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Wave-to-wire models of wells and impulse turbines for oscillating water column wave energy converters operating in the Mediterranean Sea



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ABSTRACT

Despite the huge potential, energy harnessing from sea waves is often still at a demonstrative stage. Oscillating water column (OWC) wave energy converters have proven to be one of the few suitable solutions to this end. A wave-to-wire analytical code modelling an entire wave energy converter based on the OWC technology, operating with either a Wells or an impulse turbine, was developed. The hydrodynamics, thermodynamics, and aerodynamics of the caisson were determined with a rigid piston approach. Two original low-order aerodynamic models were created for the two turbines, providing an interesting compromise between accuracy and computational cost. Finally, a control strategy was applied to monitor the instant rotor angular velocity and torque in both design and off-design conditions. The simulation tool was applied to screen the geometry of two typologies of air turbines for a specific chamber under the wave conditions of a selected Mediterranean site located in Sardinia (Italy). In particular, annual and seasonal scatter matrices were utilised to define the wave conditions of the site, providing an overview of the seasonal performance variation. The designed Wells and impulse turbines are capable of converting 47.67 and 41.14 MWh/year and operate with an overall efficiency of 5.77% and 4.98%, respectively.

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1. Introduction

It is acknowledged that the sea owns one of the most considerable potentials for renewable energy in the form of waves, currents, tides, and thermal gradients. Inter alia, sea waves seem particularly promising as they combine a notable energy density and a decent availability of possible installations. In particular, wave energy has interesting perspectives as the wave power incident on the world coastline is estimated among 0.5 and 2.2 TW [1,2], which corresponds to more than 10% of the global power consumption [3]. Moreover, an essential aspect of energy from waves is represented by not impacting and not polluting technologies for power conversion. Besides, wave energy converters (WECs) are prone to be coupled with photovoltaic plants and offshore wind turbines [4], leading to clustered production systems. There are three main

categories of WECs [5], classified based on their working principle, namely oscillating water column devices (OWCs), overtopping devices (OTDs), and oscillating body devices (OBDs).

Different from other renewable energy fields, a predominant technology for wave energy conversion has not been identified yet. However, OWC devices are currently considered the most promising technology. They comprise a chamber or caisson, a hollow and partially submerged structure made of concrete or steel, provided with two openings for the interaction with the incident waves and the air crossing the turbine. One hole is underwater and faces the main direction of wave propagation, while the other is above water, and it is connected with the duct embedding the air turbine and open to the atmosphere. The sea waves incident on the caisson cause an oscillating heave motion of the inner column of water that alternatively compresses or expands the volume of air trapped over the water level inside the structure. As a second step, the bidirectional airflow drives a self-rectifying air turbomachine, typically a Wells or an impulse turbine, embedded in the duct and coupled

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with an electrical generator [6]. Their main advantages are the structural simplicity, reliability, and the motion of mechanical components only above water. Despite these interesting features, one of the main issues still hindering their widespread diffusion is the relatively high levelised cost of energy (LCOE), ranging from 90 to 101 €/MWh for onshore installations and between 180 and 490 €/MWh for offshore installations. Comparing these values with onshore (60 €/MWh) and offshore (102 €/MWh) wind energy, solar photovoltaic (68 €/MWh), and geothermal energy (50 €/MWh), it is apparent that wave energy is presently not competitive with other renewable energies in terms of cost-effectiveness [7,8]. To reduce the LCOE, an increase of the efficiency of the overall power transformation process, from incident sea waves to electric connection, is pivotal at the present stage of the research. On this basis, for example, Shaaban investigated the cost reduction of an OWC plant embedding a Wells turbine by acting on the geometric optimisation of the rotor, resulting in the reduction of the LCOE up to 15–20% [9]. Moreover, the performance of an OWC system varies significantly with different wave climates, thus requiring a specific design of the main components for each application site [10,11].

Recently, a comprehensive literature review on existing design solutions for these devices and the modelling approaches was proposed by the authors of this study [12]. Upon examination of state of the art, it is apparent that the usual approach carried out so far relies on the separate design and optimisation of each of the three converters, represented by the chamber structure (primary converter), the air turbomachine (secondary converter), and the electrical generator (tertiary converter). However, the presence of different physical phenomena and their complementarity in the correct functioning of an OWC system makes this approach often not effective. Indeed, it leads to slow convergence towards the optimal configuration or even an incomplete exploration of the design space. Moreover, properly accounting for the fluid dynamic connection between the chamber and the global energy conversion of the system.

Up to now, only a few wave-to-wire models comprising the integrated modelling schemes of the entire system were realised [12]. In particular, they usually apply very simplified approaches for predicting the turbine aerodynamics, limiting the possibility of an inner coupling with the flow conditions delivered by the chamber.

In this framework, the main scope of the present research is to improve the analytical modelling of OWC systems with the development of a complete and reliable wave-to-wire model for engineering applications, able to account for a Wells or an impulse turbine. Overall, the three converters are solved in a coupled fashion to determine the power conversion from wave to wire.

2. Methodology

This section presents the theoretical models used to jointly solve the hydrodynamics, aerodynamics, and thermodynamics of the caisson with the aerodynamics and dynamics of the turbomachine, and the dynamics of the electric generator.

2.1. Primary converter model

The modelling of the primary converter of the oscillating water column system relies on the linear wave theory, which assumes that the fluid flow is inviscid, incompressible, and irrotational. The caisson model is based on the so-called rigid piston concept, primarily diffused in the literature [13]. This approach is suitable for the present application as the dimensions of the structure are small in comparison to the wavelength of the incident wave.

The following additional assumptions are made: the OWC device is considered fixed (not floating) and detached from the sea bottom, the seawater depth is constant, and the incoming waves are unidirectional. In these conditions, the only possible motion of the water column inside the OWC is the heave oscillation.

The force balance on the rigid piston, which models the water column inside the OWC, is expressed in Eq. (1) by applying Newton's second law.

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

where, on the left side, m_w is the water column mass, z is the free surface level related to the still water level (STL), positive if over it, B is the damping coefficient, and C is the hydrostatic restoring coefficient, while, on the right side, f_{exc} is the excitation force determined by the hydrodynamic pressure of the incident waves acting on the bottom of the water column, f_{add} is the force resulting from the added mass, and f_{pto} is the force induced by the air pressure in the upper part of the chamber.

The mass of the water column inside the caisson is defined based on the horizontal cross-section of the chamber A_c , the length of its seaward wall draught D , and the density of the seawater ρ_w , as expressed by Eq. (2).

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

The hydrostatic restoring coefficient is obtained with Eq. (3) as a function of the seawater density and the horizontal cross-section of the chamber.

$$C = \rho_w g A_c \quad (3)$$

The damping coefficient is computed with Eq. (4) as a dynamic function of the restoring coefficient and the inertial properties of the system of mass and added mass [13–15].

$$B = 0.2 \sqrt{C(m_w + m_{add})} \quad (4)$$

where g is the gravitational acceleration and m_{add} is the added mass, which was attained with experimental free decay tests previously performed [12].

According to the linear wave theory, the water surface displacement η_w for an incoming sea wave of height H is expressed by Eq. (5) as a sinusoidal function.

$$\eta_w = \frac{H}{2} \cos(kx - \omega t) \quad (5)$$

where x is the position in the axial direction, t is the time, k is the wavenumber, and ω is the wave angular frequency. The wavenumber is obtained from the solution of the dispersion relation for the distinctive water depth h , as formulated in Eq. (6).

$$\omega^2 = kg \tanh(kh) \quad (6)$$

The dynamic pressure of the seawater p_w of Eq. (7) reduces proportionally to the depth in the vertical direction.

$$p_w = \eta_w \rho_w g \frac{\cosh(k(h+z))}{\cosh(kh)} \quad (7)$$

Assuming that the dimensions of the chamber are remarkably smaller than the wavelength of the incident sea waves, the wave diffraction effects are negligible. Thus, the exciting force at the level of the front wall draught D can be calculated with Eq. (8).

$$f_{exc} = \eta_w A_c \rho_w g \frac{\cosh(k(h-D))}{\cosh(kh)} \quad (8)$$

The added mass force f_{add} acting on the rigid piston due to the added mass is expressed by Eq. (9) based on the vertical acceleration $\dot{v}_{z,w} - \ddot{z}$ of the seawater particles regarding the rigid piston.

$$f_{add} = m_{add}(\dot{v}_{z,w} - \ddot{z}) \quad (9)$$

The force f_{pto} is determined with Eq. (10) from the instant difference in pressure p_c between the air inside the caisson and the air in atmospheric conditions.

$$f_{pto} = -A_c p_c \quad (10)$$

By properly substituting the previous formulations into Eq. (1), the equation of motion of the rigid piston is defined by Eq. (11).

$$m_w \ddot{z} + B \dot{z} + A_c \rho_w g z = \eta_w A_c \rho_w g \frac{\cosh(k(h-D))}{\cosh(kh)} + m_{add}(\dot{v}_{z,w} - \ddot{\eta}_w) - A_c p_c \quad (11)$$

From a thermodynamic point of view, the hypotheses of adiabatic and isentropic transformations are applied for the alternate compression and expansion of the air inside the caisson [16]. The mass airflow rate crossing the turbine may be expressed as in Eq. (12). Its sign is positive during the outflow phase when the air flows from the caisson to the atmosphere and negative vice versa during the inflow phase.

$$\dot{m} = -\frac{d(\rho_a V)}{dt} \quad (12)$$

where ρ_a and V are the air density and volume in the chamber, respectively.

The pressure variation inside the caisson determines the mass airflow rate based on the assumptions done, as stated in Eq. (13).

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

where s is the sound speed in the air, $\dot{q}$ is the volume airflow rate, and V_0 is the unperturbed air volume of the chamber.

By applying the hypothesis of a very low ratio between the volumes $A_c z$ and V_0 , the former term can be neglected, and Eq. (14) becomes a linear relation.

$$\dot{m} = \rho_a \dot{q} - V_0 \frac{1}{s^2} \frac{dp_c}{dt} \quad (14)$$

2.2. Secondary converter model

The analytical models for the two air turbines were developed to obtain fast and sufficiently reliable engineering tools of practical combination inside the overall model. The aerodynamics of monoplane isolated Wells turbines and axial impulse turbines was solved by modelling the turbomachine cascades as groups of blades consisting of a finite set of airfoils stacked along the blade span.

2.2.1. Analytical model for the wells turbine

An actuator disc model of a monoplane isolated Wells turbine was generated [17,18]. The main hypotheses underlying the theory concern the characteristics of the flow, which has to be steady, axisymmetric, and inviscid.

The blade element momentum theory was applied, considering a single blade as a finite group of airfoils stacked along the blade

span. Each blade element is defined by every couple of adjacent airfoils, has distinctive geometry, undergoes particular flow conditions, and moves with a specific rotational speed [19]. A sensitivity analysis was carried out for these reasons, obtaining that the use of 50 blade elements is the optimal compromise between accuracy and computational time.

Radial equilibrium conditions are imposed at the inlet and outlet of the computational domain, far upstream and far downstream of the actuator disc. For a monoplane isolated Wells turbine, the flow at the domain inlet is axial and uniform for both the phases of inflow and outflow (Fig. 1).

The evaluation of the performance of Wells turbines relies on the classical airfoil theory. The lift and drag coefficients of the Sandia tables are utilised in the developed model [19]. The decrease of the power output caused by the airflow leakage through the tip clearance is considered using the induced drag coefficients, which are added to the corresponding experimental values to calculate the equivalent drag coefficients C_{Dtc} , as in Eq. (15). The induced drag is a function of the lift coefficient, the tip clearance t , the aspect ratio AR , and the pitch Δ calculated at the blade tip [20].

$$C_{Dtc,i} = C_{D,i} + 0.7 \frac{C_{L,i} t}{AR \Delta} \quad (15)$$

where subscript i refers to the generic blade element.

