# Tuning Wave Energy Converters to local wave conditions

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*Abstract*—**An offshore site is generally characterized by the presence of various wave systems including wind seas and swells. Wave parameters of dominant wave systems can be used to tune the natural period of a degree of freedom of a resonant Wave Energy Converter (WEC). To this aim, this paper introduces a methodology to put a novel WEC concept to resonate with the waves. The WEC is composed of a main vessel, hydrodynamic and dual drag force subsea structures that hang from the vessel, and a mechanism for the power take-off. To increase the natural period, which generally is less than that of swells, the inertia of the vessel is increased by using the freely available seawater. The subsea structures exert a large drag force when moving upwards, and a low drag force when moving downwards. This allows for controlled motion responses during operation and reduces the risk of having slack lines. In our case study, the roll natural period of a medium size barge is increased from 5.5 to 13.4 s, which is the peak period of dominant swells at Isabela Island. This study allows for a systematic design of resonant-type WECs, which is necessary for the development of feasible WEC alternatives of novel and existing concepts.**

*Index Terms*—**Wave energy converter, Tuning natural period, Heave, Roll, Resonance**

## I. INTRODUCTION

THE demand for energy and the need for a transition from fossil fuels to renewable energy sources promotes the development of wind, solar, and ocean energy [1]. Among them, the ocean contains a huge amount of energy, which comprises marine currents, osmotic, ocean thermal, tidal, and wave energy [2]. Regarding wave energy, it presents an enormous power potential, estimated at 2.11 +/- 0.05 TW globally [3], but just reaching a technological readiness level (TRL) of 6, which means that prototypes of wave energy converters (WECs) have been tested in an intended environment close to expected performance [4]. As can be noticed, the development of WECs is not mature, it is still unreliable, and its Levelized Cost of Energy (LCoE) is much higher than that of fossil fuels and other renewable resources such as wind energy [3].

Since 1799, when the first patent was registered [3], several types of WEC have been proposed, which differ from each other mainly in terms of operating principle, deployment site, and power take-off systems (PTO). Despite the differences, a common design criterion is to prioritize the Wave-Structure Interaction (WSI) in order to convert most of the wave energy into mechanical energy that can be further converted into more useful energy, such as electricity. In this context, an element of the WEC system is a mechanical oscillator, which performs better as the wave frequency approaches its natural frequency [5]. Several studies consider resonance or near-resonance WEC operating conditions, some of which resulted in prototypes. For instance, the Pelamis is an offshore, floating, slack-moored WEC consisting of semi-submerged cylinders linked by hinged joints [6]. The relative displacements between the cylinders are the result of the ocean wave surface and are exploited to generate usable power. The Pelamis mechanism is designed in a manner that the rotational stiffness of the hinges attenuates the local wave profile. Even though WEC systems operate following simple (often intuitive) concepts, the local conditions impose important constructive challenges. For example, drag and slamming impose the most extreme loads on the WECs because of the high water velocities and accelerations of wave particles in an extreme wave environment [7]. In this context, stormy conditions can generate a harsh environment where WEC robustness is tested frequently. This means that structural design requires considering operational and survival events but avoiding oversizing since it may lead the project to present unfeasible costs and inefficient operating conditions. In addition, it is worth mentioning that, in this kind of project, the construction and marine operation-related costs are not usually disclosed in detail and WEC efficiency is rarely reported because of corporate stealth.

By considering a distinct working principle but tunable WEC, point absorbers with one or more bodies have been widely studied both numerically and experimentally. For example, the PowerBuoy is perhaps the best-known two-body floating system in operation. The PowerBuoy is composed of two floating bodies; the outer one acts as the oscillating body and the inner one as a fixed reference. Their relative displacement is designed to resonate with local waves [7]. The operation of these devices depends on the heave oscillation amplitude of a buoy which follows the wave, although pitch and surge modes have also been examined in literature [8]. Setting resonant conditions of two-body floating WECs presents its own challenges. This is because restoring forces constantly change because of the varying waterplane area. Thus, the solution must be analyzed in two parts: (i) assuming that the point absorber is fixed, and the wave pressure is exerted on its surface, and (ii) assuming that the water surface



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is still, and the oscillating point absorber is causing radiated waves due to its dynamics [9].

