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New MoonWEC concept and its device optimization

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Abstract—The **MOONWEC, a new concept of point absorber is here presented for the first time after its patent submission. The new concept of WEC merges several working principles, with the aim to benefit from the assets of each single principle. It consists in a hollow floating structure, where the sea water fills a central whole creating a moonpool. The idea is to create a relative motion between the floater and the moonpool's water and then, absorb the energy via a Wells turbine placed in the moonpool. The model considers the structure motion in heave, surge and pitch; the relative displacement of the moonpool, the Wells turbine rotation and its coupling to a permanent magnet generator in order to obtain the absorbed instantaneous power. Three main parts of the device have been simulated: a hollow floating structure, a central moonpool and CALM mooring system. All the parts of the model have been coupled to obtain the general behaviour of the device under the effect of irregular sea states. The paper will present the new concept and the dimensioning and optimisation of the device based on a numerical potential flow model.**

Keywords—**Moonpool, MoonWEC, Potential flow model.**

I. Introduction

In the world of wave energy conversion there is a wide variety of WECs, at different development stages. WECs account with different shapes and sizes; classified as point absorbers, terminators or attenuators. Several working principles such as, oscillating water column, oscillating bodies or overtopping. Devices vary also according to their deployment location, which can be found offshore, near shore or on the shoreline. Hence, one could say that unlike the case of large wind turbines, where design convergence has been met, for wave energy conversion the race is still open. This article presents a new concept of wave energy converter (WEC) specifically designed for Mediterranean wave climates.

A new concept of WEC developed by the university of Bologna [12] is here presented. The aim of this paper is to give a first presentation of the new MoonWEC and preliminary findings on the dynamic of the devise, leaving to the reader more extended references for the model details and accessories modelling (as mooring, turbine etc.).

In the next section, the concept will be introduced and described.

Afterwards, the mathematical model, that describes the device behaviour, the numerical methods and simulation conditions will be presented. Finally, the results obtained from the numerical simulations will be presented and substantial conclusions around them will be drawn.

Further publication will present more in depth the mathematics, the model, the assessment of the expected production at several location in the Mediterranean Sea.

II. Description of the Device

The MoonWEC has been originally inspired by the OXYFLUX ([1], [2]), a downwelling device to artificially oxygenate anoxic waters. This concept cannot fit in any of the classical classification systems ([4]) as such, but at the same it could suit more than one simultaneously. The novelty of MoonWEC is that encompasses simultaneously distinct working principles.

It is composed by two bodies, a floating structure and a power take off (PTO), possibly a Wells turbine. The floater has a hollow cylinder with its axis coincident to the structure's vertical axis. When placed into the water this cylinder is filled creating what is commonly known as a moonpool. When set under the action of waves, not only the structure is excited but so does the moonpool, creating a third virtual body. The moonpool then behaves similarly to a rigid body thus being able to reach the resonant state if well-tuned. The energy conversion is supposed to be carried out by taking advantage of the relative motion between the floating structure and the moonpool. In order to maximize that relative motion, both bodies need to be

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resonating synchronously with completely opposite phases.

The MoonWEC, which is thought to be deployed offshore, is moored to the seabed through a catenary anchor leg mooring system commonly known as CALM system. Catenary systems are specifically suitable for heaving WECs since they mainly block other degrees of freedom while releasing the heave mode free and thus, not interfering with energy conversion motion. Figure (1) shows a cross section of the device. The lower plate is a damping disk which aims to introduce a phase lag between the floating structure and the moonpool oscillatory motions.

