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A submerged point absorber wave energy converter for the Mediterranean Sea

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Abstract— **Regarding the serious environmental and energy problem existing in the world, renewable technology has taken the global attention. The unexploited huge potential of wave energy challenges the world scientific community, to achieve a sustainable wave energy absorption. Mediterranean is a closed sea with lower energy potential compared to the oceans. However, an efficient WEC that maximizes the power output not with respect to the available power but to its characteristics and cost, could be a feasible solution. A preliminary assessment has been carried out of a submerged point absorber installed in the coast of the Pantelleria island. For the design and selection of the technical characteristics of the WEC a method proposed by Falnes is followed. In order to control and test the performance of the device in different situations, a mathematical model has been constructed in Matlab Simulink based on the Cummins Equation.**

Keywords—**Submerged point absorber; Wave energy device, island of Pantelleria**

I. Introduction

Taking into account the hazard environmental situation existing in the planet, renewable energy technology has become notable. One of the most promising natural green energy source is wave energy with an impressive potential of the order of 1 – 10 TW [1], capable to cover the global energy needs. However, at the time of writing, wave energy technology remains still enough immature, and as a result, ocean wave energy is not competitive in the global energy market.

Mediterranean is a closed sea with lower energy potential compared to Pacific, Atlantic and Indian Oceans. Generally, Mediterranean is considered an intermediate level between the open seas and the enclosed small-fetch basins such as Black Sea or Baltic Sea. The most energetic area in the Mediterranean Sea is the Western basin in between the Balearic Islands, Sardinia and Corsica and the Northern coast of Algeria with a yearly available mean wave power of about 10kW/m along the coast. Central and Eastern Mediterranean present average energy potential, around 6-7 kW/m [2].

Besides the fact that in closed seas the available energy is lower compared to the oceans, a sustainable wave energy conversion is possible. Falnes et.al [3] suggests that a wave energy converter and specifically a point absorber should be designed in such a way that cost of energy production minimizes. Some common point absorbers are: Ceto 6, a fully submerged ocean technology developed by the Australian Carnegie Wave Enegry Limited [4], Powerbuoy developed by Ocean Power Technologies [5], Wavebob developed by Wavebob Ltd [6], and ISWEC developed by Politecnico di Torino and Wave for Energy Srl [7], [8].

In order to achieve a cost efficient point absorber Falnes [3] proposes to select the volume and the capacity of the PTO (Power Take Off) system taking into account the targeting sea state. More specifically, it is claimed that a wave energy converter should be designed to absorb effectively only the one third of the year avoiding extra costs coming from over dimensioning the device.

Moreover, as it was proved in [9], a WEC is able to absorb more energy if they do exist more than one mode of oscillation. Also according to [10], the ability of a hull to remove energy from the waves depends on its capability to radiate waves. Hence, a point absorber should be both source and dipole wave radiator in order to maximize extraction of wave energy.

As it was presented in [9], a meaningful propose is to deploy the hull of the point absorber under the surface of the sea for two main reasons:

a) Survivability: One of the biggest challenges in the designing of wave energy converters is to ensure the security of the WEC in extreme sea conditions.

b) Social reasons: Besides the maturity of

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technology, the opinion of the local society in order to install an array of WECs is crucial. Submerged point absorbers are not visible, a fact that contributes in the evolution of wave energy market.

For the reasons mentioned above a spherical submerged point absorber is studied, able to extract power from surge, heave and pitch, suitable for the wave conditions of the Mediterranean Sea and specifically in the island of Pantelleria. The necessary numerical tool used for the analysis and the procedure of the design process is presented below.

II. Mathematical Model

A. *Numerical model*

The mathematical model considers only a 2-dimensional motion, because the hull is solid of revolution, and as a result, the response of the device will be the same, no matter the directionality of the incoming wave.

The model has been built in Matlab Simulink and it is able to simulate the dynamic response of the wave energy converter taking into account the excitation forces of the incident wave, the radiation and viscous damping phenomena and linear mooring dynamic effects etc. A linear PTO damping has been added in order to calculate the power generation and also estimate the PTO Forces applied on the hull. The model uses the solver ode45 and the time step is 0.01 seconds. The time of the simulation varies and depends on the transitory period of every situation. All the necessary hydrodynamic coefficients necessary for the execution of the time domain simulations, are obtained from the commercial software Ansys Aqwa and they are inserted in the model.

Cummins [11] on 1962, presented the well-known *'Cummins Equation'* which describes the motion of an oscillating ship under the effect of an incident wave in the sea. The equation written in frequency domain is

$$[M + A(\omega)]\ddot{X} + B(\omega)\dot{X} + KX = F_W(j\omega) \quad (1)$$

The previous equation is based on linear theory, and as a result is valid for small amplitude motions. In the expression above, M represents the mass matrix of the hull, $A(\omega)$ the added mass matrix, $B(\omega)$ the potential damping matrix, K the linear hydrostatic stiffness (due to the buoyancy forces), $Fw(j\omega)$ the wave forces vector. The wave force is expressed as the product of the half of the wave height $H/2$ multiplied by the wave force coefficients per unit of wave amplitude $fw(j\omega)$ also called Froude-Krylov coefficients. The 6 DOF frequency-domain equation was converted into a time-domain dynamic equation by Ogilvie [12]. According to the Ogilvie's decomposition, equation (1) becomes