As can be noticed, the WEC tuning represents a key issue in the development of an efficient system despite the difficulties that this entails. The challenge is more evident if one considers the fact that ocean waves are an ultra-low-frequency energy source, varying typically in the 0.1 – 1 Hz frequency bandwidth [8], and usually mechanical systems show resonance frequencies higher than that. In this context, several studies in the literature report the efforts in the implementation of active control strategies [8], [10], [11] in order to allow the systems to be tunable in situ, and systems designed with fixed properties but tuned to a specific wave system before its deployment [12], [13]. Regarding the latter, tuning can be achieved by setting inertia, stiffness, and damping as in typical mechanical oscillators. However, unfeasible weights and/or sizes represent the most important limitations since they would complicate the construction and marine operation-related activities [9]. On the other hand, this kind of system is robust and can be ideal for wave climates where there exists a predominant wave system with extremes not much different from operating conditions.

Based on the above, there are few WEC studies dedicated to the tuning of its resonant frequencies. This paper shows that it is possible to increase the natural period (lower the frequency) of a novel WEC concept by using freely available seawater. The natural period of the WEC is increased to that of predominant swells present in the Southern part of Isabela Island.

## II. Methodology

### A. Wave conditions at the offshore site

For an offshore location at the Southern part of Isabela (Galapagos) Island (-1°N, 269°W), Fig. 1 shows the presence of three main wave systems through the year. Wave system No. 1 corresponds to Southern swells, wave system No. 2 corresponds to Northern swells, and wave system No. 3. corresponds to Southern wind seas. The peak period (Tp) for swells is around 13 s, which is almost constant during the entire year, while for wind seas is 8 s. System No. 1 is the most energetic from these wave components with significant wave height (Hs) varying mostly between 0.5 and 3 m, see Fig. 2. The swells have almost a constant period, and the total Hs does not vary significantly throughout the year. In fact, extreme events do not exceed Hs= 3.5 m, which makes it possible to design a WEC for similar operating and extreme conditions. Moreover, the mean wave power during the entire year is around 12 kW/m.

A scatter diagram of the offshore site is shown in Table I. It can be observed that the most recurrent sea state has a significant wave height Hs= 1.5 m, and a peak period Tp= 13 s.

### B. Conceptual design of a resonant WEC

Flopper stoppers were proposed as a passive roll compensation system for offshore construction vessels

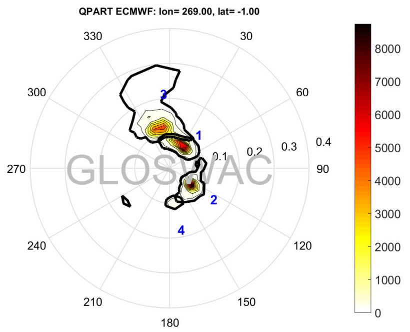


Fig. 1. Statistically defined wave systems at Isabela island (-1°N, 269°W)

