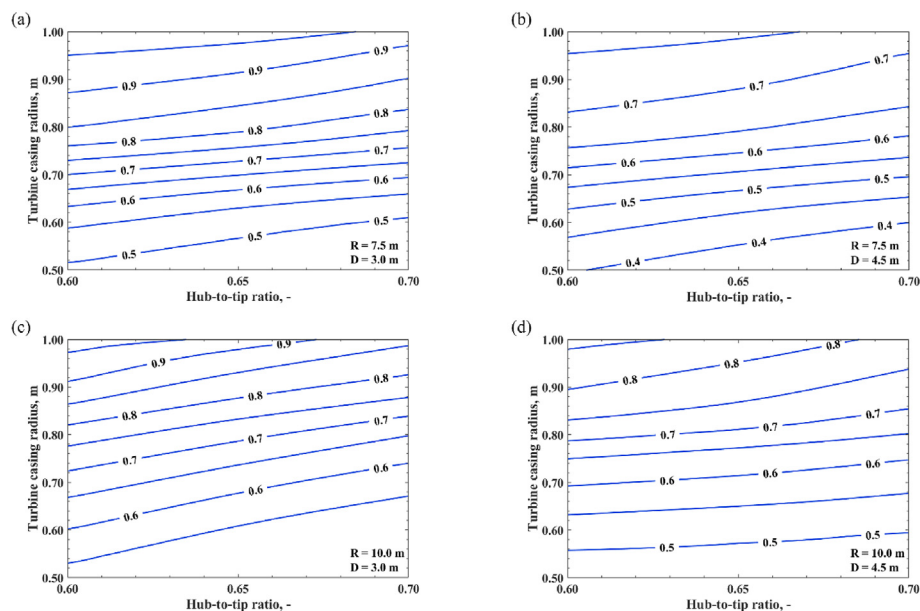
The blade velocity needs to be limited to values corresponding to a relative Mach number of 0.47 at the tip [51]. Indeed, it is necessary to avoid the losses caused by transonic conditions and shocks, which could be present around the blade tip at high velocities. In addition to the requirement of operation below the critical speed, the condition of autonomous run-up was applied for the Wells turbines. Therefore, only the turbine configurations compliant with these requisites were considered to select the optimal system layouts.

The main features of the primary and secondary converters maximising the annual energy conversion of the system in the intervals of the test plan are listed in Table 5.

Table 4

Test plan of the geometric specifications of the turbines applied in the optimisation procedures.

Parameter	Wells turbine			Impulse turbine		
	Minimum	Maximum	Interval	Minimum	Maximum	Interval
Turbine casing radius, m	0.50	1.00	0.10	0.50	1.00	0.10
Hub-to-tip ratio, -	0.50	0.70	0.10	0.60	0.70	0.05
Aspect ratio, -	0.50	1.50	0.10	—	—	—
Stator blade number, -	—	—	—	26	30	2
Rotor blade number, -	5	14	1	30	36	2

**Fig. 6.** Energy ratio of the Wells turbine to changes of turbine hub-to-tip ratio and radius for different values of chamber radius and draught.**Fig. 7.** Energy ratio of the impulse turbine to changes of turbine hub-to-tip ratio and radius for different values of chamber radius and draught.

The turbine casing dimension is equivalent for the two turbomachines. The rotational speed of the chosen Wells turbine is 4.56 times greater than the selected impulse turbine; thus the blade

speed ratio has the same value. This results in markedly higher aerodynamic acoustic emissions for the Wells turbine. The rotational inertia was determined considering the turbine rotors

Table 5
Geometric and operating parameters of the optimal system configurations.

Parameter	Wells system	Impulse system
Chamber radius, m	7.5	10.0
Chamber draught, m	3.0	3.0
Chamber height, m	9.0	9.0
Rotor angular velocity, rpm	1025	225
Turbine casing radius, m	1.0	1.0
Hub-to-tip ratio, -	0.5	0.6
Rotor blade aspect ratio, -	1.4	1.0
Stator blade number, -	—	26
Rotor blade number, -	6	30
Tip clearance, %	1	1
Rotational inertia, m ² kg	23.2	176.9

constituted by marine grade aluminium EN AW 5083 to foster their duration in a corrosive environment while limiting the manufacturing expenses [31].

The effects of chamber radius and draught on annual energy harvesting are indicated in Table 6 considering the selected values of the geometry and rotational speed of the two selected turbines.

For either turbine typologies, reducing caisson draught from 9.0 m to 3.0 m increases energy extraction progressively. In terms of chamber radius, the Wells turbine is less influenced by this parameter than the impulse turbine.