$$[M + A(\infty)]\ddot{X} + \int_0^t h_r(t-\tau)\dot{X}d\tau + KX = F_e(t) \quad (2)$$

where $A(\infty)$ represents the added mass matrix evaluated for infinite oscillation frequency, $F_e(t)$ express the wave loads, while $h_r(t-\tau)$ is the impulse response function of the radiation forces. Finally, the complete mathematical model of the submerged point absorber is

$$[M + A(\infty)]\ddot{X} + \int_0^t h_r(t-\tau)\dot{X}d\tau + F_m = F_e(t) + F_D + F_H + F_{PTO} \quad (3)$$

where F_m represents the mooring forces, F_D express the viscous non-linear drag forces, F_H stands for the permanent hydrostatic force and F_{PTO} is the force created by the PTO function. Excitation force time series $F_e(t)$ can be calculated for a monochromatic wave in a single frequence, and for a polichromatic wave using wave spectra models. The mooring system is modelled as linear with constant linear stiffness.

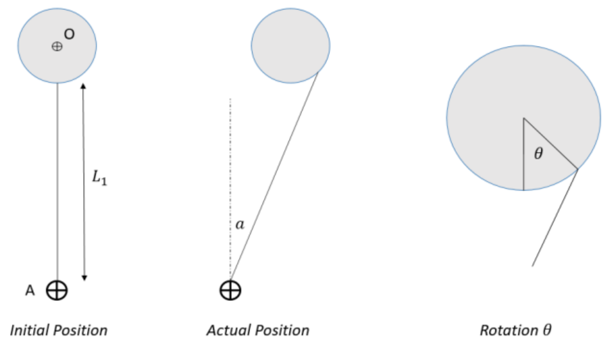

Fig. 1. Modelling of the motion of the hull.

In Fig. 1 the two-dimension planar modelled motion of the hull is presented. L_1 represents the initial length of the mooring line, α expresses the angle between the mooring line and the perpendicular axis passing from the anchor, θ stands for the rotation of the hull. We consider two different axes of reference: one in the COG of the sphere in the initial position O (x, z, θ) and one in the anchor of the mooring system A (X, Z, θ). The motion of the sphere is an input in mooring numerical section expressed in the system reference O. In this way the position of the connection point between the mooring line and the device is calculated

$$X_A = x_O + R * sin\theta \quad (4)$$

$$Z_A = L_1 + z_O + R * (1 - cos\theta) \quad (5)$$

In this way the actual length of the mooring line is calculated

$$L = \sqrt{X_A + Z_A} \quad (6)$$

Consequently, the elongation is

$$\Delta L = L - L_1 \quad (7)$$

And the tension

$$T = K_m * \Delta L \tag{8}$$

where T represents the tension of the mooring and K_m the linear stiffness. For the determination of the dynamic interaction between the mooring and the hull the following quantities are calculated considering the angle α presented in Fig. 1.

$$sina = \frac{x_A}{L} \tag{9}$$

and

$$cosa = \frac{z_A}{L} \tag{10}$$

These trigonometric number are used in order to calculate the final mooring forces applied in the hull

$$\text{Mooring force X} = T * sina \tag{11}$$

$$\text{Mooring force Z} = T * cosa \tag{12}$$

$$\text{Mooring moment } \theta = Force\ X * R * cos\theta + Force\ Z * R * sin\theta \tag{13}$$

In order to calculate the PTO forces of the device the velocity of the mooring is considered.

$$\text{PTO Force X} = Damping * Velocity * sina \tag{14}$$

$$\text{PTO Force Z} = Damping * Velocity * cosa \tag{15}$$

$$\text{PTO moment } \theta = PTO\ force\ x * R * cos\theta + PTO\ force\ z * R * sin\theta \tag{16}$$

In order to calculate the mean power

$$P = \frac{1}{2} * Bpto * |v|^2 \tag{17}$$

where v is the velocity of the mooring elongation. For what concerns the nonlinear viscous damping, the specific submerged spherical point absorber is an easier occasion to be handled. For simple geometries such as the sphere the well-known drag force can be calculated by the formula [13]

$$F_{dx} = -\frac{1}{2}\rho C_{Dx} A_x |\dot{x}| \dot{x} \tag{18}$$

$$F_{dz} = -\frac{1}{2}\rho C_{Dz} A_z |\dot{z}| \dot{z} \tag{19}$$

$$F_{d\theta} = -\frac{1}{2}\rho C_{D\theta} D^5 |\dot{\theta}| \dot{\theta} \tag{20}$$

The variable ρ represents the density of the water, C_D represents the dimensionless drag coefficient in a specific mode, Ax and Az are the cross-section areas of the buoy along surge and heave axes, D is the diameter $\dot{x}$,$\dot{z}$ and $\dot{\theta}$ are the velocity of the hull in surge and heave respectively.

B. *Evaluation of the model*

A way to validate a mathematical model is to make comparison with a known commercial and reliable software. Ansys Aqwa [14] is considered to be one of them and, as a result, it will be used for this objective. The target is to compare the motion in heave, in surge and the PTO Forces of the point absorber. The analysis has been done using regular waves of 1-meter height, periods from 5 seconds till 10 seconds and 30 m of sea depth. The characteristics of the selected hull are presented in table 1.