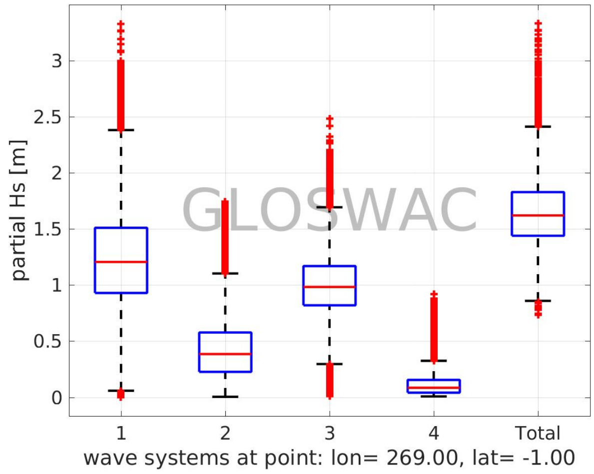


Fig. 2. Hs boxplot for the wave systems

TABLE I
Scatter diagram for Isabela Island -1.8°N, 269°W

| **Hs (m)** | **Tp (s)** 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 16 | 17 | 19 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| **0.5** | 0 | 0 | 4 | 21 | 11 | 20 | 26 | 17 | 7 | 4 | 0 |
| **1.0** | 280 | 346 | 417 | 719 | 3018 | 5506 | 5086 | 3491 | 1957 | 917 | 285 |
| **1.5** | 128 | 523 | 1485 | 1592 | 2329 | 11328 | 19391 | 14747 | 8401 | 3489 | 1079 |
| **2.0** | 0 | 20 | 216 | 814 | 655 | 764 | 4436 | 8091 | 4245 | 1561 | 406 |
| **2.5** | 0 | 0 | 0 | 14 | 131 | 78 | 127 | 789 | 993 | 388 | 69 |
| **3.0** | 0 | 0 | 0 | 0 | 6 | 5 | 11 | 27 | 103 | 85 | 19 |

[14]. These devices can increase the mass moment of inertia of the vessel about the longitudinal axis to shift its roll natural period away from the wave period. In this paper, we propose to increase the natural period of roll or heave to put the WEC to resonate with the waves. A natural period of a degree of freedom can be increased by either decreasing the stiffness or increasing the inertia. For a barge with parallel side walls, the second option is plausible, and it can be achieved by using the freely available seawater.

Figure 3 shows a subsea structure hanging from the centreline of a vessel. To shift the heave natural period of the WEC, the structural $m_{33}$ and added $a_{33}$ masses of the vessel can be increased by the entrapped $m_{entr}$, structural $m_{str}$ and added $a_{str}$ masses of the subsea structure. Likewise, Fig. 4 shows that subsea structures can be used to increase the mass moment of inertia about the longitudinal axis to achieve roll