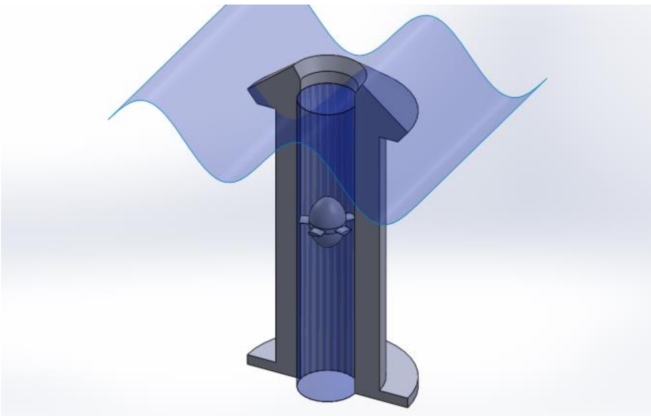

Fig. 1. Sketch of the MoonWEC

Furthermore, the bi-conical upper shape is meant to enhance structure amplitude motion. The still-water level coincides with the basis of both conical shapes. Such shape could also stimulate the overtopping effect acting as ramp for waves to climb up the structure and discharge a certain amount of water within the moonpool. It all depends on the cone's aperture; a closed and steeper cone will decrease the amount of overtopping whereas a flatter and more open cone will have the opposite effect. However, the overtopping effect and its contribution have been left for future investigations as it is of minor relevance if compared to the other working principles.

In synthesis, the MoonWEC can be considered a point absorber, with oscillating body in heave and oscillating water column as major working principles.

III. Potential flow model

A model based on the potential flow theory that follows a Lagrangian approach has been developed mainly using the commercial coding software Matlab™. Figure (2) illustrates the working scheme of the potential flow model. Input data are the environmental conditions as water surface elevation and water particle velocities; the device properties, the mechanical attributes, the mooring system characteristics and, finally, the last type of inputs are the so-called hydrodynamic parameters, the purpose of which is to link the environmental conditions to the body behaviour. An open-source code named NEMOH [5] and developed at the LHEEA laboratory in the Ecole Centrale de Nantes, France, has been chosen to obtain the hydrodynamic parameters.

The model is then able to deliver the device response for each inputted environmental condition, allowing to build the so-called power matrix, the capacity factor, the capture width ratio, the annual production and other performances results. The model is an improvement and adaptation of the potential flow model developed and applied to estimate the energy annual production of a sea bed type point absorber properly designed for the Mediterranean Sea typical wave climates ([6], [7]), and to study the surge effect on the energy production [8].

The MoonWEC has been modelled as a two-body system with four DoFs. The first body is the floating structure of the device; due to its symmetry, the body dynamics have been restricted to a single plane freeing it to move in the surge, heave and pitch modes. The second body is the water entrained in the moonpool orifice, which has been allowed to move freely only along the symmetry axis of the floating structure. Figure (2) presents the schematic of the MoonWEC used to model the device dynamics. It has a fixed coordinate system with its origin placed at the device centre of gravity G, the axis (x; y) define the surge and heave modes respectively and the rotation in pitch is described through ϑ. A moving reference system (ξ; σ) is set to account for the dynamics of the moonpool, being the axes ξ co-linear with the symmetry axes of the MoonWEC.

Following the notation in Figure (2), equation (1) shows the governing expressions for the MoonWEC structure. Despite being a matrix system, for clarity all DoF have been reported separately.

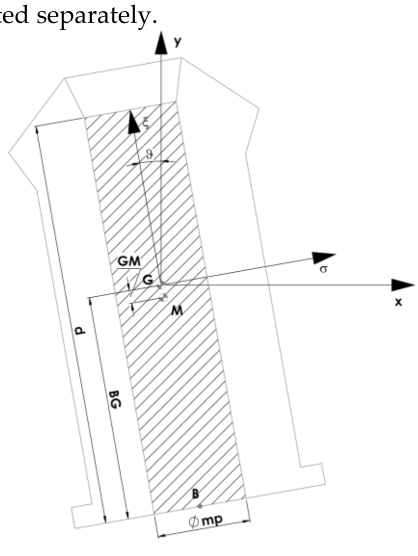

Fig. 2. Layout of the MoonWEC.