Subsequently, the lift and drag coefficients C_L and C_D of the airfoil are used in Eqs. (16) and (17) to calculate the lift and drag forces f_L and f_D exerted by a flow with relative velocity w impacting an infinitesimal blade element of chord length l .

$$df_{L,i} = \frac{1}{2} \rho_{a,i} l_i C_{L,i} w_{1,i}^2 dr \quad (16)$$

$$df_{D,i} = \frac{1}{2} \rho_{a,i} l_i C_{Dtc,i} w_{1,i}^2 dr \quad (17)$$

where r is the turbine radius and subscript 1 refers to the domain inlet.

The lift and drag forces are combined in Eqs. (18) and (19) through trigonometry to obtain the axial component f_x and the tangential component f_θ of the force exerted by the fluid on each blade of the rotor cascade.

$$df_{x,i} = df_{L,i} \cos \beta_{1,i} + df_{D,i} \sin \beta_{1,i} = \frac{1}{2} \rho_{a,i} l_i C_{x,i} w_{1,i}^2 dr \quad (18)$$

$$df_{\theta,i} = df_{L,i} \sin \beta_{1,i} - df_{D,i} \cos \beta_{1,i} = \frac{1}{2} \rho_{a,i} l_i C_{\theta,i} w_{1,i}^2 dr \quad (19)$$

where C_x and C_θ are the axial and circumferential force coefficients, respectively.

Wells turbines have significant values of midspan solidity, typically ranging from 0.4 to 0.6 to combine efficient and self-starting operation. Such values of solidity determine a mutual interference amongst the row blades brought on by either inviscid and viscous effects. Therefore, a correction factor CIF shall be introduced. This factor is defined as a function of the rotor solidity σ and is expressed by Eq. (20) to consider the blade mutual interaction [21].

$$CIF = \frac{1}{1 - \sigma_i^2} \quad (20)$$

The axial and tangential force coefficients are modified through multiplication by the cascade interference factor, as in Eqs. (21) and (22). The modified values are utilised in Eqs. (18) and (19) instead of

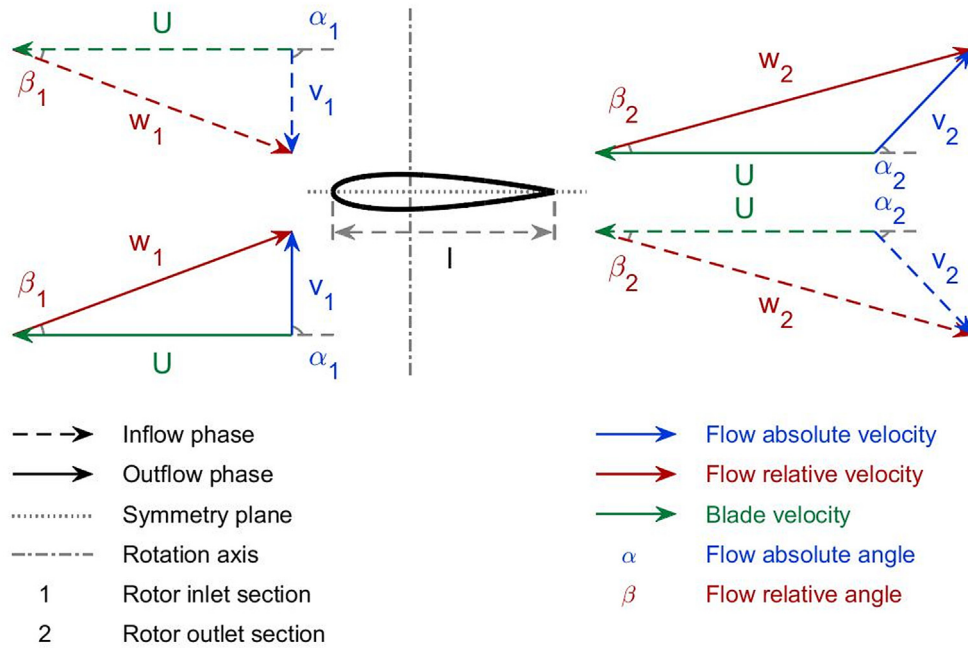


Fig. 1. Scheme of the blade-to-blade section and velocity triangles of a monoplane isolated Wells turbine with NACA0015 airfoil.

the original values of C_x and C_θ to compute the axial and tangential forces perturbed by the blade interaction.

$$C_{xCIF,i} = C_{IFC_{x,i}} \quad (21)$$

$$C_{\theta CIF,i} = C_{IFC_{\theta,i}} \quad (22)$$

The power P_T provided by the airflow to the rotor blades of the turbine is obtained with Eq. (23) based on Eqs. (19) and (22).

$$P_T = \int_{rhub}^{rtip} Z \Omega r_i df_{\theta CIF,i} = \frac{1}{2} \int_{rhub}^{rtip} Z \rho_{a,i} l_i C_{\theta CIF,i} w_{1,i}^2 \Omega r_i dr \quad (23)$$

where Ω is the rotor angular velocity and Z is the number of blades constituting the cascade.

The static pressure variation across the rotor is calculated with Eq. (24) by imposing the balance between the axial force acting on the rotor and the pressure force acting on the flow.

$$p_{1,i} - p_{2,i} = \frac{\rho_{a,i} l_i Z C_{x,i} v_{x1,i}^2}{4\pi r_i \sin^2 \beta_{1,i}} \quad (24)$$

where p is the static pressure, v is the absolute velocity of the airflow, and subscript 2 denotes the domain outlet.

The definition of the velocity triangle at the rotor outlet starts with calculating the absolute angle α_2 of the flow exiting the rotor blades by using Eq. (25). Then, the mass conservation and trigonometric relations are applied to obtain the components of the absolute and relative flow velocities and angles.

$$\cot \alpha_{2,i} = \cot \alpha_{1,i} + \frac{Z l_i C_{\theta,i}}{4\pi \beta_{1,i}} \quad (25)$$

By applying the radial equilibrium and assuming a constant total pressure along the blade span [21], the variation of the axial velocity downstream of the actuator disc can be calculated through Eq. (26).

$$\frac{d(v_{x,i}^2)}{dr} = -\frac{1}{4\pi^2 r_i^2} \frac{d(2\pi r_i v_{\theta 2,i})^2}{dr} \quad (26)$$

Finally, an iterative process based on energy conservation is implemented to compute the airflow field in the computational domain [19].

2.2.2. Analytical model for the impulse turbine

The typical configuration of an axial flow impulse turbine for wave energy converters based on the oscillating water column technology has a single rotor cascade and a couple of stator cascades (Fig. 2).

The two stators are symmetrically installed regarding the plane

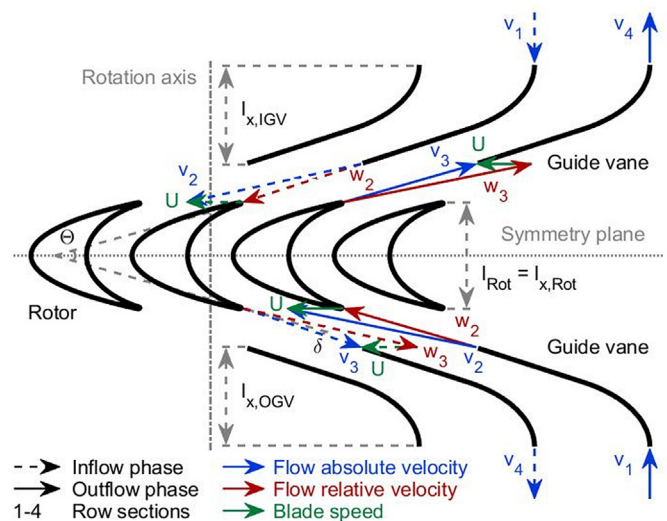


Fig. 2. Scheme of the blade-to-blade section and velocity triangles of an axial impulse turbine.

of symmetry of the three rows and are normal to the axis of rotation. The stators operate as inlet guide vane (IGV) when impacted by the airflow upstream the rotor or outlet guide vane (OGV) when the impact occurs after crossing the rotor. The operation changes based on the flow phases. The IGV is an accelerating row, while the OGV is a diffusive row [12]. Since the machine functions with a bidirectional airflow, its cascades need to have symmetric or adjustable geometry.

The analytical model of axial flow impulse turbines developed relies on applying experimental correlations representing the best practice for the theoretical design of axial turbomachines [12,19]. In analogy with the code of the Wells turbine, a blade element model was applied, and 50 blade elements were set along the blade span for the domain discretisation after having performed a sensitivity analysis.

In the beginning, the turbine geometry is defined by setting the profile shape and the blade number of either the rotor and stator cascades. Based on these parameters, the inlet flow conditions are determined. Subsequently, the criterion of Zweifel and the criterion of Carter and Hughes are applied. An iterative process to compute the direction of the airflow and the length of the blade chords is performed on the basis of the axial and uniform airflow at the machine inlet [18].

In Eq. (27), the criterion of Zweifel establishes 0.8 as the optimal value for the ratio of real to ideal blade loading for the minimisation of the aerodynamic losses.

$$\frac{\Gamma_i}{\Gamma_{id,i}} = 2 \frac{t_i}{l_{x,i}} \cos^2 \alpha_{2,i} (\tan \alpha_{1,i} + \tan \alpha_{2,i}) = 0.8 \quad (27)$$

where Γ and Γ_{id} are the real and ideal loading of rotor blades, respectively, t is the cascade pitch, l_x is the axial chord length, α is the absolute flow angle relative to the axial direction, and subscripts 1 and 2 are referred to the inlet and outlet sections of the row, respectively.

The flow deviation across a cascade is calculated with Eq. (28) as a function of its geometry using the criterion of Carter and Hughes.

$$\delta_i = M_{CH,i} \theta_i \frac{t_i}{l_i} \quad (28)$$

where δ is the airflow deviation, M_{CH} is a parameter that depends upon the stagger angle, and θ is the camber angle, which is the angle difference of the leading edge with the trailing edge (Fig. 2).

The iterative procedure starts with the initialisation of the values of the chord length for the three rows by applying Zweifel's criterion according to the geometric angles. Then, the flow deviation across the rows is calculated through the criterion of Carter and Hughes. The values of the chord length of the cascades are finally determined with Zweifel's criterion. The computations are iterated until the tolerance of 10^{-3} set for the chord length is reached to assess the convergence.

Once the velocity triangles are obtained at the sections upstream and downstream of the IGV and the rotor row, an iterative process is performed to compute the airflow deviation at the exit of the OGV and define the outlet velocity triangle of the turbine. Since the outlet guide vane diffuses the airflow, the empirical rule defined by Howell in Eq. (29) is applied to estimate the nominal deviation of the airflow based on the camber angle and the ratio between the cascade pitch and the chord length [18].

$$\delta_i^* = M_{HW,i} \theta_i \left(\frac{t_i}{l_i} \right)^{\frac{1}{2}} \quad (29)$$

where δ^* is the nominal airflow deviation and M_{HW} is a parameter depending on the blade setting and the camber line shape. A tolerance of 10^{-3} is imposed for the flow angle exiting the cascade

to assess the convergence of the iterative procedure.

The power P_T acting on the rotor blades is obtained through Euler's theorem for turbomachines, which expresses with Eq. (30) the specific work in terms of the passage area of the airflow and velocity triangles upstream and downstream the rotating cascade.

$$P_T = \int_{rhub}^{rtip} \rho_{a,i} A_{T,i} v_{x,i} \Omega (v_{\theta 2,i} + v_{\theta 3,i}) dr \quad (30)$$

where A_T is the area of the turbomachine annulus, subscripts x and ϑ represent the axial and tangential absolute velocities, respectively, and subscripts 2 and 3 specify the sections at the entrance and exit of the rotating cascade, respectively.