3.5. Annual system operation

The functioning of the selected air turbines is presented through the non-dimensional and dimensional quantities typically utilised in turbomachinery [44]. The turbine flow coefficient φ of Eq. (24) expresses the ratio between the absolute axial velocity of the air and the blade speed at the midspan section m of the blades.

$$\varphi_m = \frac{v_{x,m}}{U_m} \quad (24)$$

The input coefficient C_A and the torque coefficient C_T are expressed by Eq. (25) and Eq. (26), respectively. They indicate the primary power converted by the chamber and the torque exerted on the rotor blades, respectively, relative to the operating and geometric parameters of the turbine.

$$C_{A,m} = \frac{(p_{f,in} - p_{out})\dot{q}}{\frac{1}{2}\rho_a(v_{x,m}^2 + U_m^2)blZv_{x,m}} \quad (25)$$

$$C_{T,m} = \frac{\tau}{\frac{1}{2}\rho_a(v_{x,m}^2 + U_m^2)blZr_m} \quad (26)$$

where p_{tin} and p_{out} are the total pressure at the turbomachine entrance and the static pressure at its exit, respectively, $\dot{q}$ is the volume airflow rate, b is the blade height, and r_{tip} and r_{hub} are the tip radius and hub radius, respectively.

Table 6
Effects of caisson radius and draught on annual energy extraction for the two optimal turbines.

Energy, kWh/y		Wells turbine			Impulse turbine		
		Chamber radius, m					
		5.0	7.5	10.0	5.0	7.5	10.0
Chamber draught, m	3.0	32.09	35.92	28.86	22.43	41.35	41.95
	4.5	26.11	28.50	28.35	18.54	34.05	36.28
	6.0	20.78	24.42	26.84	16.08	27.81	30.59
	7.5	16.03	22.23	24.85	13.01	23.88	28.34
	9.0	12.53	20.45	24.46	10.47	20.00	27.88

The turbine efficiency ξ_T is defined in Eq. (27) as the ratio of the mechanical power exchanged between the airflow and rotor blades and the pneumatic power delivered by the chamber.

$$\xi_T = \frac{Q\tau}{(p_{f,in} - p_{out})\dot{q}} = \frac{C_{T,m}}{\varphi_m C_{A,m}} \quad (27)$$

The dimensional variables are the difference in the static pressure of the flow between the inlet and outlet of the turbine and the torque and power acting on the rotor blades.

The results of the two optimal device arrangements are presented for the wave status providing the highest contribution to the overall energy harnessing during one year. This state is defined by a significant height of 2.5 m and an energy period of 7.5 s of the sea wave. Thereby the two systems are evaluated and compared in design conditions.

The time variations of the position and acceleration of the incident sea wave relative to the STL are depicted in Fig. 8(a) for two periods. The height of the sea wave is determined from the significant wave height through the division by the square root of two, while the wave period is equivalent to the energy period. The vertical position of the seawater free surface inside the chamber is shown in Fig. 8(b) for both the selected configurations. The average and maximum values are higher for the case of the Wells turbine.

The operation of OWC systems is composed of the inflow phase, during which the air enters the device from the ambient in atmospheric conditions, and the outflow phase, when the air compressed inside the chamber is forced to flow with the opposite sense of motion. The higher air density implies that the performance parameters typically reach higher values for equal flow coefficient or flow rate in the latter phase. These parameters are presented by varying the midspan flow coefficient and the volume flow rate, conventionally considered negative for the inflow and positive for the outflow.

Fig. 9(a) shows that the input coefficient has a higher average value and reaches higher peaks for the impulse turbine, while it has faster growth for the Wells turbine. Thus, the primary energy transformed by the chamber to parity of turbine functioning is greater for the former arrangement. The graph denotes that the operating range of the impulse turbine is significantly wider than the Wells turbine. All the performance parameters confirm this feature.

The torque coefficient of Fig. 9(b) is significantly higher for the impulse turbomachine. This is due to the greater torque exerted on the rotor blades compared to the functioning and geometric parameters of the machine. The torque coefficient is negative for low levels of flow coefficient due to the limited flow incidence angle when the forces of the viscous effects prevail on the forces determined by the pressure. When these opposing forces are balanced, the parameter is null. It becomes positive if the contribution of the pressure forces predominates over that of the viscous forces, as occurs by increasing the flow incidence angle.

Fig. 10(a) presents the relation between the volume airflow rate and the variation in static pressure across the turbomachine. For the Wells arrangement, the function is linear except for the extremes of the flow phases. For the impulse arrangement, the relation is quadratic. The pressure oscillation required to the caisson for the functioning of the latter machine is lower to parity of airflow conditions.

Fig. 10(b) shows the average torque exchanged between the airflow and the rotor blades. The torque is markedly greater for the impulse configuration, while the rate of rise is higher for the Wells configuration.