TABLE 1
SELECTED HULL FOR THE EVALUATION OF THE MODEL

Symbol	Quantity	Unit
m	Mass	66088 kg
R	Radius	2.58 m
V	Volume	71.64 m^3
ds	Submergence of COG	4.53 M
k	Stiffness	600000 N/m
Bpto	PTO Damping	100000 N*s/m
L_1	Mooring initial length	22.89 m

It is underlined that in order to insert external forces such as the PTO forces during the simulation the option *call routine user forces* must be selected. A dynamic link library (dll file) compiled in Fortran or C++ should be used in the time domain analysis to include forces that cannot be modelled in the standard menu of Ansys AQWA [15].

The quantity which is compared is the height of the dominant wave taken out from the analysis Fourier in the time history of the surge, heave and PTO forces. In the Fig. 2 the results of the comparison between the model and Ansys Aqwa is presented.

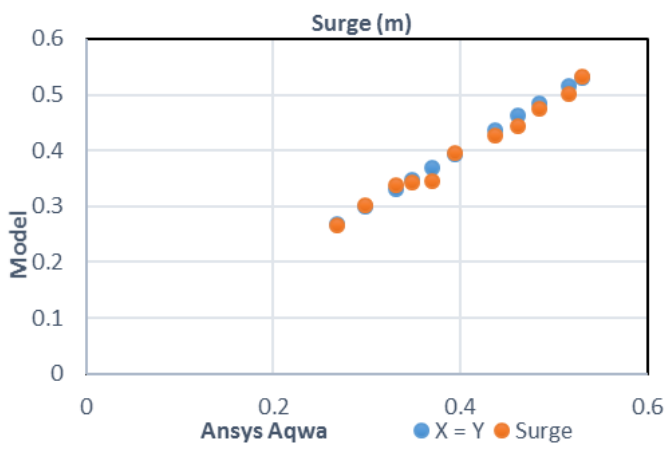

Fig. 2. Surge comparison results.

As it is demonstrated in the Fig. 2, 3, 4, the results confirm that the time histories have the same characteristics. The correspondence between the two methods in surge, heave and PTO forces are between 94% and 99%. The precise procedure for the selection of the characteristics of the device and the design process of the point absorber is presented in the following section.

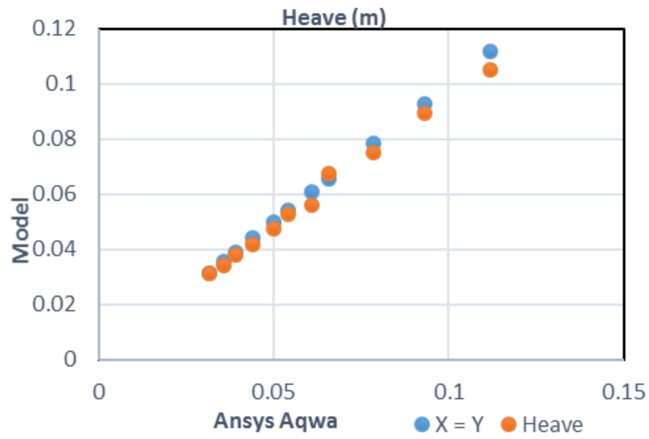

Fig. 3. Heave comparison results.

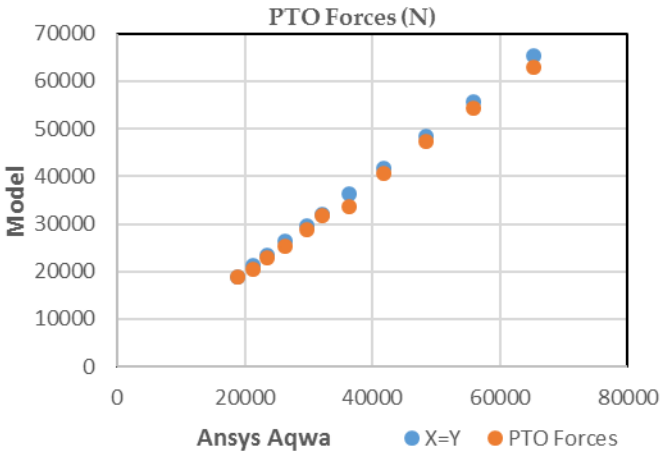

Fig. 4. PTO forces comparison results.

III. Design Process

C. Theoretical Method

In order to design successfully a wave energy converter, the understanding of the available resources and the amount of energy that could be realistically converted into electrical energy is a crucial factor. According to Falnes [3], one of the reason why wave energy conversion remains still a dream, is the fact that the initially design of the WEC must take into account the realistic limits of the wave absorption in order develop efficiently a technology which will be able to provide clean zero-emission energy, and in the same time, will be economical feasible, trustful, safe and ready to enter in the energy market to become competitive. As it was proved [3] [9] [10], two theoretical limits do exist for the wave power absorption: a) High frequency limit b) Low frequency limit.

According to [10] the high frequency limit *Pa* depends on the hull's ability to destroy waves. When a body receives the excitation forces of the passing wave, starts to oscillate and consequently, it produces waves known as radiated waves. A maximum power generation can be achieved from a point absorber which behaves like both source and dipole radiator.