$$(m_x + m_x^{\infty})\ddot{x} + (m_{x\theta} + m_{x\theta}^{\infty})\ddot{\theta} = F_e^x(t) + F_r^x(t) + F_r^{x\theta}(t) + \\ + F_d^x(t) + F_{moor}^x(t) + F_{MP}^x(t) \quad (1)$$

(2)

$$\left(m_y + m_y^{\infty}\right)\ddot{y} = F_e^y(t) + F_r^y(t) + F_h^y(t) + F_d^y(t) + F_{moor}^y(t) + +F_{MP}^y(t)$$
$$(m_\theta + m_\theta^{\infty})\ddot{\theta} + (m_{\theta x} + m_{\theta x}^{\infty})\ddot{x} = M_e^x(t) + M_r^\theta(t) + M_r^{\theta x}(t) + +F_h^{\theta x}(t) + M_{moor}^\theta(t) + M_{MP}^\theta(t) \tag{3}$$

$$(m_{MP}(t) + m_{MP}^{\infty})\left(\ddot{\xi} + \ddot{x}\theta + \ddot{y}\right) = F_e^{\xi}(t) + F_d^{\xi}(t) + F_r^{\xi}(t) + +F_e^{\xi}(t) + F_h^{\xi}(t) + F_{MW}^{\xi}(t) \tag{4}$$

Where, m is the mass of the body, m^{∞} is the added mass at ∞ frequency, $m_{MP}(t)$ is the time varying moonpool mass, $F_e(t)$ are the wave excitation forces, $F_r(t)$ are the radiation damping forces, $F_d(t)$ are the viscous drag forces, $F_h(t)$ is the hydrostatic restoring force, $F_{moor}(t)$ are the forces exerted by the mooring system, $F_{MP}(t)$ and $M_{MP}(t)$ are the forces and moment that the moonpool applies to the MoonWEC structure and $F_{MW}(t)$ is the force that the MoonWEC exerts to the moonpool.

The hydrostatic restoring force, based on the Archimedes principle yields the buoyant effect as if it was a linear spring; the wave excitation and radiation forces have been obtained from the frequency coefficients using the Boundary Element Method (BEM), based on the potential flow theory.

The last component of the hydrodynamic forces accounts for the fluid viscous effects. The drag force is formulated as part of the Morison equation. When looking at Figure (1), three main parts can be identified, the upper double-cone, the central cylinder and the bottom disc. Hence drag force acting on each of these parts was calculated. Drag coefficients values are taken from [9].

Modelling the mooring is a difficult part, which was developed properly for the case. A CALM mooring system was modelled based on the classical analytical expression

$$F_{moor} = -T_0 - C_{dl}\dot{x} - m_c\ddot{x}_c \tag{5}$$

where T_0 is the tension at the equilibrium position, C_{dl} is the linearized drag coefficient of the line, m_c is the mass of the line and x_c is the acceleration of the line to be added to matrix system, shown in equations (1).

IV. PRELIMINARY OPTIMIZATION OF THE DEVICE

Tuning and optimization of the device was performed for the three parts of the MoonWEC: the moonpool was dimensioned, then the floating structures has been designed in order to be in resonance and out of phase with the moonpool itself, and finally the anchoring system tuned in order to modify as little as possible the device dynamics for the working conditions while preventing it from drifting away.

A. *Tuning of the device*

Firstly, the floating structure and the moonpool are dimensioned. The fulfilment of several requirements must be assured in order to capture the energy contained in waves in an efficient way. In other words, resonance of the floating structure and the moonpool must be reached. Furthermore, both resonant motions must have phase lag in order to maximise its relative movement.

A floating structure can be simplified as mass spring-damper system. As such, it develops a resonant state if properly excited with a specific frequency, known as the natural frequency of the system, which can be found with:

$$\omega_0 = \sqrt{\frac{k}{m}} \tag{6}$$

Where k is the elasticity constant of the system and m is its total mass. For complex geometries, k and m might not be constants, as the elasticity may vary with hydrostatics and the mass with the added mass due to wave radiation. However, for the heave mode of the moonpool its derivation is immediate [10]; due to its simple cylindrical configuration equation (6) can be simplified to:

$$\omega_0 = \sqrt{\frac{g}{d}} \tag{7}$$

Where g is the gravity acceleration and d is the moonpool draft.