From a perusal of Fig. 10(c) it follows that the rotor power converted by the two turbines is higher for the Wells turbine under the

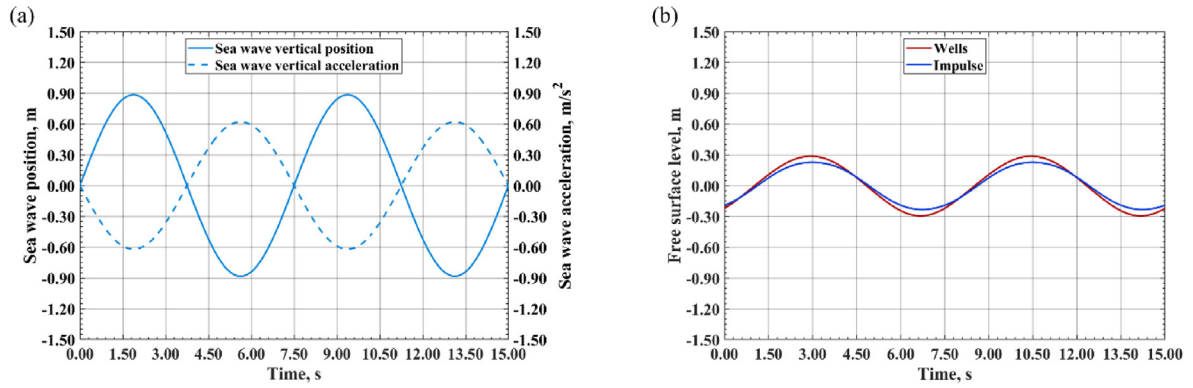


Fig. 8. (a) Time series of the incident wave position and acceleration and (b) time series of the vertical position of the seawater free surface within the chamber for the two selected devices.

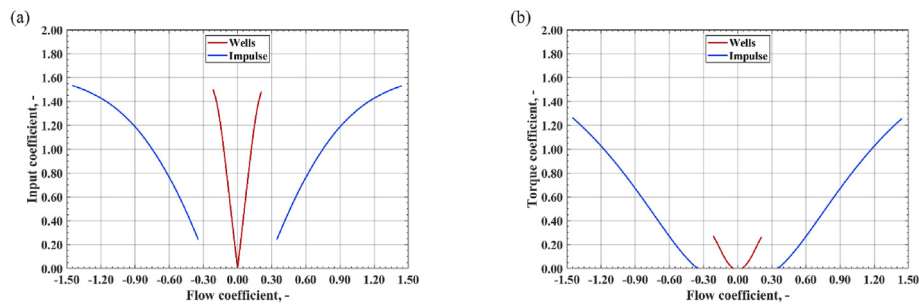


Fig. 9. (a) Input coefficient and (b) torque coefficient to changes of the midspan flow coefficient under the wave conditions maximising energy harnessing.

wave conditions providing the highest contribution to global energy harnessing. Conversely, the impulse turbine has a greater average power extraction considering all the wave conditions of the site.

The efficiency curves of Fig. 10(d) show that the average efficiency of the Wells turbine is greater compared to the impulse turbine.

Regarding the global results, the annual energy converted by the two optimised OWC systems is 35.92 MWh for the Wells turbine and 41.95 MWh for the impulse turbine. Implementing a regulation algorithm for controlling the air turbine and the electricity generator intensifies the energy conversion of 17.88% for the former case and 0.32% for the latter. The effect is significantly greater for the Wells turbomachine as its operation is subject to higher variations of power and efficiency in off-design conditions.

The computed efficiency of power conversion from waves to wire is 7.39% for the Wells configuration and 8.64% for the impulse configuration.

The levels of efficiency calculated for the main stages of power transformation of the two OWC systems are reported in Table 7.

The efficiency of the WECs is reduced to 6.67% for the Wells case and 7.80% for the impulse case referring to the entire energy conveyed by waves.

The annual energy ratio is the non-dimensional parameter defined in Eq. (28) to assess the contribution to the energy harvesting of each wave status during one year. For every turbine typology, it is given by the ratio between the energy converted by a distinctive wave class and the energy converted by the system for all the classes that characterise the wave climate.

$$\Sigma_{ij,y} = \frac{E_{ij,y}}{E_y} \quad (28)$$

where subscripts i and j are the row and column indices of the scatter matrices referring to a single wave condition.