In other words, a point absorber should be free to move in three degrees of freedom (surge, heave, pitch) and steal energy from all of them. The high frequency limit is presented below assuming deep water conditions ($\omega^2 = kg$)[9].

$$P_a = a\frac{J}{k} = a\frac{\rho g^2 A^2}{4\omega k} = a\frac{\rho g^3 (H/2)^2}{4\omega^3} = c_\infty T^3 H^2 \quad (21)$$

where $J = \rho g^2 D(kh) A^2/(4\omega)$ is the wave energy transport per unit frontage of the interfacing wave, α is a coefficient depending on the vibration mode ($\alpha = 1$ for heave, $a = 2$ for surge or pitch, $a = 3$ when the hull moves in heave, surge and pitch), k is the wave number, A is the wave amplitude, ρ is the water density, ω is the wave frequency and D(kh) is the depth function which is equal to 1 for deep water assumption, $c_\infty = \rho(\frac{g}{\pi})^3/128$ [9].

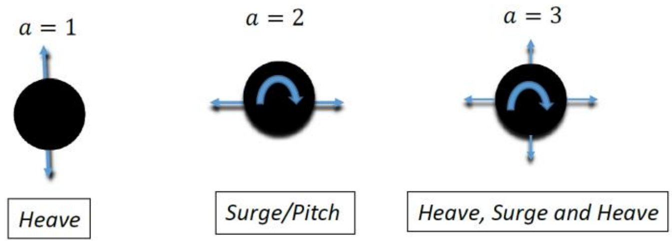

Fig. 5. Coefficient α or different occasions of the oscillation.

The first bound corresponds to the maximum amount of energy that can be removed from an incident wave, however it considers optimum conditions and unreasonable motions from the body without logical displacement bounds. The low frequency limit P_B indicates how a realistically sized immersed or submerged point absorber has maximum border for what concerns the power absorption [3]. The second limit for a submerged point absorber [16] is

$$P_b = 4\pi^3 \rho e^{-kd_s} s_{3,max}\, V_s H/T^3 \quad (22)$$

where V_s is the maximum swept volume of the wave energy converter, H is the height of the incident wave, T is the period of the incident wave, e^{-kd_s} expresses the influence of the submergence of the body in the sea and $s_{3,max}$ stands for the maximum oscillation amplitude in heave . As it is shown, P_b are inversely proportional, a contrary situation with the high frequency limit.

Attention must be paid in the fact that equation (22) is valid only for a floating one degree of freedom (heave) point absorber. Concerning the multiple degrees of freedom, a submerged body which oscillates in heave and in surge is bounded by a low frequency limit P_b which can be twice as (22), because of the bigger swept volume and velocities. In Fig. 6 a schematic representation of the swept volumes in surge and heave is presented. The combination of these two situations result the higher P_b limit [9].

An interesting issue is the relation between the swept volume and the physical volume of the device. Falnes [3] claims that the swept volume of a body cannot be higher

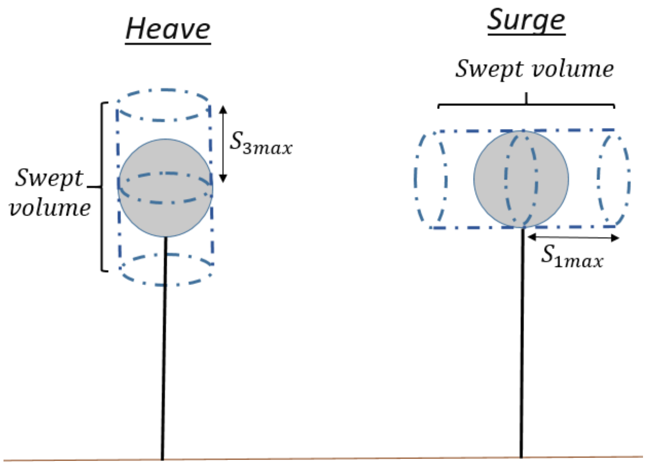

Fig. 6. Swept volume in heave and surge.

than its physical volume. The best case scenario would be a point absorber which achieves a swept volume equal to the volume of the hull, and consequently the relative cost in the energy production of the system decreases. The control of the PTO plays an important role in the efficiency of the wave energy conversion, as it is responsible to maintain the system in resonance. Many different control strategies for various wave energy converters have been developed so far [17]. Reactive control tries to keep the device function in resonance by the configuration of the PTO stiffness and damping as follows

$$k = \omega^2 \big(m + A(\omega)\big) \tag{23}$$

and

$$Bpto = B(\omega) \tag{24}$$

where k is the linear stiffness of the mooring system, and $Bpto$ is the PTO damping. Also m is the mass of the hull, ω is the angular velocity of the incident wave, $A(\omega)$ is the added mass coefficient and $B(\omega)$ is the radiation damping coefficient.

In order to specify the physical and swept volume of the hull, the connection between these two quantities must be determined

$$V_s = b * V \tag{25}$$

where V_s is the swept volume, V is the physical volume and b a factor which indicates their connection. The existing two upper bounds P_a and P_b for the power P that can be received for a sinusoidal wave of height $H = 2|A|$ and period $T = 2\pi/\omega$, is a useful indicator for the potential of the point absorber. Using the two bounds, it is possible to construct a very useful tool called Budal Diagram [3]. By equating P_a and P_b the swept volume can be found and consequently the physical volume and the PTO characteristics. An interesting methodology in order to choose the appropriate regular wave for the designing of the wave energy converter was proposed by Falnes [4] and it is resumed in the following steps:

a) Select a candidate sea site location and determine a wave power threshold $J_T(kW/m)$ which is being exceeded only one third of the year
b) Identify the peak period, or the peak energy period of the most frequent waves according to the sea site probability data (T_P or T_e).
c) Relate J_T and $T_P(T_e)$ to the regular wave of period (for the fully developed uni-modal sea) with the same wave power level. Find the wave height pf the corresponding regular wave using the equation $J_T = \frac{\rho g^2 H^2 T}{32\pi}$