The assessment of the aerodynamic losses, referable to the inlet guide vane and the rotor, is carried out with the correlation of Ainley and Mathieson [18]. For the two rows, the overall loss coefficient γ_{AM} is obtained with Eq. (31) by superimposing the profile, secondary, tip clearance leakage, and trailing edge loss coefficients γ_{pr} , γ_{se} , γ_{tc} , and γ_{te} , respectively [12].

$$\gamma_{AM} = [(\gamma_{pr} + \gamma_{se})cf + \gamma_{tc}] \gamma_{te} \quad (31)$$

where cf is a correction factor depending upon the Reynolds number.

Since the outlet guide vane is a diffusive cascade, its aerodynamic losses are evaluated by applying the correlation of Howell [18]. The estimation of the global losses of the rows relies on the definition of an equivalent global drag coefficient C_{DHW} provided by the sum of the profile, skin friction, and secondary drag coefficients denoted in Eq. (32) as $C_{D,pr}$, $C_{D,sf}$, and $C_{D,se}$, respectively [12].

$$C_{D,HW} = C_{D,pr} + C_{D,sf} + C_{D,se} \quad (32)$$

Since the overall loss coefficient γ_{AM} and the equivalent global drag coefficient C_{DHW} are related to the total pressure losses across the cascades, the values of pressure corresponding to all the sections at the inlet and outlet of the rows are obtained [12,18].

2.3. Tertiary converter model

The third energy conversion phase of an oscillating water column device involves the joint analysis of the turbomachine and electrical generator. The implemented control algorithm monitors the instant rotor angular velocity and torque in both design and off-design conditions.

In the simplified control model applied, the aerodynamic friction losses due to the windage are neglected. This implies that the relation between the turbine and generator torque can be formulated with Eq. (33).

$$J\dot{\Omega} = \tau_T - \tau_g \quad (33)$$

where J and $\dot{\Omega}$ are the rotational inertia and angular acceleration of the rotor, respectively, τ_T is the torque of the turbomachine, and τ_g is the torque of the electric generator.

For negative values of the rotor torque, the connection between the turbine and the electric generator is interrupted with a consequent reduction of speed based on the rotational inertia. Conversely, when the torque is positive, the logic relying on the design rotational speed tracking is pursued with reference to the design value of the rotor Ω_0 , as in Eq. (34).

$$\begin{cases} \Omega(t) \geq \Omega_0 \rightarrow \tau_g = \tau_T \\ \Omega(t) < \Omega_0 \rightarrow \tau_g = 0 \end{cases} \quad (34)$$

The turbomachine is allowed to increase its rotational velocity, and the electric load is cut off up to the achievement of the design velocity if the turbine torque is positive. When the angular speed is equal or superior to the design velocity, the generator is immediately coupled. This approach allows the system to balance the torque and keep the system rotating, avoiding an uncontrolled speed increase.

Finally, the electrical power transformed by the generator is computed with Eq. (35) by reducing the turbomachine power according to the organic efficiency ξ_o and the electric efficiency ξ_e , based on Eq. (23) or Eq. (30). The hypothesis of organic and electrical efficiencies equivalent to 0.97 is applied in compliance with the typical magnitude of these losses.

$$P_e = \xi_o \xi_e P_T \quad (35)$$

2.4. Wave-to-wire model

The structure of the developed code is illustrated in the flow chart of Fig. 3. At first, every wave condition representative of the wave spectrum of a specific application site is determined. Subsequently, an iterative procedure for the joint solution of the mathematical models of the primary, secondary, and tertiary converters is performed. This approach allows the combined solution of the thermo-fluid dynamics of the seawater and air inside the caisson and the fluid dynamics of air turbomachine, dynamically coupled with the electric generator. The machine functioning maps are utilised to iteratively match the characteristics of the airflow generated by the chamber and the damping imposed by the turbine, instantly coupled and regulated in terms of torque and rotational speed through the control strategy.

3. Results and discussion

The developed model was utilised to assess the operation of a wave energy converter relying on the oscillating water column technology for a possible future installation in a selected hot spot of a coastal area of Italy.

3.1. Study case

The site chosen to apply the model is placed in Italy, in the north-west of Sardinia. The WGS84–UTM32 coordinates of the location are: latitude 4,509,971 m N and longitude 424,733 m E, as illustrated in Fig. 4. The point has a water depth of 50 m and is characterised by an average specific wave power of 10.5 kW/m regarding the wavefront length. Therefore, it was identified as a highly energetic spot for the Italian wave climate [12,22]. The OWC system under consideration could be conceived as integrated into an offshore construction, as a tension-leg platform or a very large floating structure [23]. Indeed, devices fixed and detached regarding the sea bottom are often studied in the literature with reference to intermediate or deep-water conditions [24–27].

A 20 years-long hindcasting dataset was obtained with the spectral wave model WAVEWATCH III® [28,29] to evaluate the wave energy potential of the site. The model provides hourly results for the wave conditions of the whole Mediterranean Sea from 2000 to 2019. Every sea state is defined by a couple of significant wave height H_{m0} and wave energy period $T_{m-1,0}$ using the JONSWAP spectrum. The corresponding specific value of deep-water power P_w regarding the wavefront length is obtained through Eq. (36). The use of this formula may introduce some approximations, especially for the longest waves, due to the dissipation of part of the energy in propagating from deep waters to the depth of 50 m. However, this effect is very limited in the present case.

$$P_w = \frac{1}{64} \frac{g^2}{\pi} \rho_w H_{m0}^2 T_{m-1,0} \quad (36)$$

The distribution of annual specific wave energy AE resulting from combining the specific wave power of Eq. (36) and the time of occurrence of every wave condition during one year is reported in Fig. 4 for the selected site.

For every wave status, the rigid piston model is applied for a regular wave with height H and period T conveying the same energy of the related irregular wave motion. Hence, the relations of Eq. (37) were assigned.

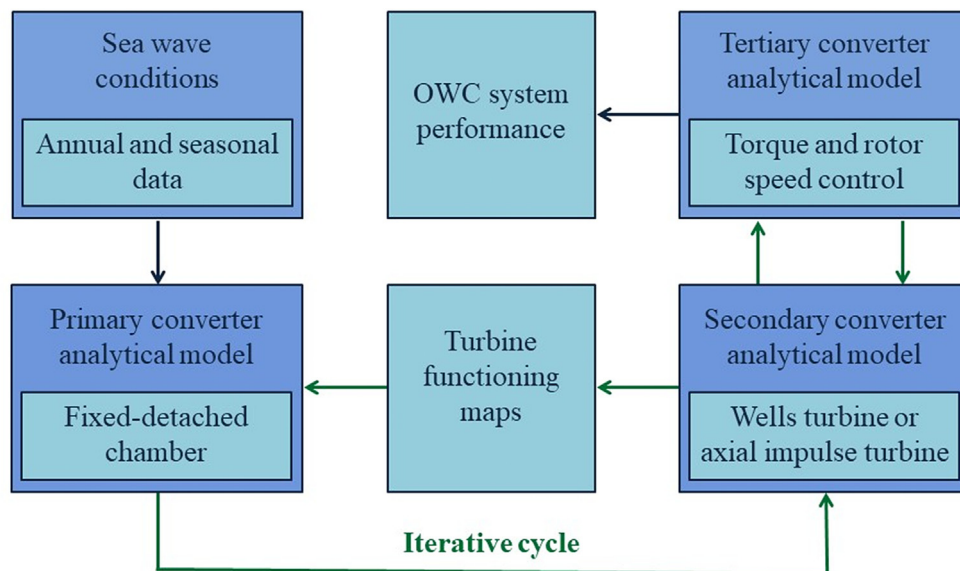


Fig. 3. Flow chart of the main stages of the developed wave-to-wire model.

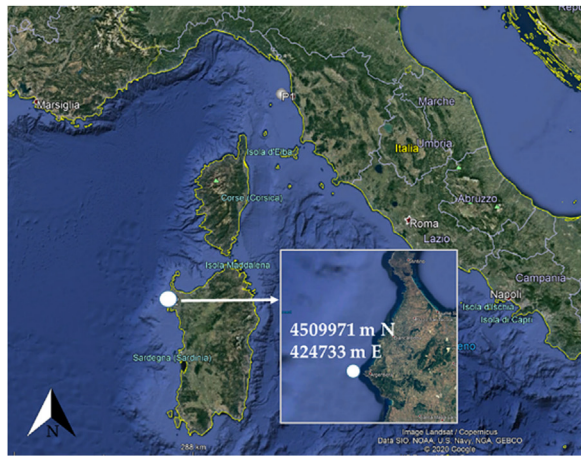


Fig. 4. (Left) Geographical map and coordinates of the highly energetic wave climate chosen to apply the developed model and (right) the scatter matrix of the distribution of annual specific wave energy.

AE [kWh/m]		ANNUAL VALUES											
		$T_{m-1,0}$ [s]											
H_{m0} [m]	> 9	<3	3-4	4-5	5-6	6-7	7-8	8-9	9-10	10-11	> 11	a	12 > 12
	8-9	0	0	0	0	0	0	0	0	135	401	0	0
	7-8	0	0	0	0	0	0	0	41	1379	263	0	0
	6-7	0	0	0	0	0	0	0	1772	2601	123	0	0
	5-6	0	0	0	0	0	0	816	6316	1491	62	0	0
	4-5	0	0	0	0	0	324	7570	5834	718	65	0	0
	3-4	0	0	0	0	277	5733	10390	3057	323	29	4	0
	2-3	0	0	0	338	4468	10345	5334	1227	75	5	6	0
	1-2	0	9	923	5550	6829	3047	609	34	4	0	0	0
	< 1	21	708	1033	499	99	6	1	0	0	0	0	0

$$\begin{cases} H = \frac{H_{m0}}{\sqrt{2}} \\ T = T_{m-1,0} \end{cases} \quad (37)$$

The scatter matrices of specific wave energy and relative frequency of occurrence rf were determined on a yearly basis for each wave condition. The data of Tables 1 and 2 applied in the wave-to-wire model were extracted from the data of Fig. 4 by limiting the maximum wave height considered to 5 m for avoiding the entry of seawater in the turbine duct. Despite the data reduction regarding the complete set, the wave conditions analysed contribute to about 83% of the global energy available on the site.

The values of the specific wave energy and relative frequency of the wave conditions investigated for the four seasons of the year are provided in the scatter matrices reported in the Appendix.

3.2. Turbine selection

A selection procedure of a monoplane isolated Wells turbine and an axial impulse turbine was performed to provide a preliminary design of two wave energy converters to evaluate their possible

future installation in the chosen site. Two sets of parametric analyses were carried out, defining as objective function the overall annual energy harvested by the system, which was maximised.

The geometry and properties of the chamber were defined as indicated in Table 3 based on experiments previously conducted at a laboratory scale. A chamber with a calibrated orifice was tested under wave conditions typical of a moderate climate of the Mediterranean Sea to optimise primary energy conversion [23]. The rules of similitude analysis were applied to scale up by a factor of 50 the data of the prototype [12].

The parametric analyses were performed by varying the geometrical features and operating conditions of the two turbomachines in compliance with the ranges indicated in Table 4. The intervals were defined based on the existing prototypes of Wells and impulse turbines to investigate an effective test plan for the preliminary selection of the secondary converter [6,30–34]. A constant tip clearance equal to 1% of the blade height was considered.

Once the best-suited parameters of the test plan were determined, the analysis of the optimal design velocity was further refined. A range of 300 rpm centred around the value previously obtained was examined with intervals of 25 rpm for both machines.