Fig. 11 shows that the functioning of the two systems has one region with high peaks of energy harvesting and another region with considerable levels of this parameter. The first region belongs to the highly energetic wave classes with a significant wave height ranging from 1 to 3.5 m for the Wells device and from 1 to 4.5 m for the impulse device, and with an energy period between 7 and 9 s. The relevant amounts of energy converted are mainly due to the great power of these wave classes. The second region is identified by waves with low significant wave height and energy period. For these wave conditions, the relevant energy transformation is determined by their high frequency of occurrence. This region is markedly narrower for the Wells system compared to the impulse system. For the former case, the extension of the region is defined by values with H_{m0} between 0.5 and 1.5 m and with $T_{m-1,0}$ from 3 to 4 s. For the latter case the ranges are between 0.5 and 2 m and from 3 to 6 s, respectively. The system performance reduces significantly in the other wave states.

By comparing the selected WECs, the Wells configuration reaches greater peaks of energy harnessing, while the impulse one has a smoother functioning. The global energy converted is higher for the impulse turbine owing to the improved ability to exploit a wide variety of sea states, especially those with great wave height and period.

3.6. Seasonal system functioning

The analysis was extended taking into account the average seasonal distributions of energy transport and frequency of the wave states reported in the Appendix.

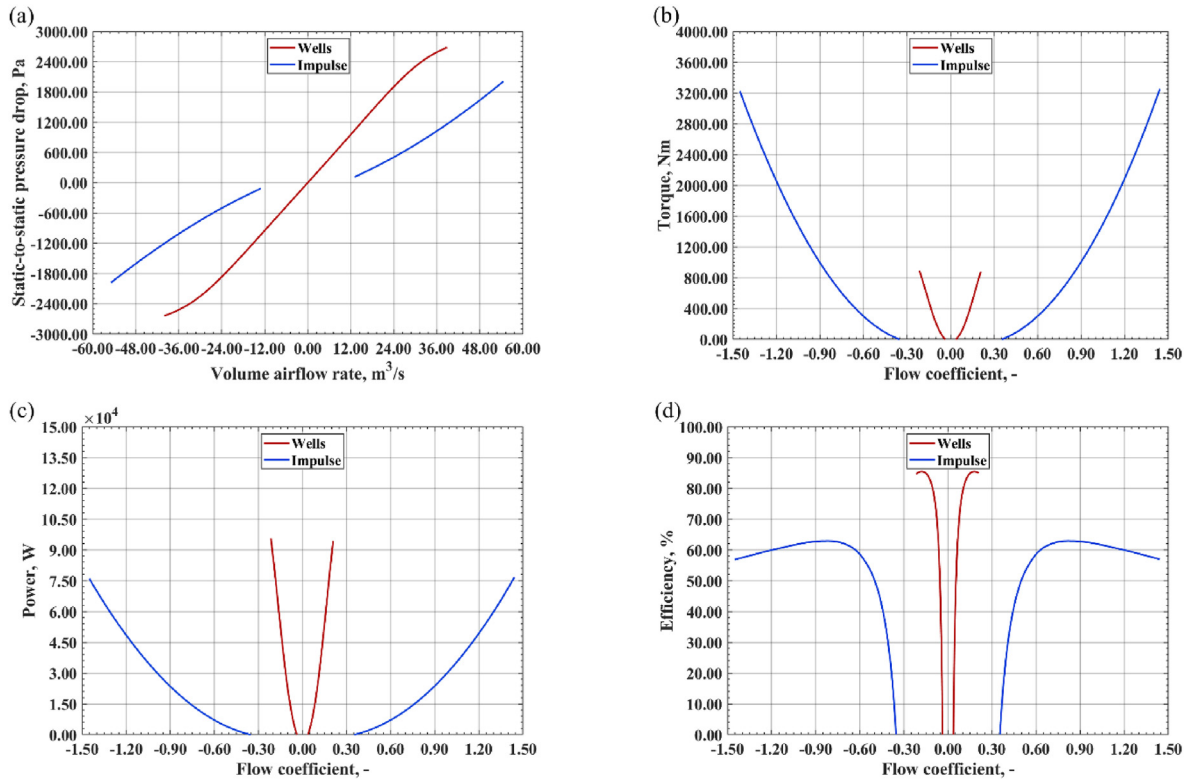


Fig. 10. (a) Static-to-static pressure drop as a function of the volume flow rate, and (b) turbine torque, (c) power, and (d) efficiency to changes of the midspan flow coefficient under the wave conditions maximising energy harnessing.

Table 7

Principal levels of efficiency computed for the converters of the optimised WECs.