As our intention is to optimize the MoonWEC for relatively short waves, typical of the Tyrrenean Sea, we have selected as peak period of wave Tp around 6s [11]. For this target sea state condition derivation of the Moonpool draft d yield to d=9m.

Having determined on a first attempt the dimensions of the moonpool, the structure wrapping the moonpool has to be sized according to the required natural frequency. In first approach, this is done in the frequency domain, by applying the Fourier transforms to the time domain equation of motion an analytical solution is obtained resulting in no need for time integration. Non-linear effects cannot be modelled in the frequency domain; however, at this stage, this is not a major concern as a general outlook of the device response is the goal of this phase of the designing process.

In the frequency domain, after applying the Fourier transform of the linearized equations of motion (1), the following expression is obtained:

$$F_e = X_0\left(-\omega^2\ (m + A) + i\omega B + KH\right) \tag{8}$$

Where ω is the frequency of the monochromatic wave exciting the structure, m is the mass matrix of the system, A is the added mass matrix, B is the radiation damping matrix, *KH* is the hydrostatic stiffness matrix, F_e is the excitation force coefficient vector and X_0 is the Response Amplitude Operator (RAO), which reflects the unitary response of the system. Except the RAO, which is the unknown variable, all the other coefficients are known; *m* and *KH* are solved using the internal forces equations and the hydrodynamic coefficients A, B and Fe are obtained with the BEM method. A preliminary mesh convergence analysis was performed, in order to obtain the minimum number of mesh element in order to obtain an accurate solution.

Since the modelled structure is the same, the element size influences only the total number of elements of the mesh. Hence, a mesh with a smaller element size will have a larger number of elements. For the heave mode, peak location convergence is found at 500 elements; peak amplitude convergence is observed after 700 elements. The same trend is identified for the pitch mode where both, peak location and amplitude convergence is reach for 600 elements. For all the stated above, being 700 the number of elements assuring convergence for both modes, it has been selected as the minimum number of elements for the modelling of the structure with the open-source code NEMOH.

Once the minimum number of elements has been found, a recursive analysis on the MoonWEC structure is carried out. The goal is to obtain the resonant state for an excitation period of T=6 s which yields an angular frequency of ω = 1.05 rad/s. As the draft of the structure is fixed by the moonpool constraint, the shape and diameter of the structure are the only variables left to modify. The structure has a cone shape on top in order to guarantee a smooth transition in the free surface region. Also, a damping plate has been installed at the bottom to tune the phase of the device; this effect, however, cannot be modelled in the frequency domain as the drag introduced by the plate is non-linear. Therefore, the diameter of the of the body is the parameter upon which the RAO sensitivity analysis has been executed. Table (1) reports the resonant response peak frequency in heave of several structures with different diameters.

TABLE I
HEAVE RESONANCE PEAK LOCATION FOR SEVERAL STRUCTURES WITH DIFFERENT MAIN BODY DIAMETERS

D (m)	ω_0 *rad/s*
3	1.32
4	1.22
5	1.08
6	0.9
7	0.8
8	0.66

As the diameter increases so does the mass of the structure and thus, the natural frequency of the system decreases. The obvious choice according to the target frequency is the structure with a main body diameter of 5 m since, its natural frequency ω_0 = 1.08 rad/s, close to the target value of *Tp=6 s*.

Finally, the hydrodynamic parameters of the selected structure are computed, then if substituted to equation (8) allow to obtain the response of all 6 DoFs of the device.