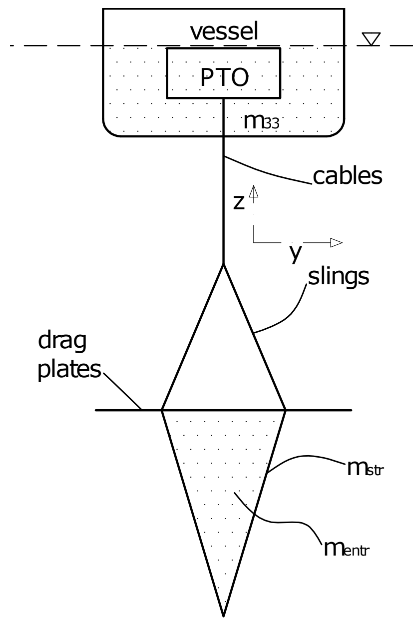


Fig. 3. WEC concept based on heave resonant mode

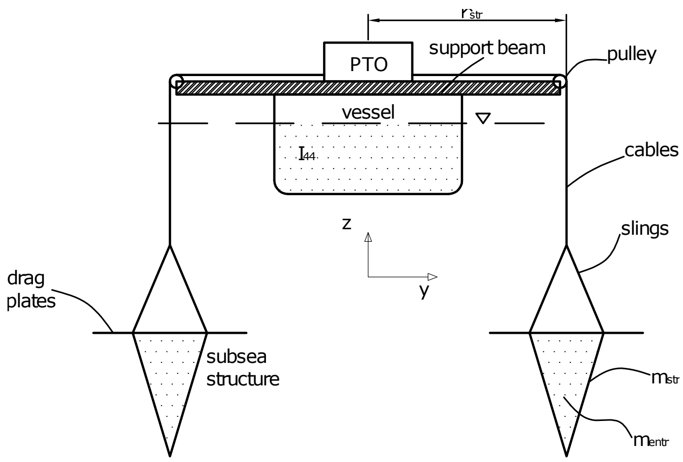


Fig. 4. WEC concept based on roll resonant mode

resonance. At resonance, motion responses and forces can be unacceptable and resulting forces and stresses can be destructive for mechanical components, but they can be reduced or controlled using passive systems. Figures 3 and 4 show that drag plates on the subsea structures can be used to reduce dynamic responses to acceptable values. Fig. 5 (a) shows the subsea structure moving and accelerating downwards. The only force that avoids slack lines (T $\leq$ 0) is the submerged weight because the inertia $F_{str}^{in}$ and drag $F_{drag}$ forces act opposite to T. In contrast, Figure 5 (b) shows a free body diagram of the structure when moving upwards and accelerating downwards. $F_{drag}$ acts against $F_{str}^{in}$, and thus, the drag force together with the submerged weight $W_{str}$ increases the tension T in the cable while diminishing the motion response of the vessel. Thus, the drag plates should exert a large drag force when the subsea structures move upwards, and low drag when the structures move downwards. Figure 6 shows an innovative dual drag force hydrodynamic structure with leaf springs in the perimeter. These springs deflect downwards when the subsea structure moves upwards and deflect upwards when the structure moves downwards. These leaf springs have to be designed to provide structural flexibility and fatigue endurance for the intended service life.

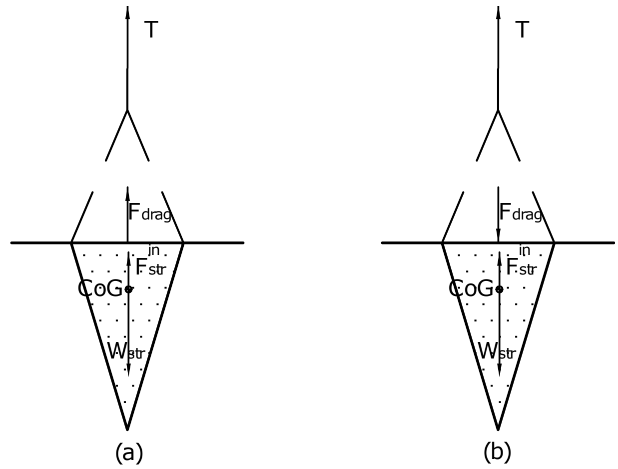


Fig. 5. Free body diagrams of a subsea structure. (a) Subsea structure moving downwards; (b) Subsea structure moving upwards.

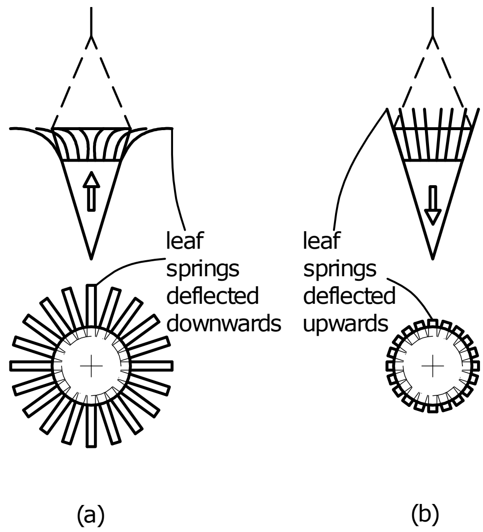


Fig. 6. Leaf springs on the subsea structures. (a) Springs in opened position; (b) Springs in closed position.

The resulting dynamic tension in the cable can be used to power a mechanical device. For the roll resonant WEC concept shown in Fig. 4, the difference in dynamic tensions in the cables, which mostly depend on roll can be used to power a mechanical gearbox of the PTO. Figure 7 shows that the main cables can introduce a stochastic horizontal motion of a sliding beam. The upper and lower flanges of the beam are welded to racks, which drive gears that convert this horizontal motion into a unidirectional rotational motion of a mechanical shaft. To reduce the effect of impact loads, compression springs are placed at the ends of the beam.

Figure 8 illustrates that there are two sets of racks, gears, and ratchets to power a main shaft. When both racks move to the left, rack No.1 moves gear No. 1 and by means of a pawl and ratchet No.1 the shaft is rotated in a counter-clockwise direction. During this motion cycle, rack No.2 does not power the shaft, see

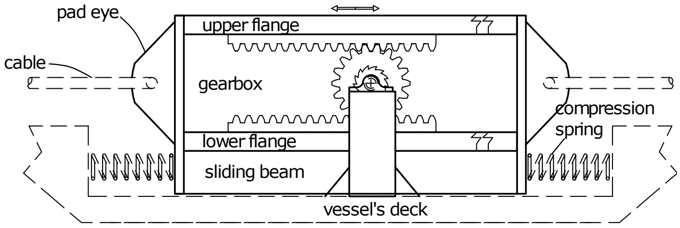


Fig. 7. Schematinc view of the PTO's sliding beam.