Table 1

Scatter matrix of the annual specific wave energy AE distribution of the wave conditions appraised, for classes of different significant height H_{m0} and energy period $T_{m-1,0}$ [12].

AE, kWh/m		$T_{m-1,0}$, s										
H_{m0} , m	4.5	0	0	0	0	0	324	7570	5834	718	65	0
	3.5	0	0	0	0	277	5733	10390	3057	323	29	4
	2.5	0	0	0	338	4468	10345	5334	1227	75	5	6
	1.5	0	9	923	5550	6829	3047	609	34	4	0	0
	0.5	21	708	1033	499	99	6	1	0	0	0	0
	0.5	21	708	1033	499	99	6	1	0	0	0	0

Table 2

Scatter matrix of the annual relative frequency rf distribution of the wave conditions appraised, for classes of different significant height H_{m0} and energy period $T_{m-1,0}$ [12].

rf , %		$T_{m-1,0}$, s										
H_{m0} , m	4.5	0	0	0	0	0	0.051	1.049	0.723	0.081	0.007	0
	3.5	0	0	0	0	0.083	1.488	2.380	0.626	0.060	0.005	0.001
	2.5	0	0	0	0.235	2.623	5.263	2.394	0.493	0.027	0.002	0.002
	1.5	0	0.028	2.174	10.70	11.14	4.306	0.760	0.038	0.004	0	0
	0.5	0.795	19.31	21.90	8.658	1.457	0.082	0.013	0.002	0	0	0
	0.5	0.795	19.31	21.90	8.658	1.457	0.082	0.013	0.002	0	0	0

Table 3

Principal geometric parameters and properties of the caisson [12].

Chamber outer section	Chamber inner section	Chamber height	Front wall draught length	Front wall freeboard length	Added mass	Undamped natural frequency
A_{cext}, m^2	A_c, m^2	G, m	D, m	F_c, m	m_{add}, kg	$\omega_{un}, rad/s$
100.00	82.50	14.50	4.50	8.00	535000	0.95

Table 4

Design space of the parametric analyses performed for the Wells and impulse turbines.

Parameter	Wells turbine			Impulse turbine		
	Minimum	Maximum	Interval	Minimum	Maximum	Interval
Rotational speed, rpm	300	1200	300	300	1200	300
Turbine casing radius, m	0.50	1.00	0.05	0.50	1.00	0.05
Hub-to-tip ratio, -	0.50	0.70	0.05	0.60	0.70	0.05
Rotor blade aspect ratio, -	0.50	1.50	0.10	—	—	—
Stator blade number, -	—	—	—	26	30	2
Rotor blade number, -	5	12	1	30	40	2

Table 5

Geometry and rotor angular velocity of the selected Wells and impulse turbines.

Parameter	Wells turbine	Impulse turbine
Rotational speed, rpm	950	300
Turbine casing radius, m	1.00	0.85
Hub-to-tip ratio, -	0.50	0.70
Rotor blade aspect ratio, -	1.10	0.76
Stator blade number, -	—	26
Rotor blade number, -	5	30
Tip clearance, %	1.00	1.00
Rotational inertia, $m^2 \text{ kg}$	25.61	104.12

For the Wells turbine, a rotational speed of 950 rpm was chosen, while the value of 300 rpm was confirmed for the impulse turbine.

The geometric parameters and the rotational speed of the Wells and impulse turbines leading to the maximisation of the annual energy harvesting according to the design space are resumed in Table 5.

The rotor angular velocity of the selected Wells turbine is approximately 3.17 times greater and the radius of its casing is 17.65% bigger concerning the impulse turbine, leading to a ratio of 3.73 between the blade speeds. This implies that the acoustic emissions attributable to aerodynamics are significantly higher for the former turbomachine. The rotational inertia was determined considering the rotors of the two turbines made of marine grade aluminium EN AW 5083 to either achieve corrosion resistance and limit the production costs [35]. The greater inertia of the impulse turbine provides the advantage of levelling the oscillations of power conversion due to flywheel energy storage and has the drawback of increasing the forces necessary to run up the rotor at the design speed.

3.3. Operating maps

The operating maps were obtained to identify the most relevant parameters regulating the turbine functioning by considering a yearly energy extraction scenario. The annual energy harvesting ratio $\Sigma_{d,y}$ was defined in Eq. (38) to rank the performance of the system configurations. For each turbine typology, this parameter is the ratio of the annual energy transformed by a specific configuration to the maximum energy converted amongst the configurations examined. Therefore, the annual energy harvesting ratio of the best-suited layout investigated is unitary for either Wells and impulse turbines.

$$\Sigma_{d,y} = \frac{E_{d,y}}{E_{max,y}} \quad (38)$$

where E is the energy harvested, subscript d refers to the device configuration examined, subscript max indicates the optimal layout maximising energy extraction, and subscript y refers to the year.

Figs. 5 and 6 show the annual energy harvesting ratio for the Wells and impulse turbines, respectively, as functions of the hub-to-tip ratio and the turbine casing radius for different values of rotational speed. For both the system typologies, only the turbine configurations functioning at a velocity lower than the critical value were included in the presented results. Indeed, the relative Mach number at the blade tip has to be limited to 0.47 for avoiding the aerodynamic losses provoked by transonic flow and shock waves that may occur in this region at a high rotational speed [31]. For the Wells turbines, the condition of self-starting functioning was imposed in addition to the previous requirement for ensuring the ability to run up autonomously to the design velocity.

The maps of the energy harvesting ratio of Fig. 5 show that the Wells turbine requires a relatively high velocity of around 900 rpm for properly exploiting the resource. For the cases with 300 and 600 rpm, the increase of the turbine radius determines a positive variation of annual energy extraction for all the levels of hub-to-tip ratio due to the rise of the flow rate crossing the turbomachine. At the speed of 900 rpm, the increase of the turbine casing radius has a positive effect on the performance for the lowest values of hub-to-tip ratio analysed, while for the highest values the impact of an excessive increase of the radius is negative. In particular, a value of 1 m maximises the energy extraction for the ranges of the test plan. For the velocity of 1200 rpm, the magnitude of the turbine radius enhancing the performance raises with the reduction of the hub-to-tip ratio. This geometrical parameter has a significant influence on the energy extraction for the two highest velocities analysed. A reduction of the hub-to-tip ratio generally enhances the performance, as it increases the airflow rate. Especially, a value of 0.5 maximises the annual energy harvesting for the design space. Furthermore, reducing the hub-to-tip ratio decreases the rotor solidity required to independently run up the rotor to a specific rotational speed. This is beneficial to the system operation, as limiting solidity diminishes the blockage of the airflow across the turbine and thus the requirement of pneumatic power from the primary converter. The rotor solidity calculated at blade tip ranges from 0.35 to 0.45 for the best-suited configurations.

The maps of the annual energy harvesting ratio of Fig. 6 indicate that impulse turbines harvest the maximum energy at the

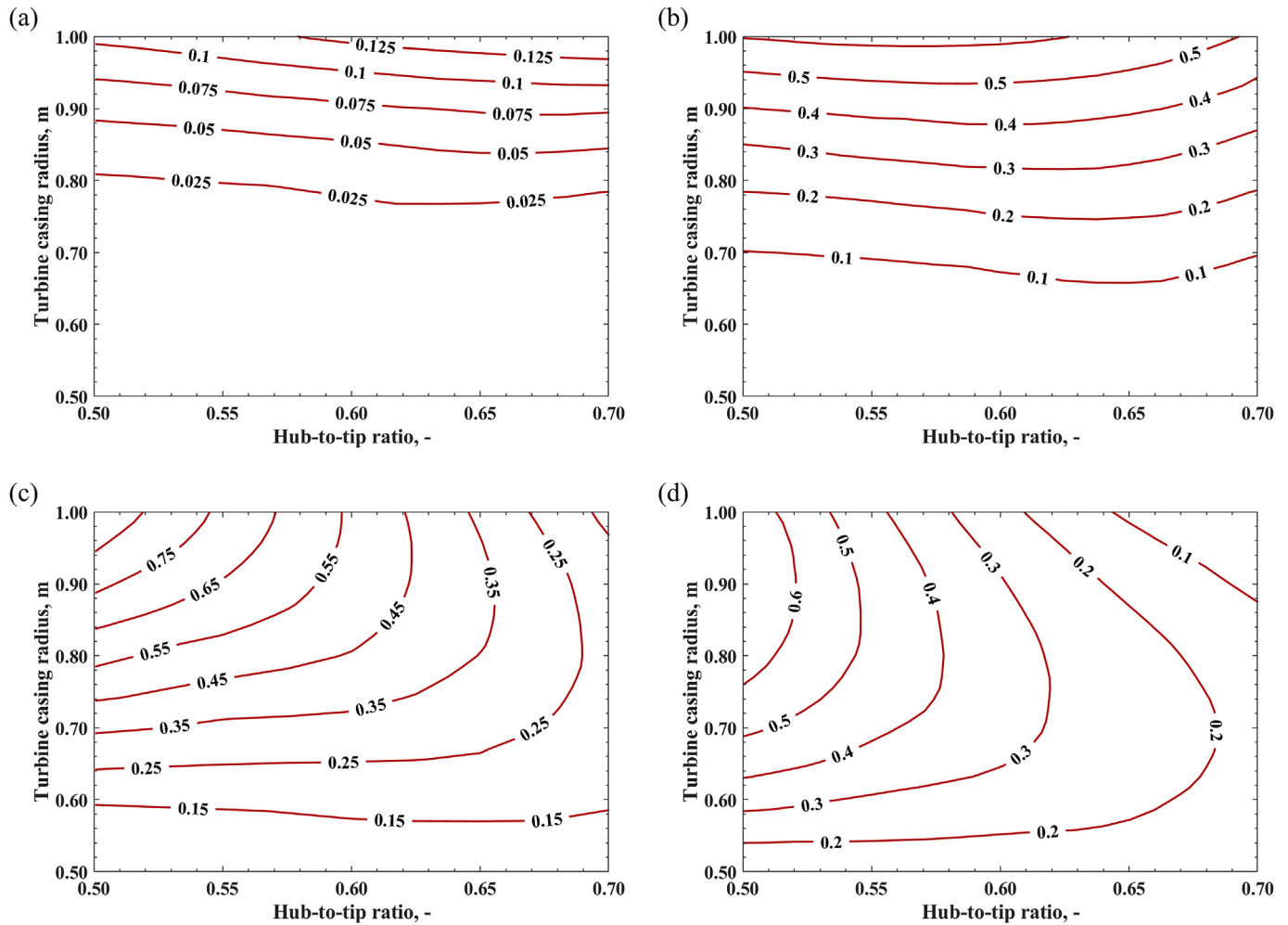


Fig. 5. Energy ratio of the Wells turbine as a function of the hub-to-tip ratio and turbine casing radius for values of rotor angular velocity equal to (a) 300, (b) 600, (c) 900, and (d) 1200 rpm.

rotational speed of 300 rpm. The turbine casing radius has significant effects for all the achieved maps. For the two lowest values of velocity, the decrease of the system performance occurs with an excessive rise of the radius, while for the two highest velocities increasing this parameter determines the reduction of energy conversion in the entire design space. In particular, a radius of 0.85 m maximises the energy extracted during one year. Regarding the hub-to-tip ratio, the effects on the system operation are mainly noticeable for the two lowest values of rotational velocity. This geometric parameter is between 0.65 and 0.70 for the best-suited configurations.

For the Wells turbine, the aspect ratio of the rotor blades was assigned according to the test plan of Table 4. Conversely, for the impulse turbine, this parameter depends on the length of the blade chord determined by applying the criterion of Zweifel. This criterion imposes an optimal value of the real blade loading based on its ideal value for all the investigated configurations. The aspect ratio of the rotor blades maximising energy harvesting ranges from 1.00 to 1.20 for the Wells turbine and between 0.75 and 1.15 for the impulse turbine. The most suited values of the aspect ratio of the stators are between 1.00 and 1.50 for the latter machine.