Efficiency	Wells system	Impulse system
Caisson, %	13.03	16.33
Turbine with control, %	60.31	56.23
Organic, %	97.00	97.00
Electric, %	97.00	97.00

In analogy with the evaluation of the annual operation, the seasonal energy ratio $\Sigma_{ij,s}$ of Eq. (29) is determined to present the different contributions of every season to the global energy harvesting. This quantity is provided by the quotient of the energy converted during a specific season s for each wave condition and the corresponding value calculated during the entire year.

$$\Sigma_{ij,s} = \frac{E_{ij,s}}{E_{ij,y}} \quad (29)$$

The perusal of Figs. 12 and 13 indicates that each season has a distinctive contribution to the annual energy harvested. For either system configuration, the highest energy exploitation occurs during winter when the wave climate is characterised by the very high frequency of occurrence of waves with great values of height and period. Conversely, resource exploitation is minimum in summer for the low frequency of these sea states. The energy harnessing is intermediate and comparable during spring and autumn, with a higher contribution of the wave conditions with a low height and a considerable period for the former season.

The comparison of the two system configurations shows that using the selected impulse turbine extends the width of effective operation. This arrangement provides higher values of energy harvesting during all seasons apart from summer. The magnitude of the difference is around 20% for winter and autumn and about 12% for the other seasons.

The optimal design angular velocity was determined even on a seasonal basis to increase the overall energy conversion during the

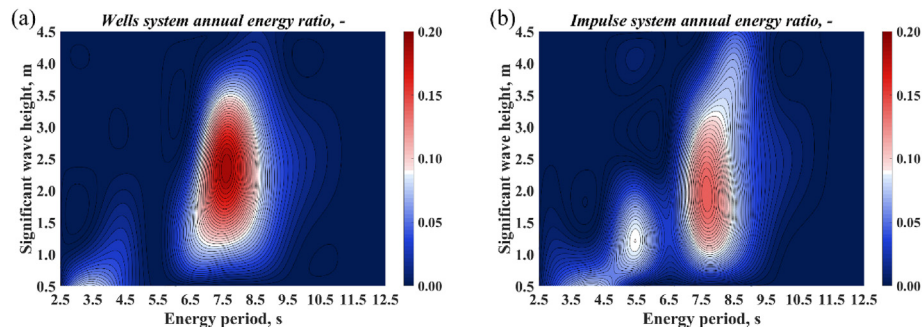


Fig. 11. Distributions of the annual energy ratio of the entire set of wave classes investigated for the WECs functioning with the (a) Wells and (b) impulse systems.

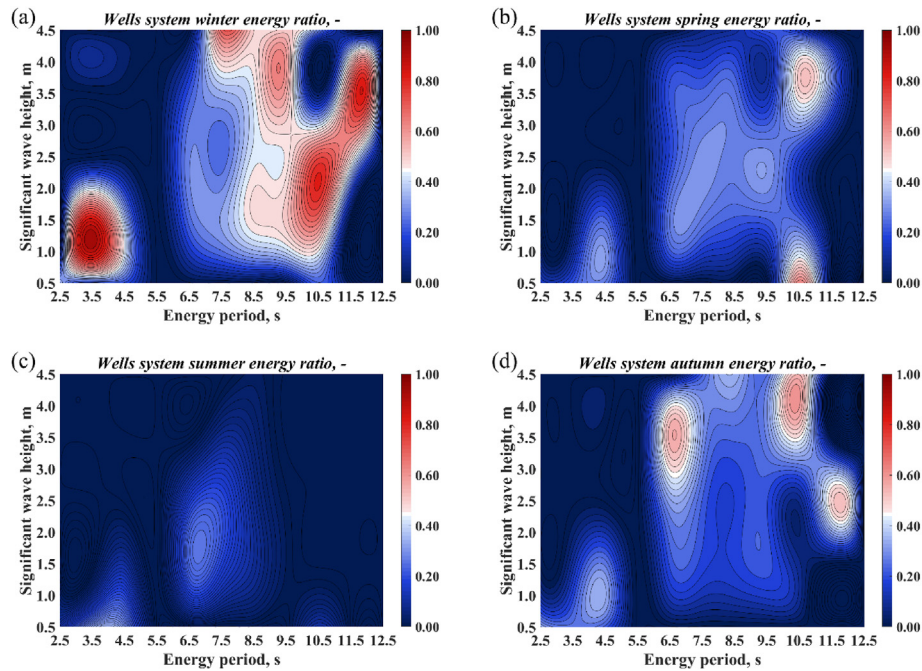


Fig. 12. Distributions of energy ratio during (a) winter, (b) spring, (c) summer, and (d) autumn for the optimal system with the Wells configuration.