To sum up, The first decision to be taken is the location of the deployment of the wave energy converter in the sea. The development of the machine and the selection of its characteristics must be connected with the morphology of the wave scatter of the selected site. Using the methodology which has been presented above, the main regular wave of the project is specified. This monochromatic wave is the fundamental quantity in order to construct the Budal Diagram, and as a result the power absorption bounds, the physical volume and the maximum stroke of the converter are calculated. In order to force the converter to function always in resonance, optimal control should be applied in order to change dynamically the stiffness and the damping coefficient of the Power Take Off (PTO).

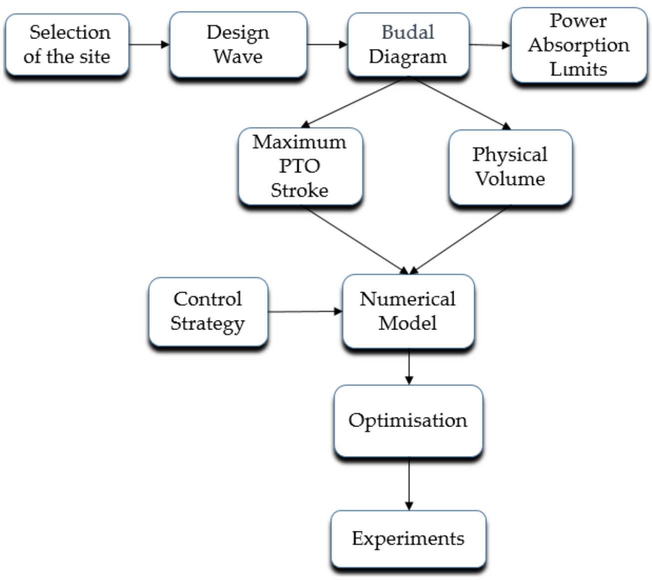

Fig. 7. Flow chart of the design of the Point Absorber.

D. Implementation in the island of Pantelleria

As it was referred in the previous pages the selected deployment site in Mediterranean Sea is the island of Pantelleria, Italy. In Fig. 8 a scatter of the wave occurrences in Pantelleria is presented, taken from an experimental campaign conducted by Politecnico di Torino in 2010. As it is shown, the majority of the waves have a period between 4.5 and 5.8 seconds. Furthermore, the majority of the waves have H_s between 0.3 and 1.5 meters and only a small fraction of the total annual waves reach the height between 2 – 3.5 meters. The selection of the characteristics of the wave energy converter has a very big connection with the most frequent waves in order to achieve satisfactory energy production during all the periods of the year. In Fig. 9 the energy density of the waves in Pantelleria is presented, while Fig.10 shows the wave spectrum in the coast of Pantelleria.

The first step in order to identify the principal regular wave of the design process is to determine a wave power

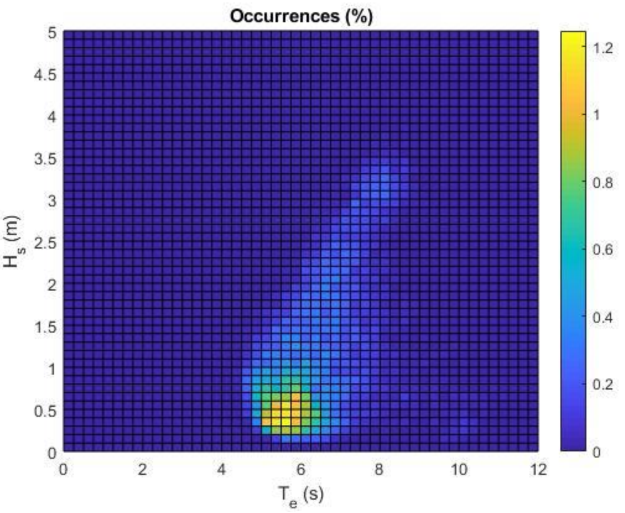

Fig. 8. Occurrences of the various waves in Pantelleria.

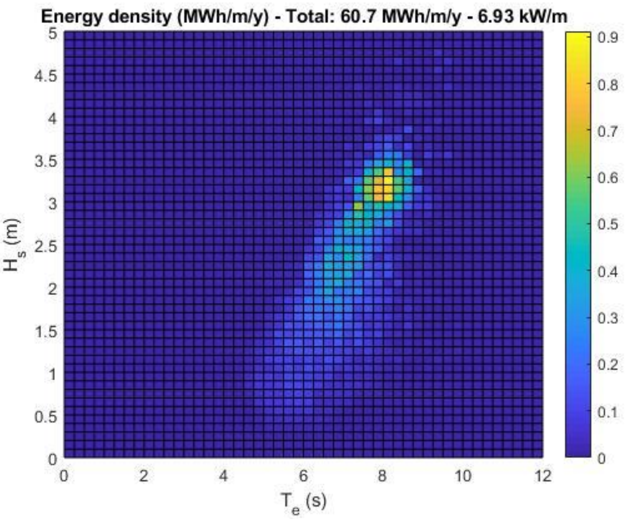

Fig. 9. Energy Density in Pantelleria.