3.4. Annual system performance

The results of the aerodynamics of the Wells and impulse

turbines achieved with the wave-to-wire model are shown using the non-dimensional and dimensional parameters traditionally applied in turbomachinery [18]. The non-dimensional parameters of equations from Eq. (39) to Eq. (42) are the turbine flow coefficient ϕ , the input coefficient C_A , the torque coefficient C_T , and the turbine efficiency ξ .

$$\phi_m = \frac{v_{x,m}}{U_m} \quad (39)$$

$$C_{A,m} = \frac{(p_{t,lin} - p_{out})\dot{q}}{\frac{1}{2}\rho_a(v_{x,m}^2 + U_m^2)blZr_m} \quad (40)$$

$$C_{T,m} = \frac{\tau}{\frac{1}{2}\rho_a(v_{x,m}^2 + U_m^2)blZr_m} \quad (41)$$

$$\xi_T = \frac{\Omega\tau}{(p_{t,lin} - p_{out})\dot{q}} = \frac{C_{T,m}}{\phi_m C_{A,m}} \quad (42)$$

where $p_{t,lin}$ is the total pressure at the machine entrance, p_{out} is the static pressure at the machine exit, b is the span of the blade, and subscript m indicates the midspan of the rotor blades.

The dimensional parameters are the static-to-static pressure variation of the airflow across the turbomachine and the rotor torque and power.

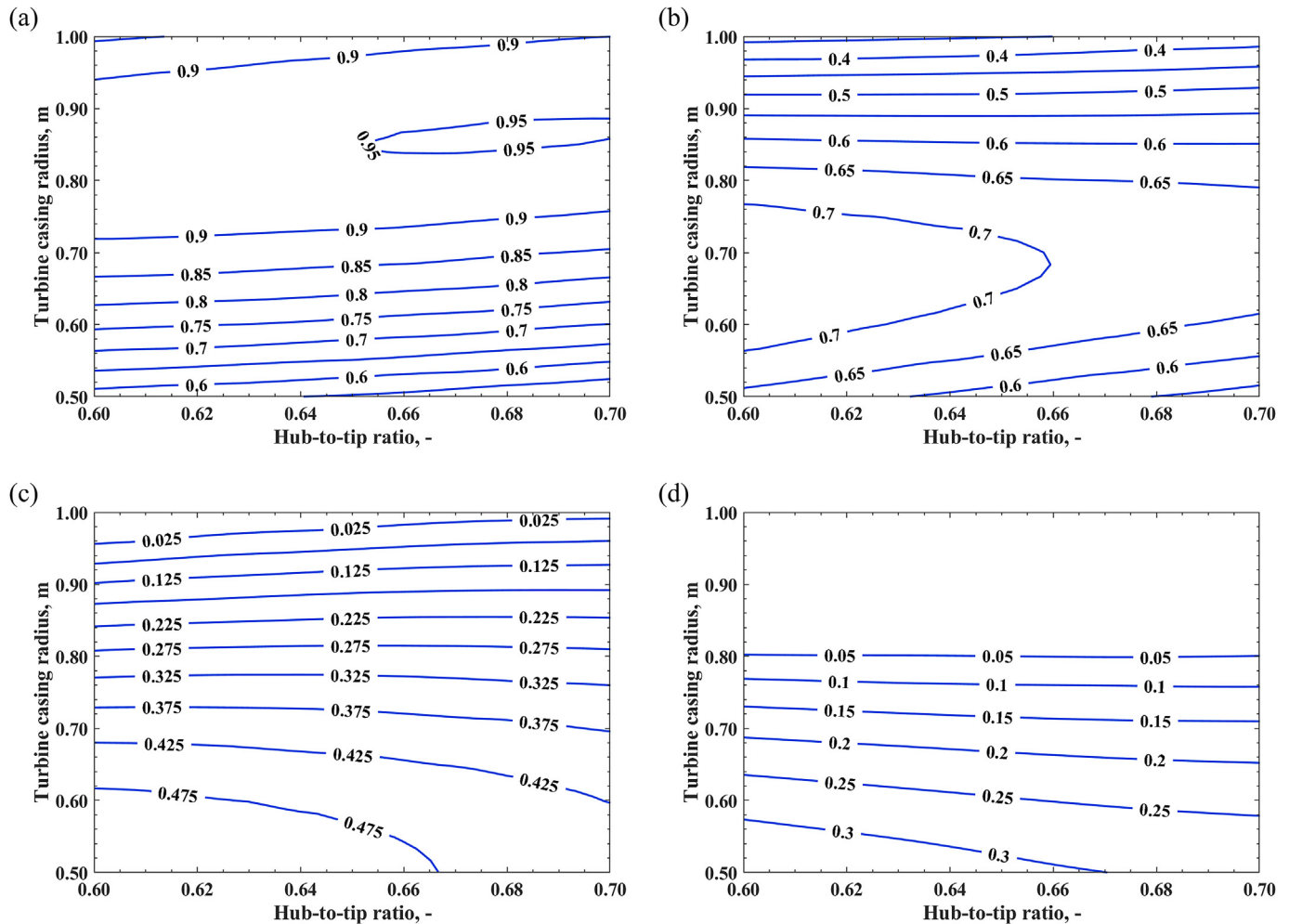


Fig. 6. Energy ratio of the impulse turbine as a function of the hub-to-tip ratio and turbine casing radius for values of rotor angular velocity equal to (a) 300, (b) 600, (c) 900, and (d) 1200 rpm.

The operating curves of the two selected WECs are presented for the wave status that yields the most significant contribution to global energy harvesting every year. Either the systems achieve this condition for the significant wave height of 3.5 m and the wave energy period of 8.5 s.

In Fig. 7(a), the relation between the input coefficient and the flow coefficient is presented. This performance parameter expresses the primary energy necessary for the turbine functioning regarding its geometry and operation. The speed of rise and the maximum values of the input coefficient are lower for the impulse turbine than for the Wells turbine.

Fig. 7(b) indicates that the torque coefficient is negative corresponding to small absolute values of flow incidence angle for both the turbines. This is due to the predominance of the viscous forces over the pressure forces. Increasing the magnitude of the incidence angle, the torque coefficient becomes positive as the latter contribution exceeds the former. The growth of the torque coefficient is faster for the Wells turbine, while the impulse turbine reaches significantly higher peaks. This implies that the torque extracted from the flow relative to the geometric and operating parameters is significantly greater for the impulse turbine.

Fig. 8(a) shows the static-to-static pressure drop between the turbine inlet and outlet as a function of the volume airflow rate. For the Wells turbine, the relation is linear and slightly deviates from linearity only for relatively high absolute values of the flow rate.

Conversely, for the impulse turbine, the curve is quadratic during the whole functioning.

The analysis of Fig. 8(b) indicates that the rotor torque has a negative sign for small absolute values of the flow coefficient, as the viscous forces dominate over the pressure forces. The torque is null if the two opposite actions are equivalent, while it is positive when the impact of the pressure forces prevails. The Wells turbine has a significantly lower torque.

The curves of the rotor power output of Fig. 8(c) denote that the average values and the peaks are higher for the Wells turbine than for the impulse turbine due to the greater rotational speed.

In Fig. 8(d), the efficiency curves rise up to reaching their maximum values and afterwards reduce during either the flow phases for both the machines. Thus, the curves are not monotonous at the extremes of the inflow and outflow. For these specific wave conditions, higher values of efficiency are reached by the Wells turbine.

To better understand the power transformed by the turbine rotor for the two configurations of the OWC system, the time series of the rotor power is shown in Fig. 9 for the significant wave height of 3.5 m and the wave energy period of 8.5 s.

The height of the seawater free surface in the caisson relative to the STL is presented in the time domain in Fig. 10. For the wave conditions of the scatter matrix maximising energy conversion, the water column variation is slightly higher for the system operating

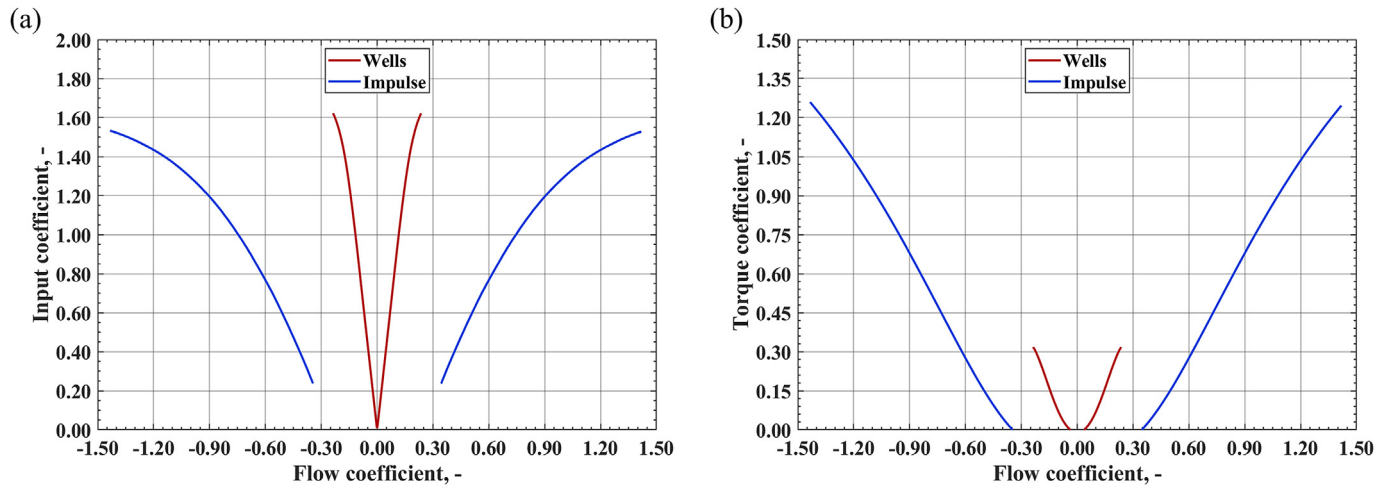


Fig. 7. (a) Input coefficient and (b) torque coefficient versus flow coefficient for the wave status maximising energy harvesting.