Fig. 9. Likewise, the racks move to the right, only rack No. 2 powers the shaft in the same counter-clockwise direction, see Fig. 10.

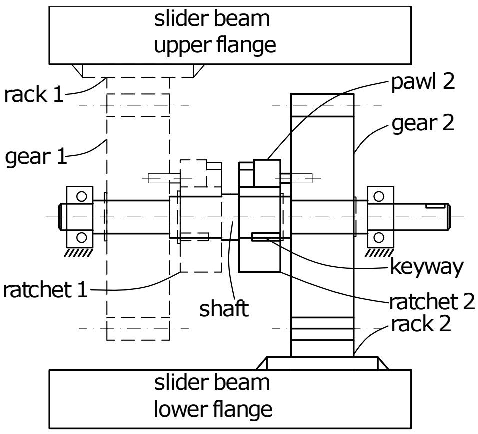


Fig. 8. Mechanical components of the PTO's gearbox.

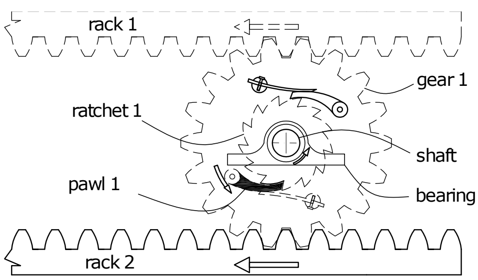


Fig. 9. Side view with rack No. 2 powering the main shaft.

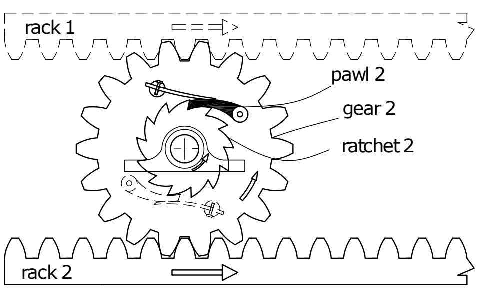


Fig. 10. Side view with rack No. 1 powering the main shaft.

## C. *Tuning the WEC to local conditions*

Based on the physical parameters shown in Fig 3, the new heave natural frequency can be expressed by equation (1). Where $\rho$ is the seawater density, g is the acceleration of gravity and $A_w$ is the still water plane area.

$$\omega_{n33} = \sqrt{\frac{\rho g A_w}{m_{33} + a_{33} + m_{str} + a_{str} + m_{entr}}} \quad (1)$$

Similarly, the new roll natural period can be calculated from equation (2). Where $GM_{new}$ is the metacentric height of the multibody WEC system, $k_{44}$ is the radius of gyration about the longitudinal axis, $\nabla$ is the vessel displacement, $a_{44}$ is the mass moment of inertia about the longitudinal axis and $r_{str}$ is the lever arm from which the subsea structures are hanging and N is the number of subsea structures.

$$\omega_{n44} = \sqrt{\frac{\rho g \nabla GM_{new}}{k_{44}^2 \rho \nabla + a_{44} + N \times r_{str}^2 \left(m_{str} + a_{str} + m_{entr}\right)}} \quad (2)$$

## D. *Numerical modeling*

In this paper, we develop a numerical model to harvest the roll resonant motions of the WEC device. For a vessel with length L= 30 m, breadth B= 14 m, and draught S= 2.6 m, two conical subsea structures with diameter D= 9 m and height Z= 11 m, are suspended from the sides at the vessel midships. Using $r_{str}$= 10 m, $m_{str}$= 2.2×10$^5$ m, $m_{entr}$= 3.1×10$^5$ m, $a_{str}$= 1.7×10$^5$ m, $GM_{new}$= 3.5 m, $I_{44}$= 2.8×10$^7$ kg$m^2$, $a_{44}$= 2.04×10$^7$ kg$m^2$, equation (2) gives $\omega_n$44= 0.47 rad/s or the corresponding natural period $T_{n44}$= 13.2 s.