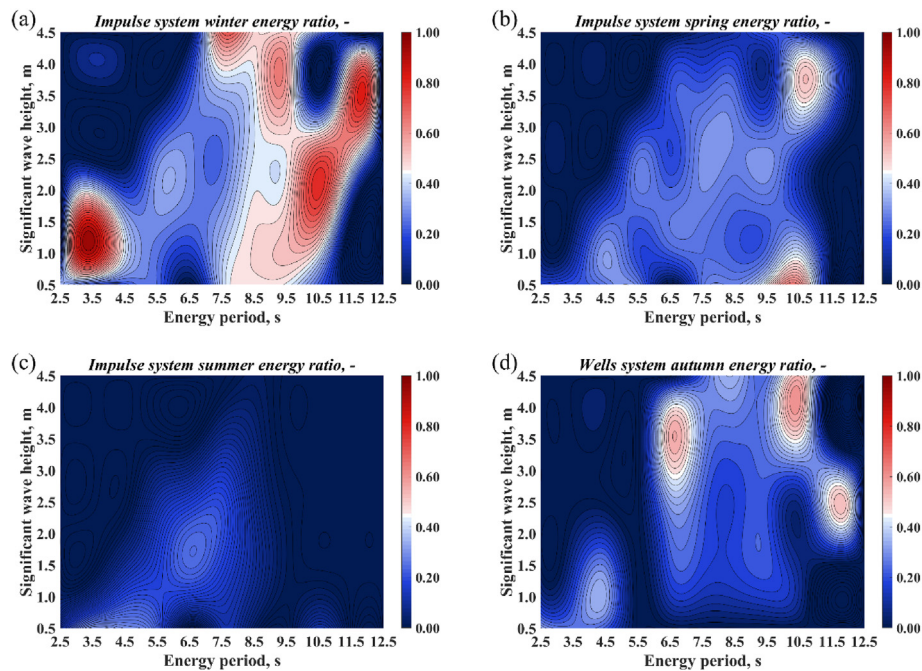


Fig. 13. Distributions of energy ratio during (a) winter, (b) spring, (c) summer, and (d) autumn for the optimal system with the impulse configuration.

Table 8

Design angular speed velocity of the turbine rotor optimised for each season of the year.

Season	Wells system	Impulse system
Winter	1100	250
Spring	1000	225
Summer	900	200
Autumn	1075	225

year. The values maximising power transformation for either turbine configuration are listed in Table 8.

The definition of the seasonal design values of angular velocity increases global energy harvesting from 35.92 to 36.71 MWh/year considering the wave energy converter with the optimal Wells configuration and from 41.95 to 42.17 MWh/year for the optimal impulse configuration. The related efficiency determined based on the scatter matrices is 7.56% and 8.68%, respectively, and it is

reduced to 6.82% and 7.84%, respectively, based on the entire wave climate of the application site.

The results achieved indicate that the determination of the best-suited configuration of an OWC system depends on the wave conditions of the application site and the geometry of the chamber and the geometry and rotational speed of the turbine regulated by the control strategy. Therefore, a joint optimisation of the whole system for a specific location is required.

4. Conclusions

A comprehensive wave-to-wire model for evaluating energy conversion from the sea resource to the grid was developed for WECs based upon the OWC principle. The research aimed to create a fast and reliable tool for the system design during the early project stages when a wide variety of system configurations is analysed.

The model implements an integrated solution of the chamber (primary converter), air turbine (secondary converter), and electric generator (tertiary converter). The rigid piston modelling was utilised for studying the hydrodynamics of the water column and the thermodynamics and aerodynamics of the airflow delivered by the chamber, which is fixed and detached relative to the seabed. An analytical model based on the actuator disc and the blade element momentum theory was generated to determine the aerodynamics of monoplane isolated Wells turbines. An original method relying on the blade element momentum theory and experimental correlations widely used to design axial impulse turbines was applied for the other machine typology. A monitoring algorithm was utilised to increase power conversion by regulating rotor speed and torque. The integration of the models of the three converters into a framework in which physical quantities are propagated from sea waves to electricity is the main point of strength of the presented analyses.

A wide variety of OWC systems was investigated to evaluate their future deployment in a selected location in Italy. The site is positioned in the central area of Tuscany. The sea depth is 50 m and the WGS84-UTM32 coordinates are 4826150 m N for the latitude and 587255 m E for the longitude. Joint optimisation procedures were performed to define the most suited geometry of the chamber and the best geometry and rotor angular velocity for either turbomachine by maximising energy harvesting.

The optimal system configurations with the Wells and impulse turbines provide an annual energy conversion of 35.92 MWh and 41.95 MWh, respectively. The regulation of the turbine speed and generator torque determines the increase of 17.88% and 0.32% of the energy harnessing, respectively. An additional enhancement of energy extraction is fostered by defining the specific design values of rotor angular velocity for the seasons. This allows raising annual energy harvesting to 36.71 MWh for the Wells configuration and 42.17 MWh for the impulse configuration. The efficiency of power transformation is 7.56% and 8.68% for the two selected cases,

respectively. For the entire power of the chosen site, the values decrease to 6.82% and 7.84%, respectively.