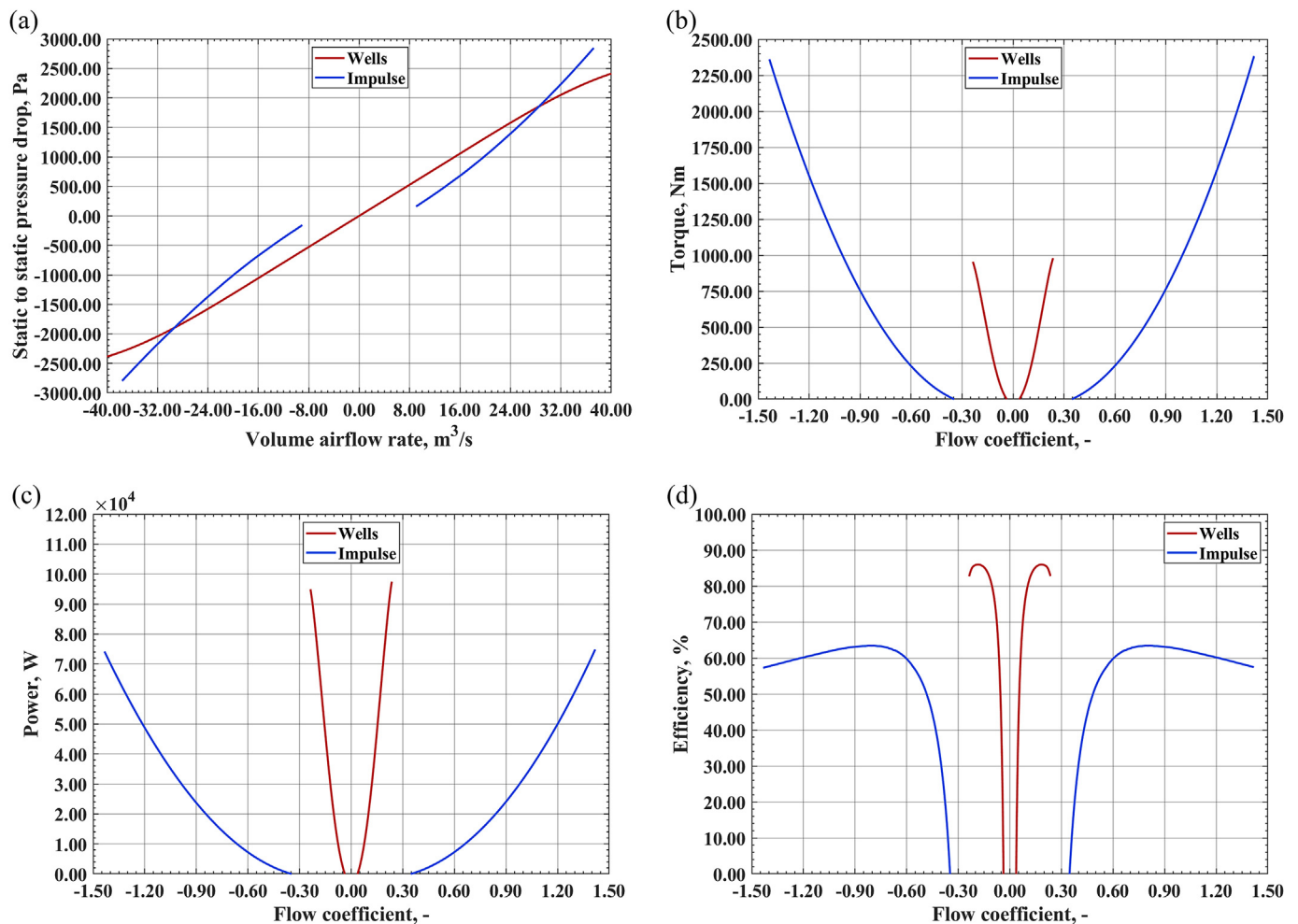


Fig. 8. (a) Static-to-static pressure drop as a function of the volume airflow rate, (b) turbine torque, (c) power, and (d) efficiency versus flow coefficient for the wave status maximising energy harvesting.

with the Wells turbine due to the lower damping imposed on the chamber.

Focusing now on the global results, the two OWC wave energy converters operating with the selected Wells and impulse turbines

harvest 47.20 MWh and 40.97 MWh during one year, respectively. These values were obtained using the annual wave scatter matrices of [Tables 1 and 2](#) for the wave distribution of the selected site located in Sardinia. The control strategy enhances the yearly energy

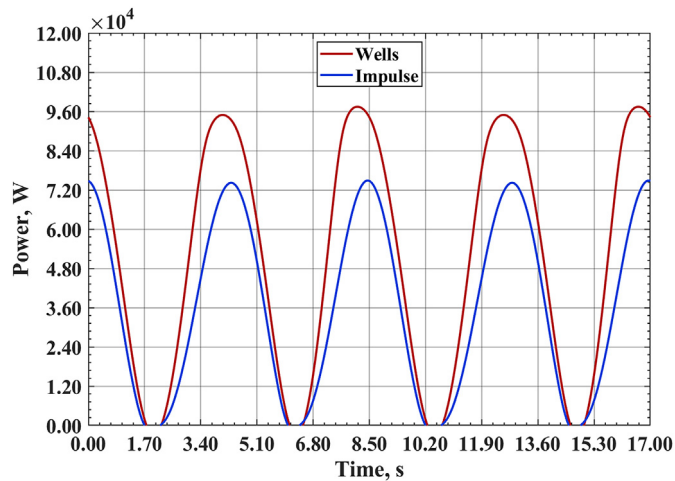


Fig. 9. Time variation of the rotor power for the systems operating with the Wells and impulse turbines for the wave status maximising energy harvesting.

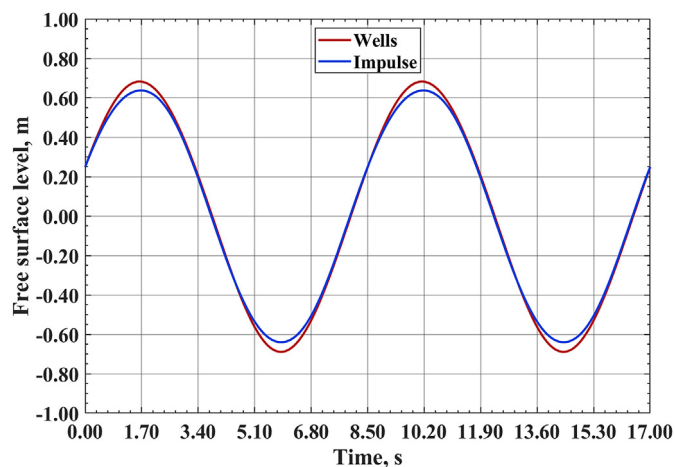


Fig. 10. Time variation of the height of the seawater free surface in the caisson for the systems operating with the Wells and impulse turbines for the wave status maximising energy harvesting.

harvesting of 11.30% and 3.46% for the two configurations, respectively. This marked difference is due to the lower variations of the power output and efficiency of the impulse turbine in off-design conditions regarding the Wells turbine.

The calculated efficiency of power transformation from the resource to the grid is 6.88% for the system operating with the Wells turbine and 5.98% for the device working with the impulse turbine. The efficiencies of the three converters composing the wave energy converters are reported in Table 6.

The efficiency reduces to 5.71% and 4.96%, respectively, regarding the overall energy transported by the sea. Although these values appear to be low, they are unfortunately not too far from the

Table 6
Main values of efficiency calculated for the two selected OWC systems.

Efficiency	Wells configuration	Impulse configuration
Chamber, %	10.61	10.75
Turbine including control, %	68.96	59.06
Organic, %	97.00	97.00
Electric, %	97.00	97.00

present state of the art. In particular, the efficiencies of the secondary conversion stage are in line with the literature. On the other hand, the current system configuration is penalised by the low efficiency of the primary converter. This proves that an integrated simulation method accounting for the mutual interactions, like the one presented in this study, can clearly indicate the components that need to be optimised. To increase the overall efficiency of the wave energy converters, the optimisation of the primary converter will be performed in future for the specific wave conditions of the selected site, jointly with the definition of the best-suited Wells and impulse turbines.

The maps of the annual distributions of energy ratio and efficiency ratio are presented in Figs. 11 and 12, respectively, for the two OWC systems. For each point of the scatter matrix, the annual energy ratio $\Sigma_{i,y}$ was obtained with Eq. (43) by dividing the computed values of energy conversion for the global energy transformed by the system in one year. In analogy, the annual efficiency ratio $\Xi_{i,y}$ is defined in Eq. (44) considering the values of energy conversion efficiency of each wave condition and the overall yearly efficiency of the system.

$$\Sigma_{i,y} = \frac{E_{i,y}}{E_y} \quad (43)$$

$$\Xi_{i,y} = \frac{\xi_{i,y}}{\xi_y} \quad (44)$$

where subscripts i and y are the indices of the rows and columns of the scatter matrices used to indicate a specific wave status.

Fig. 11(a) and (b) indicate that the two systems reach the maximum energy extraction for wave conditions with a significant wave high of around 3.5 m and an energy period of about 8.5 s. The energy conversion markedly decreases for wave conditions outside of the highly energetic range, which has an energy period between 7.5 and 9.5 s and a significant height between 2.5 and 4.5 m. A lower peak of energy extraction is present for either the systems for wave conditions with H_{m0} in the range from 1 to 1.5 m and $T_{m-1,0}$ equal to 5.5 s. The system with the Wells turbine reaches higher maximum values, while the system with the impulse turbine has a smoother operation.

From a perusal of Fig. 12, it is noticeable that either the configurations convert energy with higher efficiency for waves with low values of energy period and significant wave height. This feature is more marked for the impulse turbine than for the Wells turbine.

3.5. Seasonal system performance

The operation of the two wave energy converters was investigated also considering the seasonal wave conditions of the selected application site located in Sardinia.

The energy extracted during each season is compared with the overall annual value through the seasonal energy ratio $\Sigma_{i,s}$ for the two selected systems. For each of them, the parameter is defined in Eq. (45) as the ratio of the seasonal energy harvested for every point of the scatter matrix to the corresponding energy conversion computed over the year.

$$\Sigma_{i,s} = \frac{E_{i,s}}{E_{i,y}} \quad (45)$$

where subscript s relates to the season.

In Figs. 13 and 14, it is noticeable that the distributions of the seasonal energy ratio for the systems operating with the Wells and impulse turbines are strongly dependent on the specific season of

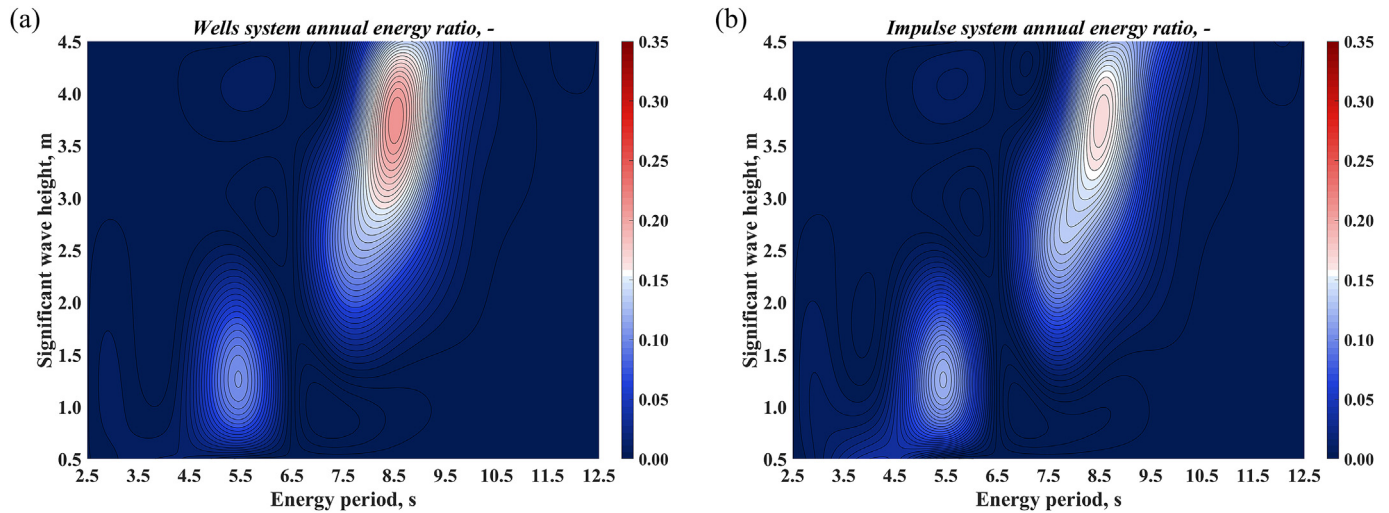


Fig. 11. Maps of annual energy ratio distribution for the two systems operating with the selected (a) Wells turbine and (b) impulse turbine for sets of significant wave height and energy period.