For the parameters given above, figure 11 shows a schematic view of the numerical model developed in the state-of-the-art computer code Ansys-AQWA [15]. The compression springs in Fig 7 are assumed to have a maximum compression of 0.3 m under the action of a 1000 kN tension force. Thus, the stiffness of the spring dominates over that of the cable; a value of 3.5×10$^6$ N/m is applied.

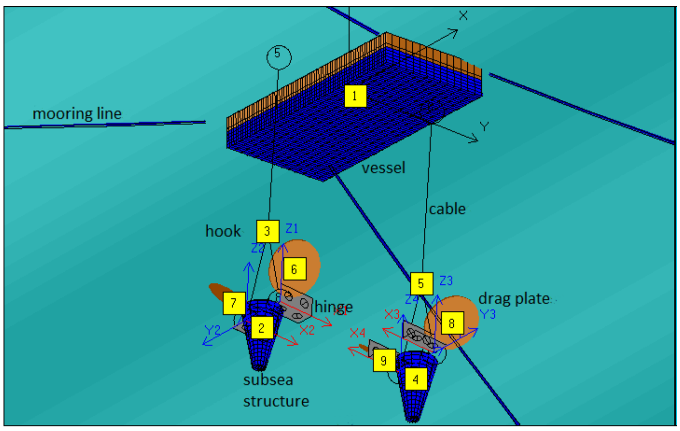


Fig. 11. Schematic view of the numerical model of the resonant roll WEC concept

Because the system is highly non-linear, dynamic responses of the vessel are solved in the time domain (using numerical integration methods), see equation (3). Where M and A are the structural and added mass matrices, h is the retardation function that includes the added mass and damping terms, K is the stiffness matrix, $F^{ext}$ is the external force vector that includes wave, mooring, and coupling forces and $x$, $\dot{x}$, $\ddot{x}$ are the position, velocity and acceleration vectors. For the subsea structures, the responses can also be computed by solving equation (3), but the external force also includes the non-linear drag force acting on the drag plates. For thin plates with a large width-to-thickness ratio, a drag force coefficient of $C_D$= 4 can be applied [16]. These plates are modelled as discs with an equivalent area to the leaf springs connected by means of hinges and a torsional spring with a rotational stiffness $k_r$= $6 \times 10^5$ Nm/rad.

$$[\boldsymbol{M} + \boldsymbol{A}^{\infty}]\, \ddot{\boldsymbol{x}}(t) + \int_0^t \boldsymbol{h}(t-\tau)\, \dot{\boldsymbol{x}}(\tau)\, d\tau + \boldsymbol{K}\boldsymbol{x}(t) = \boldsymbol{F}^{ext}(\boldsymbol{x}, \dot{\boldsymbol{x}}, t) \quad (3)$$

## III. Results

### A. Linearized roll response amplitude operators (RAOs)

Using a white noise spectrum with Hs= 1 m, the linearized roll RAOs for the vessel alone and the WEC in beam seas are plotted in Fig. 12. It is observed that the roll natural period shifts from 5.5 to 13.4 s.