threshold $J_T(kW/m)$ which is being exceeded only one third of the year in the selected sea site location. In other words, the one third of the waves that pass in the specific sea state annually, exceeds the critical power threshold of 5.048 W/m. According to Falnes [3], in order to design efficiently a point absorber and make it sustainable, it should function at full capacity for at least one third of the year. The second step is to identify the peak period or the peak energy period of the most frequent waves according to the sea site probability data (T_P *or* T_e). It was calculated

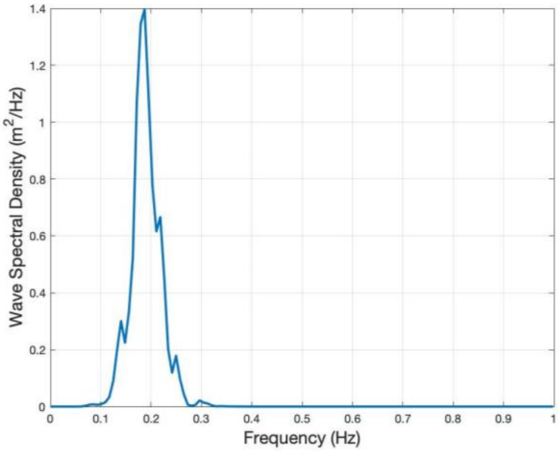

Fig. 10. Wave Spectrum in the coast of Pantelleria

analyzing the scatter that $T_P = 6.5\, seconds$. According to the Pierson-Moskowitz spectrum the period of the regular wave would be $T = 0.858 * T_P = 5.57\, sec$. Proceeding to the third step The resulting regular wave of the project has $T = 5.57\, s$ and $H = 1\, m$.

Afterwards the Budal diagram will be built using the resulting wave above. The first characteristic to be defined is the physical volume of the buoy. However, it must be clear that from the equation $P_a = P_B$ only the optimal swept volume can be identified. The next step is to decide and select the physical volume of the hull that it would be able to reach during the oscillation the maximum desired swept volume. If the swept volume is selected to be the same with the physical one, means that the minimum possible volume was selected and consequently lower cost. On the other hand, the main disadvantage in this situation is the fact that with minimum possible volume it is not easy to accomplish the desired swept volume. This happens because of the different phase which is created between the incident wave and the oscillation of the body, and only with advanced control methods could be solved. This is not the case when the physical volume is bigger than the swept volume and the body has the benefit to stay in phase with the wave. In any case buoys with different

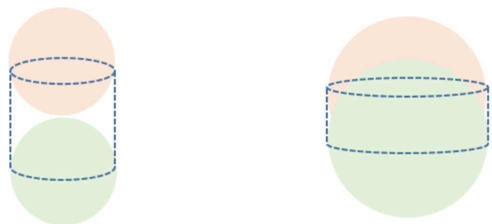

Fig. 11. Schematic representation of different buoys with similar swept volumes

volumes can achieve the same swept volume but with different oscillation amplitudes, as it is shown in Fig.11.

Using the project wave and considering $V = V_S$, the Budal diagrams were designed for a submerged point absorber. The resulting system is a sphere of 2.44 m $|S_3|max = 1.63\ m$ and Ds (submergence) equal to 4.8 m. In Fig. 12 the power absorption limits are presented.

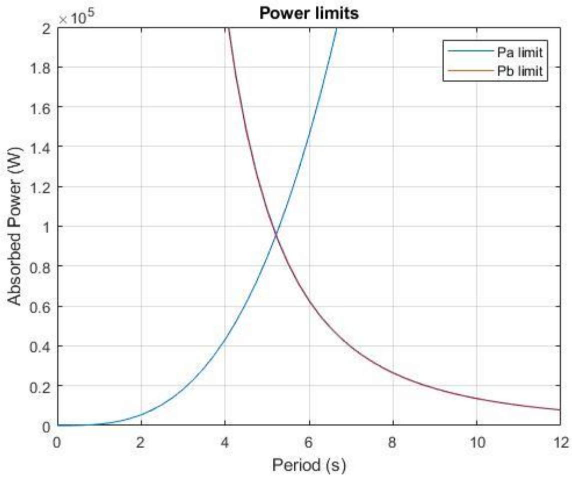

Fig. 12. Power limits.

However, in order to investigate also bigger hulls, using different b in the equation (25), five more hulls have been carried out from the equation $P_a = P_B$ .The characteristics of the hulls are presented in Table 2. It is underlined that in order to identify these volumes, a heaving point absorber is considered where the swept volume is

$$V_s = 2 * \pi * r^2 * S_{max} \quad (26)$$

Combining equations (25) and (26)

$$2 * \pi * r^2 * S_{max} = b * V = b * \frac{4}{3} * \pi * r^3 \quad (27)$$

And finally

$$S_{max} = b * \frac{2}{3} * r \quad (28)$$

Calculating the maximum displacement in Z axis S_{max} properly, it can be ensured that the buoy will not reach the surface of the sea. However, in order to increase the security, the coefficient c is introduced which indicated the depth csaf where the point absorber arrives in the maximum displacement:

$$csaf = d_s - (r + s_{max}) \quad (29)$$

where d_s the submergence of the center of gravity of the submerged hull:

$$d_s = c * \left(r + b * \frac{2}{3} * r\right) \quad (30)$$

and

$$safety\ factor\ c \geq 1$$

By introducing different values of b in the equation $P_a = P_B$ the following systems have been calculated. All the devices have been simulated in regular waves of H = 1 m and periods from 3 to 12 seconds using reactive control. The results are presented in Fig. 14. In Fig.13 a schematic representation of the device with all the technical detail is presented.