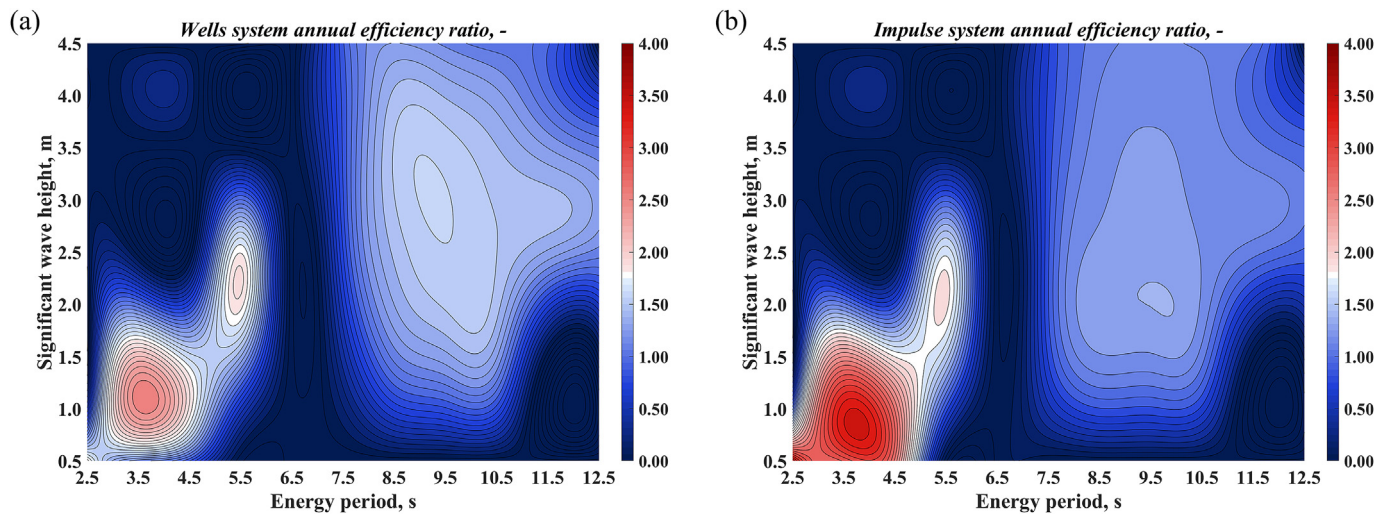


Fig. 12. Maps of annual efficiency ratio distribution for the two systems operating with the selected (a) Wells turbine and (b) impulse turbine for sets of significant wave height and energy period.

the year. For both configurations, the energy harnessing is maximum during winter. The values of energy extraction are considerably lower during summer, while in spring and fall, they are intermediate and comparable among themselves. The reason for these differences is the frequency of occurrence of highly energetic waves with great values of amplitude and period.

Comparing the two systems, the effects of the different wave conditions of the four seasons on the operation are similar. Even on a seasonal basis, the highest peaks of energy extraction are reached by the system operating with the Wells turbine. In contrast the maps of the system working with the impulse turbine have a smoother trend. For each season, the energy extraction performed by the Wells system is higher regarding the impulse system.

With the aim of enhancing the global energy harvested by the

two wave energy converters, the analysis was further detailed by determining the most suited design rotational speed for each season. The values of rotor speed that maximise the overall energy extraction are resumed in Table 7 for both configurations.

The global energy conversion is increased from 47.20 to 47.67 MWh/year for the system operating with the Wells turbine and from 40.97 to 41.14 MWh/year for the device working with the impulse turbine. The corresponding calculated efficiency is 6.95% and 6.00%, respectively, and it reduces to 5.77% and 4.98%, respectively, by considering the global energy conveyed by sea waves.

The efficiency values calculated for the two systems are presented in Fig. 15 for the entire year and all the seasons. The selection of the Wells turbine determines a greater efficiency with higher

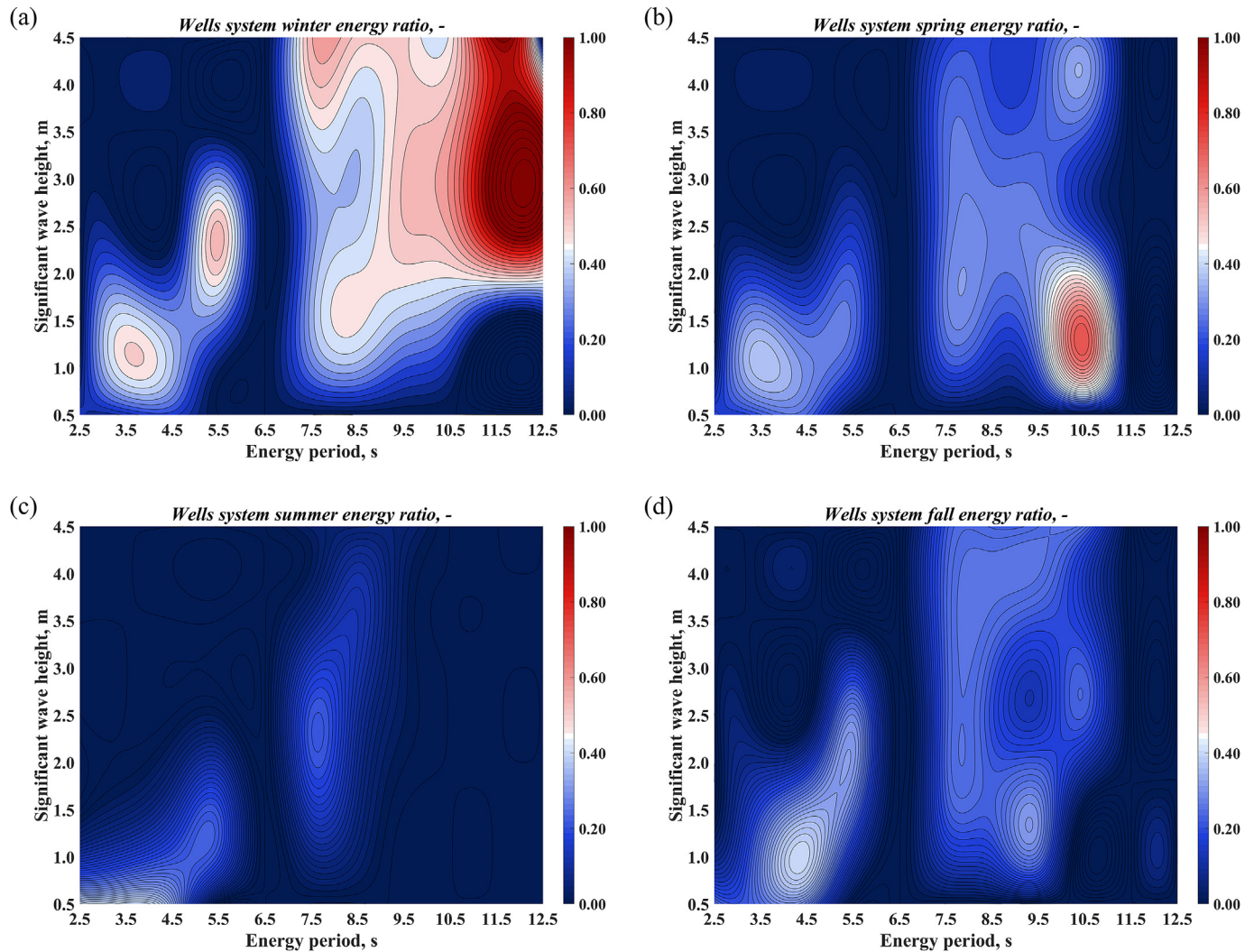


Fig. 13. Seasonal maps of the energy ratio distribution for the system operating with the selected Wells turbine during (a) winter, (b) spring, (c) summer, and (d) fall.

variations during the year regarding the impulse turbine.

4. Conclusions

A complete wave-to-wire model of a wave energy converter relying on the oscillating water column technology was developed. The proposed model applies an analytical approach to pursue a good compromise between accuracy and computational cost. These features make it particularly recommended in the early design stages to evaluate the energy extractable from sea waves and identify the siting and the main geometric and operating parameters of the device.

A rigid piston approach is used to analyse the caisson. Two air turbines were investigated by applying original low-order aerodynamic models, providing an interesting compromise between accuracy and computational cost. In both cases, the rows were modelled as groups of blades, composed of a finite set of airfoils. A regulation algorithm was implemented to enhance the electric energy delivered to the grid by controlling the rotor torque and

angular velocity.

The model was exemplarily applied in a selected location situated in Italy, in the north-west of Sardinia. The geometry and properties of the OWC chamber were determined by scaling up of factor 50 the data obtained from an experimental campaign previously performed under wave conditions typical of a moderate Mediterranean climate. A turbine selection procedure was applied to determine suitable geometric and operating parameters of a Wells and an impulse turbine to maximise the annual energy conversion from waves to wire.

The analysis of the annual and seasonal performance of the system states that the energy harvesting is maximum in winter owing to the considerably higher energetic level regarding the other seasons. The values of energy extraction in spring and fall are similar and intermediate between winter and summer.

The selected Wells and impulse turbines for the operation in the chosen site are capable of harvesting 47.20 MWh/year and 40.97 MWh/year, respectively. The application of a control strategy to regulate the rotor speed and torque enhances the yearly energy

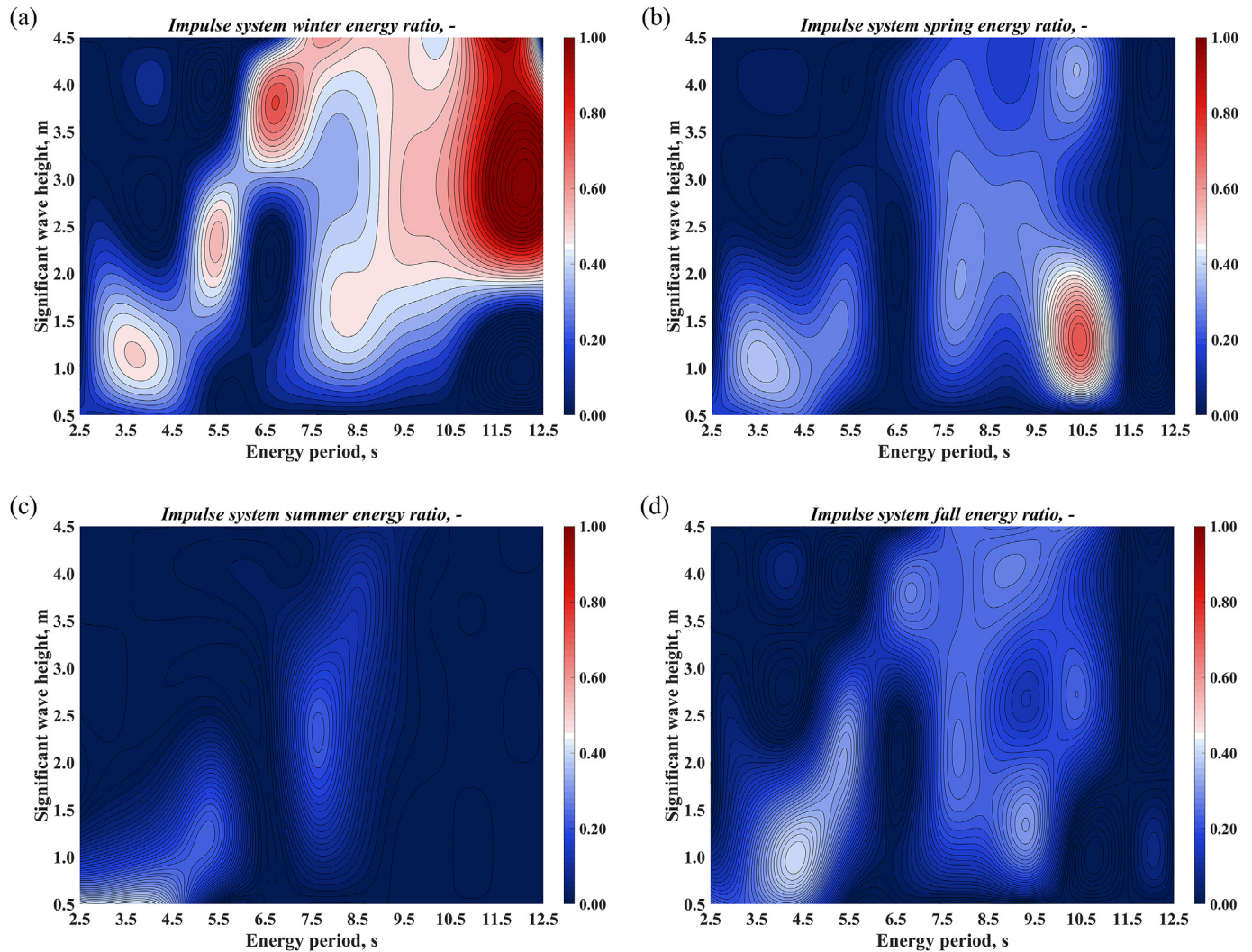


Fig. 14. Seasonal maps of the energy ratio distribution for the system operating with the selected impulse turbine during (a) winter, (b) spring, (c) summer, and (d) fall.