Since this study is carried in the plane (x; y), the modes of sway, roll, and yaw have been neglected leaving only the surge, heave and pitch modes. Initially, in order to take into account, the effect that the moonpool has on the floating structure, its inertia has been added to the floating structure's only in the surge and pitch DoFs. The heave mode has been left unaltered since the moonpool is free to move within the structure in the vertical direction. Fig. 3 and Fig.4 show the hydrodynamic coefficients and the RAOs of the MoonWEC, respectively.

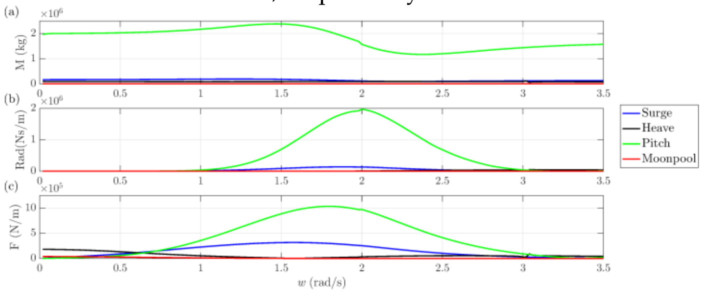

Fig. 3. Hydrodynamic coefficients vs. frequency obtained with the BEM approach for the different modelled DoFs. a) added mass coefficients, b) radiation damping coefficients and c) excitation force coefficients.

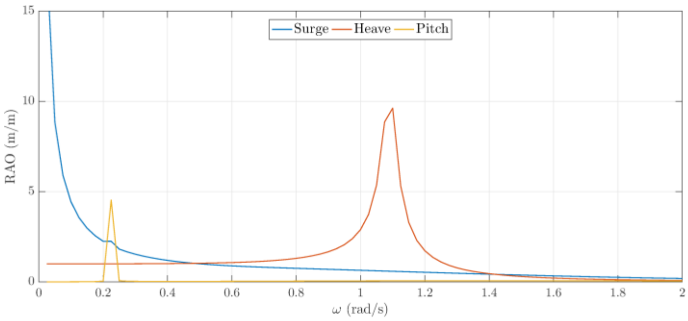

Fig. 4. Response amplitude operators of the MoonWEC for the studied DoFs.

A very differentiated peak can be distinguished for every mode in figure (4). The resonance peak in heave mode coincides to the one highlighted in table (1), having an amplitude of ten metres for wave height meter. The pitch natural frequency is located at 0.2 rad/s, a relatively low frequency that suits the designing requirements; since at the selected sites, waves matching those frequencies are highly improbable. Unlike in the heave case; the resonance in pitch is an undesired effect which should be avoided.

Not only for the device survivability, the more it swings the less stable the whole WEC behaviour will be, including its production; also, it should also be avoided for numerical reasons.

During the model derivation (see chapter 4), the hypothesis of small angle has been used recursively for two main purposes: computational cost reduction and simplicity of equation derivation. The WEC behaviour in pitch will be a key aspect on the mooring system designing; whose goal will be, among others, to minimize the rotation in pitch.

The surge mode shows an asymptotic resonant behaviour towards frequency 0. This is explained because no mooring system has been inserted in the model at this stage and therefore, in the absence of stiffness in the horizontal direction, the device tends to resonate at frequency zero, backing equation's (6) prediction.

The next step after concluding the frequency domain analysis is to switch to time domain in order to consider the non-linear effects.

In the time domain, the determination of the natural frequency of the system for each DoF is achieved through the performance of the decay tests. These tests consist in giving the system an initial state different than that of the equilibrium. Under no external influence, the system will tend to go back to the equilibrium state. This response, measured over the time, gives the information about the natural period of the system and its damping.

Before performing the tests, a little numerical tuning is needed. In order to compute the radiation effect, the added mass at infinite frequency and the Impulse Response Function (IRF) need to be calculated from the frequency dependent hydrodynamic coefficients (Fig. 3). Subsequently, the IRF is approximated with the Prony's method. Details on the use of the methods and on its hypotheses are deeply described in [12]. Figures (5-7) show the computed IRFs and their approximation with the Prony's method for the surge, heave, pitch, moonpool respectively.