The operating maps of the two machines show that impulse turbines have a better capacity for maximising annual energy harvesting than Wells turbines for the selected site. Furthermore, the turbines of the former typology are optimised at a lower rotational speed. This reduces the mechanical stresses acting on their components, increasing their duration and decreasing their costs. In addition, impulse turbines have a wider operating range, require lower pressure drop levels to parity of volume flow rate, and have smoother functioning. This implies that the selection of an impulse turbine is recommended for the location examined.

Possible future development of the present research could aim to evaluate the performance of the two OWC systems in terms of levelised cost of energy, although unfortunately, reliable data on cost for these systems are scarce, especially for turbines.

Author contributions

Conceptualisation, G.M., Lorenzo Ciappi, A.B., and Lorenzo Ciappi; methodology, G.M., Lorenzo Ciappi, A.B., and Lorenzo Ciappi; software calculations (entire model), Lorenzo Ciappi; software calculations (resource and primary converter), I.S.; validation, Lorenzo Ciappi, I.S., and A.B.; formal analysis, investigation, and resources, all authors; data curation, Lorenzo Ciappi and I.S.; writing (original draft preparation), Lorenzo Ciappi; writing (review and editing), Lorenzo Ciappi, A.B., and I.S.; visualisation, Lorenzo Ciappi; supervision, G.M., Lorenzo Ciappi, and A.B.; project administration, G.M. and Lorenzo Ciappi.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Appendix

The distributions of specific wave energy AE and frequency of occurrence rf are indicated in the Tables from A.1 to A.8 for each season

Table A.1
Scattering of specific wave energy AE during winter for the location selected in Tuscany [35,36].

AE, kWh/m		$T_{m-1,0}$, s										
		2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5
H_{m0} , m	4.5	0	0	0	0	0	35	590	166	22	0	0
	3.5	0	0	0	0	4	639	766	219	3	18	0
	2.5	0	0	0	6	380	833	643	223	133	4	0
	1.5	0	11	102	608	1071	958	650	182	23	0	0
	0.5	33	218	229	170	100	46	20	3	0	0	0

Table A.2Scattering of relative frequency rf of the wave conditions during winter for the location selected in Tuscany [35,36].

$rf, \%$		$T_{m-1,0}, s$										
		2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5
H_{m0}, m	4.5	0	0	0	0	0	0.02	0.33	0.08	0.01	0	0
	3.5	0	0	0	0	0	0.68	0.72	0.18	0	0.01	0
	2.5	0	0	0	0.02	0.91	1.73	1.18	0.37	0.20	0	0
	1.5	0	0.14	0.98	4.79	7.14	5.53	3.31	0.83	0.09	0	0
	0.5	5.08	24.27	19.82	12.03	5.96	2.41	0.91	0.13	0.01	0	0

Table A.3Scattering of specific wave energy AE during spring for the location selected in Tuscany [35,36].

$AE, kWh/m$		$T_{m-1,0}, s$										
		2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5
H_{m0}, m	4.5	0	0	0	0	0	4	131	73	11	0	0
	3.5	0	0	0	0	4	457	476	53	42	7	0
	2.5	0	0	0	6	259	1083	449	158	33	0	0
	1.5	0	1	68	474	1209	815	319	83	8	0	0
	0.5	16	252	315	201	82	32	13	3	1	0	0

Table A.4Scattering of relative frequency rf of the wave conditions during spring for the location selected in Tuscany [35,36].

$rf, \%$		$T_{m-1,0}, s$										
		2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5
H_{m0}, m	4.5	0	0	0	0	0	0	0.07	0.04	0	0	0
	3.5	0	0	0	0	0	0.47	0.43	0.04	0.03	0	0
	2.5	0	0	0	0.02	0.60	2.17	0.79	0.25	0.05	0	0
	1.5	0	0.01	0.63	3.60	7.76	4.53	1.57	0.36	0.03	0	0
	0.5	2.40	27.06	26.27	13.70	4.73	1.61	0.56	0.14	0.03	0	0

Table A.5Scattering of specific wave energy AE during summer for the location selected in Tuscany [35,36].

$AE, kWh/m$		$T_{m-1,0}, s$										
		2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5
H_{m0}, m	4.5	0	0	0	0	0	0	31	0	0	0	0
	3.5	0	0	0	0	0	166	100	0	0	0	0
	2.5	0	0	0	3	230	691	131	8	0	0	0
	1.5	0	0	30	369	1037	548	137	8	0	0	0
	0.5	11	371	334	161	53	14	2	0	0	0	0

Table A.6Scattering of relative frequency rf of the wave conditions during summer for the location selected in Tuscany [35,36].