Table 7

Design values of the rotational speed of the Wells and impulse turbines selected on a seasonal basis.

Season	Wells turbine	Impulse turbine
Winter	1000	300
Spring	950	300
Summer	875	275
Fall	950	300

transformation of 11.30% and 3.46%, respectively. For further increasing the global energy conversion, the operation at different values of design rotational speed during the four seasons was considered, obtaining 47.67 MWh/year for the system operating with the Wells turbine and 41.14 MWh/year for the device working with the impulse turbine. The calculated efficiency of the power transformation from the resource to the grid is 6.95% for the configuration with the Wells turbine and 6.00% for the configuration with the impulse turbine. The efficiency decreases to 5.77% and 4.98%, respectively, by considering the global energy transported by the sea.

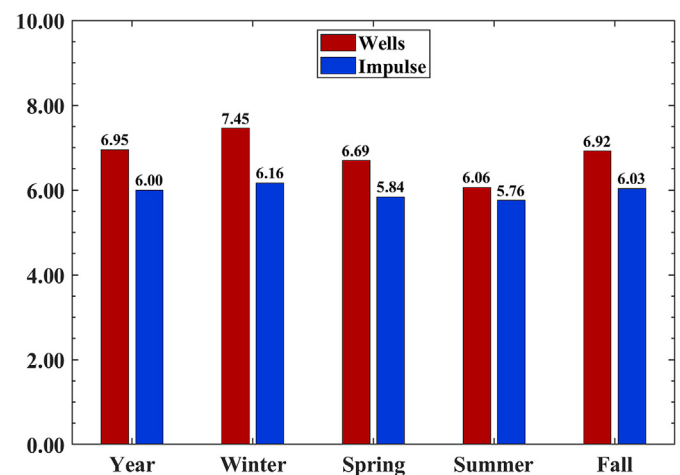


Fig. 15. Values of efficiency of the two selected systems operating with a Wells or an impulse turbine, calculated for the entire year and all the seasons.

Author contributions

Conceptualisation, G.M., Lorenzo Ciappi, A.B., and Lorenzo Ciappi; methodology, G.M., Lorenzo Ciappi, A.B., and Lorenzo Ciappi; software calculations, Lorenzo Ciappi and Lapo Cheli; validation, Lorenzo Ciappi, A.B., Lapo Cheli, I.S., and L.T.; formal analysis, investigation, and resources, all authors; data curation, Lorenzo Ciappi; writing (original draft preparation), Lorenzo Ciappi; writing (review and editing), Lorenzo Ciappi, I.S., and L.T.; visualisation, Lorenzo Ciappi; supervision, G.M., Lorenzo Ciappi, and A.B.; project administration, G.M. and Lorenzo Ciappi. All authors have read and agreed to the published version of the manuscript.

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Lorenzo Ciappi: Conceptualisation, Methodology, Software calculations, Validation, Formal analysis, Investigation, Resources, Data curation, Writing (original draft preparation), Writing (review and editing), Visualisation. Lapo Cheli: Software calculations, Validation, Formal analysis, Investigation, Resources. Irene Simonetti: Validation, Formal analysis, Investigation, Resources, Writing (review and editing). Alessandro Bianchini: Conceptualisation, Meth-

Supervision. Lorenzo Talluri: Validation, Formal analysis, Investigation, Resources, Writing (review and editing). Lorenzo Ciappi: Conceptualisation, Methodology, Formal analysis, Investigation, Resources, Supervision, Project administration. Giampaolo Manfreda: Conceptualisation, Methodology, Formal analysis, Investigation, Resources, Supervision, Project administration.

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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.

Appendix. The scatter matrices of specific wave energy AE and frequency of occurrence rf are provided in the Tables from A.1 to A.8 for the four seasons of the year

Table A.1

Scatter matrix of the winter specific wave energy AE distribution of the wave conditions appraised, for classes of different significant height H_{m0} and energy period $T_{m-1,0}$.

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	185	3935	2959	335	65	0
	3.5	0	0	0	0	178	2440	3958	1600	193	29	4
	2.5	0	0	0	185	1743	3031	2113	657	45	6	6
	1.5	0	4	294	1530	1736	1070	284	12	1	0	0
	0.5	3	98	187	98	22	1	0	0	0	0	0

Table A.2

Scatter matrix of the winter relative frequency distribution of the wave conditions appraised, for classes of different significant height H_{m0} and energy period $T_{m-1,0}$.

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	0.118	2.212	1.488	0.152	0.027	0
	3.5	0	0	0	0	0.216	2.570	3.678	1.331	0.145	0.020	0.002
	2.5	0	0	0	0.521	4.153	6.257	3.848	1.070	0.066	0.007	0.007
	1.5	0	0.049	2.807	11.96	11.48	6.136	1.439	0.056	0.005	0	0
	0.5	0.462	10.83	16.09	6.91	1.336	0.074	0	0	0	0	0

Table A.3

Scatter matrix of the spring specific wave energy AE distribution of the wave conditions appraised, for classes of different significant height H_{m0} and energy period $T_{m-1,0}$.

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	57	1493	1162	208	0	0
	3.5	0	0	0	0	30	1480	2390	727	75	0	0
	2.5	0	0	0	51	1060	2751	1557	361	13	0	0
	1.5	0	3	219	1459	1867	860	140	11	3	0	0
	0.5	4	156	253	144	35	2	1	0	0	0	0

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Table A.4Scatter matrix of the spring relative frequency distribution of the wave conditions appraised, for classes of different significant height H_{m0} and energy period $T_{m-1,0}$.

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	0.036	0.821	0.572	0.093	0	0
	3.5	0	0	0	0	0.036	1.525	2.173	0.591	0.055	0	0
	2.5	0	0	0	0.140	2.470	5.553	2.774	0.575	0.019	0	0
	1.5	0	0.036	2.043	11.16	12.08	4.823	0.695	0.050	0.012	0	0
	0.5	0.675	16.84	21.28	9.911	2.031	0.098	0.026	0.005	0	0	0

Table A.5Scatter matrix of the summer specific wave energy AE distribution of the wave conditions appraised, for classes of different significant height H_{m0} and energy period $T_{m-1,0}$.

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	11	381	271	0	0	0
	3.5	0	0	0	0	6	483	1412	82	0	0	0
	2.5	0	0	0	6	589	2214	583	27	0	0	0
	1.5	0	0	112	1139	1585	462	43	0	0	0	0
	0.5	8	278	320	127	17	2	1	0	0	0	0

Table A.6Scatter matrix of the summer relative frequency distribution of the wave conditions appraised, for classes of different significant height H_{m0} and energy period $T_{m-1,0}$.

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	0.007	0.209	0.133	0	0	0
	3.5	0	0	0	0	0.007	0.497	1.282	0.067	0	0	0
	2.5	0	0	0	0.017	1.370	4.464	1.037	0.043	0	0	0
	1.5	0	0.005	1.050	8.701	10.25	2.589	0.214	0	0	0	0
	0.5	1.225	30.04	26.91	8.763	1.004	0.086	0.024	0.005	0	0	0

Table A.7Scatter matrix of the fall specific wave energy AE distribution of the wave conditions appraised, for classes of different significant height H_{m0} and energy period $T_{m-1,0}$.

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	70	1761	1443	174	0	0
	3.5	0	0	0	0	63	1330	2630	648	56	0	0
	2.5	0	0	0	97	1076	2349	1081	182	17	0	0
	1.5	0	2	298	1422	1641	655	141	10	0	0	0
	0.5	5	177	273	130	25	1	0	0	0	0	0

Table A.8Scatter matrix of the fall relative frequency distribution of the wave conditions appraised, for classes of different significant height H_{m0} and energy period $T_{m-1,0}$.

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	0.044	0.979	0.718	0.078	0	0
	3.5	0	0	0	0	0.076	1.385	2.417	0.533	0.042	0	0
	2.5	0	0	0	0.269	2.535	4.797	1.948	0.294	0.025	0	0
	1.5	0	0.022	2.817	11.00	10.74	3.714	0.706	0.044	0	0	0
	0.5	0.804	19.32	23.19	9.018	1.456	0.071	0.002	0	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_L	Lift 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_c	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^2kg
k	Wavenumber, 1/m
l	Chord length, m

Greek symbols

α	Absolute flow angle
β	Relative flow angle
Γ	Blade loading, N
δ	Flow deviation angle
η	Sea wave displacement, m
Θ	Camber angle
Ξ	Efficiency ratio
ξ	Efficiency
ρ	Density, kg/m^3

Subscripts and superscripts

*	Nominal value
0	Initial condition
1, 2, 3, 4	Turbine sections
a	Air
add	Added
AM	Ainley and Mathieson
c	Chamber
D	Drag
e	Electrical
exc	Excitation
ext	External
HW	Howell
i	Blade element index
l	Row index
id	Ideal
in	Inlet
	Column index

Acronyms

AR	Aspect ratio
CFD	Computational fluid dynamics
CIF	Cascade interference factor
IGV	Inlet guide vane
LCOE	Levelised cost of energy
m	Mass, kg
$\dot{m}$	Mass airflow rate, kg/s
M_{CH}	Flow nominal deviation coefficient, m
M_{HW}	Flow deviation coefficient, $m^{1/2}$
P	Power, W
p	Pressure, Pa
$\dot{q}$	Volume airflow rate, m^3/s
r	Rotor radius, m
rf	Relative frequency, %

s	Sound speed, m/s
t	Time, s
λ	Rotor pitch, m
$\bar{t}$	Tip clearance, m
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
Z	Blade number
z	Vertical coordinate, m
Σ	Energy ratio
σ	Rotor solidity
τ	Torque, Nm
Y	Loss coefficient
$\dot{v}$	Water particle acceleration, m/s^2
ϕ	Turbine flow coefficient,
Ω	Turbine rotational speed, rad/s
ω	Seawater angular frequency, rad/s
m	Blade midspan
o	Organic
out	Outlet
pr	Profile
pto	Power take-off
Rot	Rotor
se	Secondary
L	Lift
sf	Skin friction
T	Turbine
f	Total condition
tc	Tip clearance
te	Trailing edge
un	Undamped natural
w	Seawater
x	Axial direction
θ	Tangential direction
OGV	Outlet guide vane
OWC	Oscillating water column
PTO	Power take-off
SWL	Still water level
WEC	Wave energy converter

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