TABLE 2

TECHNICAL DETAILS OF THE SIX DIFFERENT HULLS

Symbol	Hull 1	Hull 2	Hull 3	Hull 4	Hull 5	Hull 6
m (kg)	56085	66088	126045	169851	246565	404449
R (m)	2.44	2.58	3.2	3.53	4	4.71
V (m^3)	60.8	71.64	136.63	184.12	267.13	438.44
Vs(m^3)	60.8	64.48	81.98	92.06	107	131.53
c	1.1	1.1	1.1	1.1	1.1	1.1
csaf	0.41	0.41	0.45	0.47	0.51	0.57
b	1	0.9	0.6	0.5	0.4	0.3
ds (m)	4.47	4.53	4.92	5.18	5.57	6.22
s3max (m)	1.63	1.55	1.28	1.18	1.07	0.94

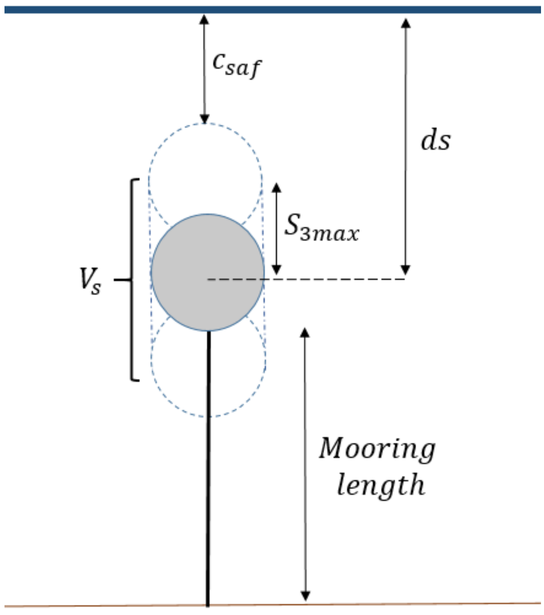

Fig. 13. Schematic representation of the point absorber.

TABLE 3
SELECTED CHARACTERISTICS FOR THE IRREGULAR ANALYSIS

Symbol	Quantity	Unit
k	Stiffness	193070 N/m
Bpto	PTO damping	9966 N*s/m

Budal diagram is presented in Fig. 14, the power absorption limits were calculated for a multiple degree of freedom motion, and the performances of all the devices are lower than the limits. Obviously the power production

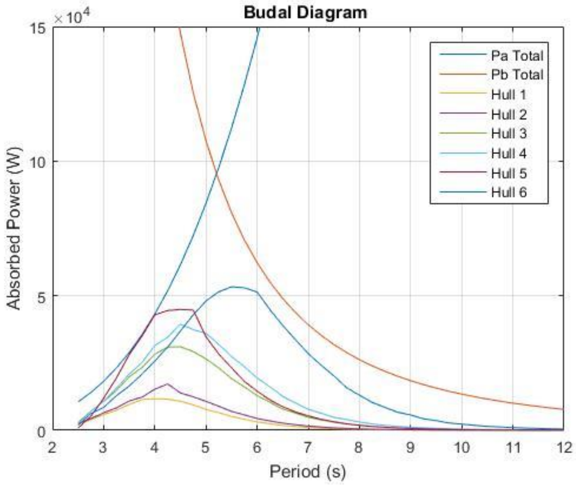

Fig. 14. Budal diagram

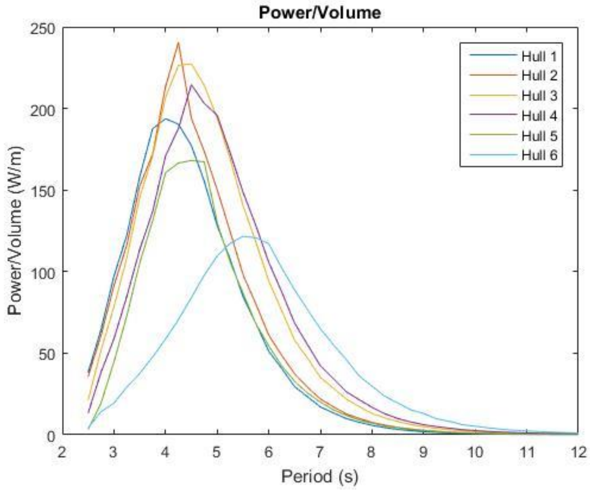

Fig.15. Power per volume performance.

is proportional with the physical volume of the body. Furthermore, as the volume increases, the pick of the productivity moves towards higher periods as well as the resonance period.

Hull 6 has the higher performance, however, it should not be forgotten that the target is to design a technological unit which will be feasible maximising the power output, not with respect to the available wave potential but to itself and resulting a competitive cost.

Taking into account that volume is an important factor of the device cost, the absorbed power per volume of each hull has been calculated.