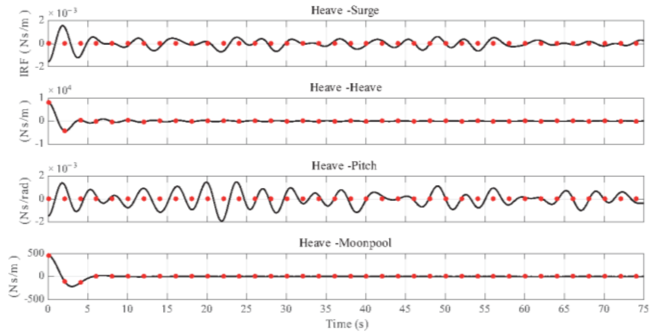

Fig. 5. Radiation impulse response functions for the heave mode. Analytical curve (solid line) and approximation with Prony's approach (dots).

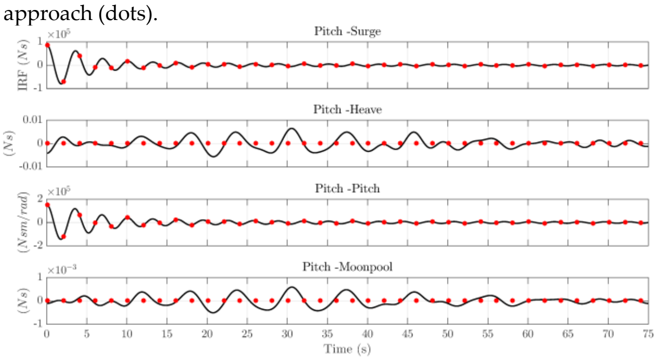

Fig. 6. Radiation impulse response functions for the pitch mode. Analytical curve (solid line) and approximation with Prony's approach (dots).

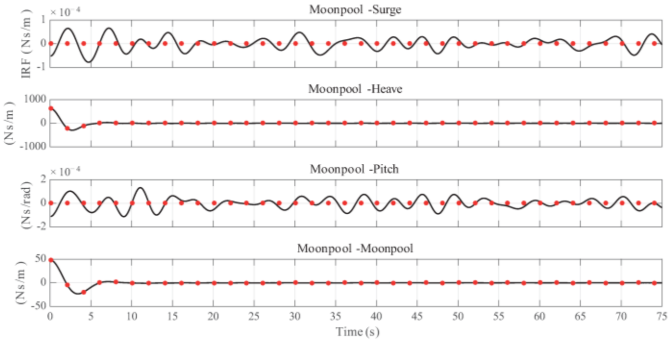

Fig. 7. Radiation impulse response functions for the moonpool DoF. Analytical curve (solid line) and approximation with Prony's approach (dots).

Figures (5-7) show a good agreement between the actual curve a its approximation. Not all the radiation IRF have been modelled; some crossed-modes have really reduced impact on the general structure behaviour since they are up to 8 orders of magnitude smaller than their counterparts and therefore, they have been neglected. No relevant interaction is found between surge-heave modes, heave-pitch modes, moonpool-surge modes and moonpool-pitch, which makes a total of 8 neglected interactions, this number times 6 times 2 results in 96 dimensions saved in the numerical solver.

Having tuned the numerical model, the time domain model was used for the decay tests. The dynamic response over time of the system is shown in Fig.8.

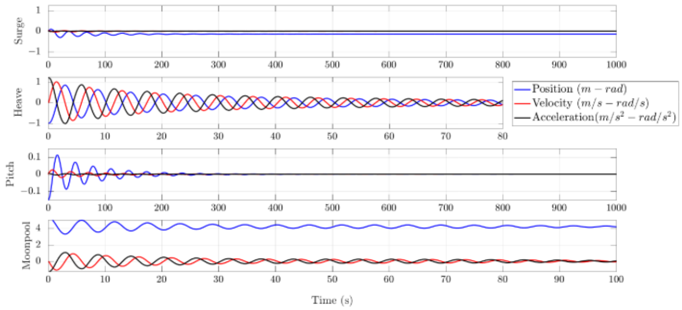

Fig. 8: Decay test results time series for the studied modes.