$rf, \%$		$T_{m-1,0}, s$										
		2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5
H_{m0}, m	4.5	0	0	0	0	0	0	0.02	0	0	0	0
	3.5	0	0	0	0	0	0.17	0.09	0	0	0	0
	2.5	0	0	0	0.01	0.53	1.38	0.23	0.01	0	0	0
	1.5	0	0	0.28	2.80	6.66	3.05	0.67	0.03	0	0	0
	0.5	1.58	39.82	27.84	10.9	3.08	0.68	0.07	0	0	0	0

Table A.7Scattering of specific wave energy AE during autumn for the location selected in Tuscany [35,36].

$AE, kWh/m$		$T_{m-1,0}, s$										
		2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5
H_{m0}, m	4.5	0	0	0	0	0	12	419	83	38	0	0
	3.5	0	0	0	0	10	580	428	112	39	0	0
	2.5	0	0	0	7	431	792	278	104	17	4	0
	1.5	0	2	76	594	915	572	260	78	3	0	0
	0.5	31	312	252	160	83	26	8	1	0	0	0

Table A.8Scattering of relative frequency rf of the wave conditions during autumn for the location selected in Tuscany [35,36].

$rf, \%$		$T_{m-1,0}, s$										
		2.5	3.5	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5
H_{m0}, m	4.5	0	0	0	0	0	0.01	0.24	0.04	0.02	0	0
	3.5	0	0	0	0	0.01	0.61	0.40	0.09	0.03	0	0
	2.5	0	0	0	0.02	1.02	1.63	0.50	0.17	0.02	0	0
	1.5	0	0.02	0.73	4.63	6.04	3.27	1.31	0.35	0.01	0	0
	0.5	4.78	34.42	21.62	11.25	4.95	1.36	0.36	0.02	0	0	0

Nomenclature*Latin symbols*

A	Area, m ²
AE	Annual specific wave energy, kWh/m
B	Radiation damping coefficient, Ns/m
b	Blade height, m
C	Hydrostatic restoring coefficient, N/m
C _A	Input coefficient,
C _D	Drag coefficient,
C _T	Torque coefficient,
C _x	Axial force coefficient,
C _θ	Tangential force coefficient,
cf	Correction factor,
D	Front wall draught length, m
f	Force, N
F	Front wall freeboard length, m
G	Chamber height, m
g	Gravitational acceleration, m/s ²
H	Sea wave height, m
h	Seawater depth, m
H _{m0}	Significant sea wave height, m
J	Rotational inertia, m ² kg
k	Wavenumber, 1/m
l	Chord length, m
m	Mass, kg
$\dot{m}$	Mass airflow rate, kg/s
P	Power, W
p	Pressure, Pa
$\dot{q}$	Volume airflow rate, m ³ /s
R	Chamber radius, m
r	Rotor radius, m
rf	Relative frequency, %
s	Sound speed, m/s
t	Time, s
T	Period, s
T _{m-1,0}	Sea wave energy period, s
U	Blade velocity, m/s
V	Volume, m ³
v	Air absolute velocity, m/s
w	Air relative velocity, m/s
x	Axial coordinate, m
Y	Loss coefficient,
Z	Blade number,
z	Vertical coordinate, m

Greek symbols

β	Relative flow angle, °
η	Sea wave displacement, m
Ξ	Efficiency ratio,
ξ	Efficiency,
ρ	Density, kg/m ³
Σ	Energy ratio,

τ	Torque, Nm
$\ddot{u}$	Water particle acceleration, m/s ²
φ	Turbine flow coefficient,
Ω	Turbine rotational speed, rad/s
ω	Seawater angular frequency, rad/s

Subscripts and superscripts

0	Initial condition
1, 2	Turbine sections
a	Air
add	Added
AM	Ainley and Mathieson
c	Chamber
D	Drag
d	System design
e	Electrical
exc	Excitation
g	Generator
HW	Howell
i	Blade element index
ι	Row index
in	Turbine inlet section
j	Column index
max	Maximum
m	Blade midspan
o	Organic
out	Turbine outlet section
pr	Profile
pto	Power take-off
s	Season
se	Secondary
sf	Skin friction
T	Turbine
t	Total condition
tc	Tip clearance
te	Trailing edge
w	Seawater
x	Axial direction
y	Year
z	Vertical direction
θ	Tangential direction

Acronyms and abbreviations

CIF	Cascade interference factor
IGV	Inlet guide vane
LCOE	Levelised cost of energy
OBD	Oscillating body device
OTD	Overtopping device
OGV	Outlet guide vane
OWC	Oscillating water column
PTO	Power take-off
SWL	Still water level
WEC	Wave energy converter

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