As it is demonstrated in Fig. 15, the power production is not the only indicator for the successful design of the wave energy converter. Hull 6 presents the poorest performance from this point of view in contrary with the diagram in Fig. 14. Hulls 2, 3 and 4 present satisfactory power per volume performance. However, as it was mentioned before more advanced control methods should be applied in order to improve the function and the performance of hull 1 which targets to have equal swept and physical volume and as a result better cost efficiency.

IV. NON LINEAR WAVE ANALYSIS

In order to calculate a realistic performance of the point absorber, Hull 2 has been selected in order to carry out simulations using 228 irregular waves from the coast of Pantelleria. In contrast with regular wave analysis, fixed values of the PTO stiffness and damping have been used. According to [3], it was found that a potential heaving device should not exceed a power capacity of 5 kW/m. Hence for a spherical hull which has 5.16 m diameter, the capacity equals to 25 Kw. However, for a point absorber which can absorb energy also from surge, heave and pitch, a double or a triple capacity could give satisfactory results. In the Fig. 16, 17, 18 the Energy Matrix of different capacities is presented.

TABLE 4
ANNUAL ENERGY PRODUCTION

Power Capacity	Annual Production	Capacity Factor
25 kW	108.3374 MWh	49.32%
50 kW	274.1678 MWh	62.56%
75 kW	346.5451 MWh	52.66%
100 kW	349.6523 MWh	39.84%

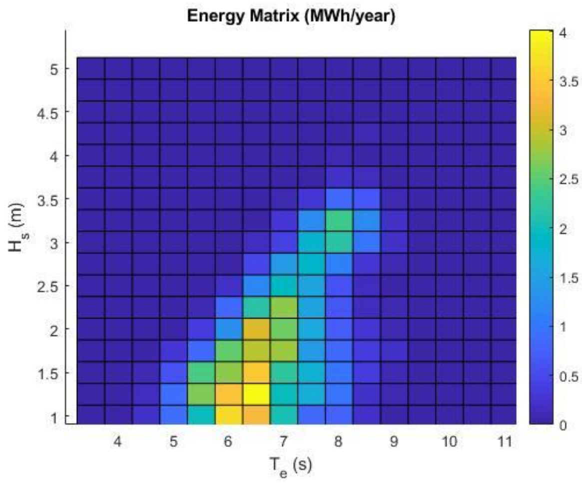

Fig. 16. Annual productivity of 25 kW device.

It is demonstrated that a bigger capacity than 75 kW is not efficient because of the PTO limits that have been taken into account in the simulation. The maximum PTO stroke of the device is 1.55 m equal to S_{max}. The value of the annual production doesn't rise proportionally to the power capacity, a fact to be investigated. For instance, besides the capability of the device to absorb energy, a spherical hull is not suitable for large pitch amplitudes. A different geometry could be able to oscillate in bigger pitch amplitudes. Also, advanced control should be investigated, together with real time sea state evaluation, in order to maximize power absorption [18].

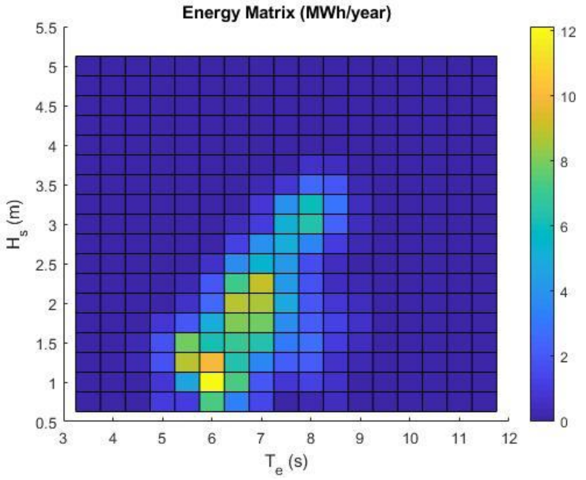

Fig. 17. Annual productivity of 50 kW device.

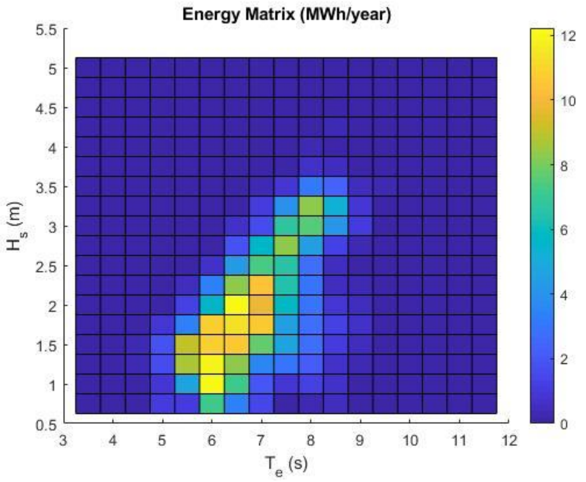

Fig. 18. Annual productivity of 75 kW device.

V. CONCLUSION

A submerged point absorber has been designed taking into account the waves of the coast of the island Pantelleria in Mediterranean Sea. For the hydrodynamic analysis in both regular and irregular waves, a mathematical model based in the Cummins equation was used. The technical details of the device have been selected in order to construct a WEC which will be efficient taking into account its cost. According to a preliminary assessment the power capacity of the point absorber should not exceed 75 kW. Further study and experiments will take place in order to use different geometries, advanced control methods and economic analysis to optimise the technology.

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