No anomalies are observed in figure (8), the heave and pitch modes show symmetrical oscillations damped at different rates. The moonpool has an offset of the mean oscillatory position. This shift has been introduced to express that the results for the moonpool are obtained at its free water surface, as opposed to the results of the floating structure which are computed at its centre of gravity. The observed moonpool dynamics are the same of the heave mode but in reversed direction. This makes sense as the computed coordinate of the model is the relative motion between the structure and the water in the moonpool. Hence, having no external input the water inside the moonpool remains practically still and thus, moving almost oppositely to the floating structure. The surge mode shows small oscillations, which are the result of the cross-radiation effect surge-pitch (see figure 6). Finally, it assumes a resting position different than zero confirming the lack of a mooring system.

Decay tests are very useful for the natural period determination. However, the extent of the system response can't be fully assessed with it due to the lack of forcing input and information on the phase lag between the moonpool and the floating structure. To such purpose the forced oscillation tests are carried out by bringing the device under the action of a monochromatic wave of H = 0,5 m and T = 6 s.

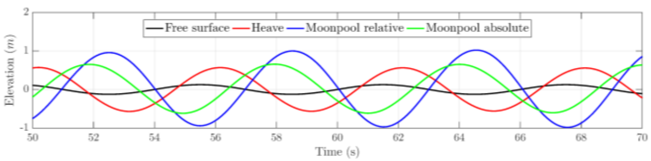

Fig. 9: Forced motion test results time series for the studied modes.

In Fig. 9 the heave displacement and the absolute and relative (to the structure) moonpool displacements are plotted. After performing the phase analysis of the curves with respect to the free surface; the delays found for the structure, the relative displacements between the structure and the relative motion of the moonpool is approximately 148°, 75% larger with respect to the absolute structure and moonpool amplitudes.

With the moonpool's relative velocity, its captured mechanical power can be computed as shown in equation (6):

$$P_{MP}(t) = M_{MP}(t)\dot{\xi}(t) \qquad (6)$$

Where $M_{MP}(t)$ is the moonpool water mass, variable in time due to its level oscillations and $\dot{\xi}(t)$ is the relative moonpool velocity. This is known as the net moonpool power since it is the input power for the PTO, whose aim is to convert it into electrical power. Averaging $P_{MP}(t)$ over 1000 thousand waves for each sea state yields the net power matrix. The maximum value of the net power matrix states the MoonWEC net rated power, which is of 18 kW.

Provided that wave climates are known for certain locations, with the power matrix one can estimate the performance indicators for that location. Annual Energy Production (AEP), the mean power output (P), the Averaged Capture Width Ratio (CWR) and the Capicity Factor (CF). Table (2) reports such indicators at Alghero, in Sardinia and Mazara del Vallo, in Sicily.

TABLE 2
MOONWEC'S PERFORMANCE INDICATORS

Location	AEP (MWh/y)	P (kW)	CWR (%)	CF (%)
Alghero	36	4,1	41,9	22,8
Mazara del Vallo	33,5	3,8	44,3	21,8

V. DISCUSSION

The new device for wave energy conversion MoonWEC has been here presented, and the implementation of the numerical model developed in order to describe its dynamic behaviour has been discussed. Preliminary optimisation of the device dimension has been performed, the optimal dimension for the MoonWEC installed in short wave conditions (approx. Tp=6s) typical of closed seas like the Mediterranean Sea, is estimated. In particular, the diameter resulted to be equal to 5m. Performance indicators give reasonable outcome to think that the MoonWEC could be feasible if installed in array.

Discussion on and the CALM mooring system and the turbine choice, significant aspects of the development, will be object to other further publications.

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