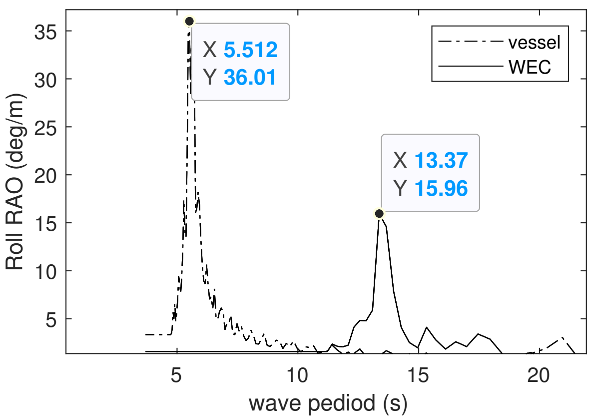


Fig. 12. Vessel and WEC roll RAOs for beam seas

### B. Hydrodynamic forces

Figure 13 shows an example of time series for hydrodynamic forces and the dynamic tension in a cable. It is observed that the inertia forces control the dynamic tension. Froude-Krilov, diffraction, and drag forces have less contribution to the tension. This is expected considering the large structural, added, and especially entrapped water masses in the susbea structures. For this reason, the maximum tension, and thus, the difference in dynamic tension occurs when the roll is maximum. At this instant of time, the velocity of the sliding beam should be zero. Figure 14 shows that the roll and the difference in tension are in phase. The velocity of the beam is plotted according to the actual magnitude difference in dynamic tension, roll period and stiffness of the helical compression springs, see Sec. II-D.

Figure 14

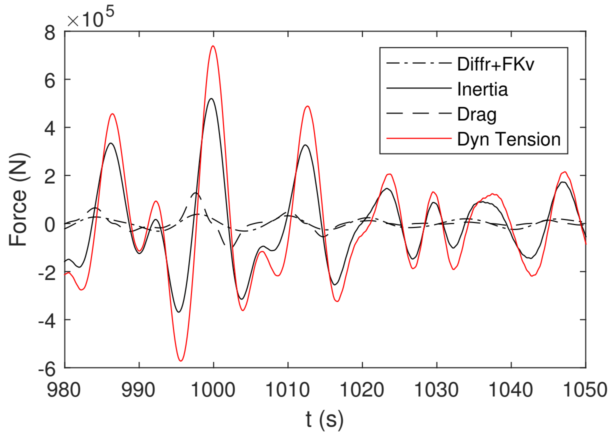


Fig. 13. Example of time series for hydrodynamic forces, using Gaussian spectrum with Hs= 3 m and Tp= 13 s.

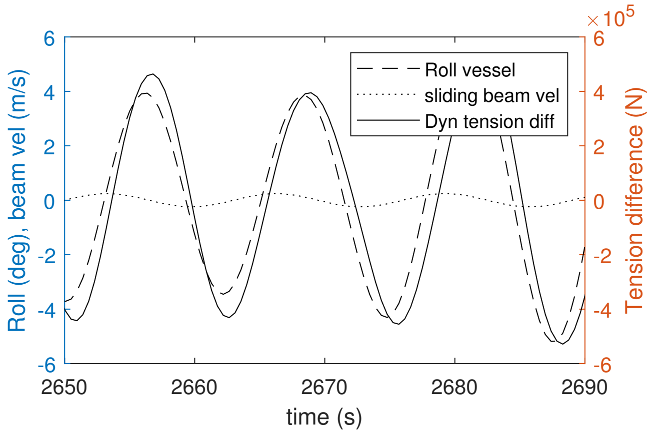


Fig. 14. Example of time series for roll, tension difference and velocity of the sliding beam using Gaussian spectrum with Hs= 1.5 m and Tp= 13 s.

### C. Mechanical power

Based on figure 14 the instantaneous power can be computed by multiplying the instantaneous difference in dynamic tension with the corresponding velocity of the sliding beam. The power is averaged over a 3-hour duration per sea state of the scatter diagram shown in Table I. For beam seas, Fig. 15 summarizes the mechanical power per sea state. The yearly average power is 47 kW.

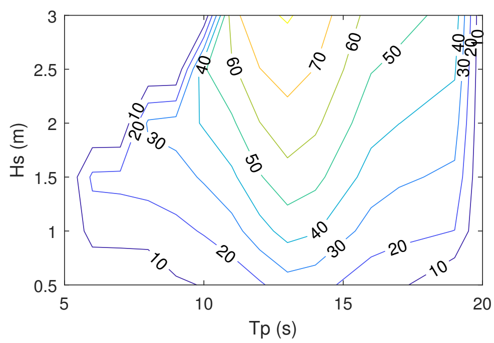


Fig. 15. Mechanical power on the main shaft for the Isabela Island scatter diagram using beam seas.

## IV. Discussion

Each offshore site has its own wave climate, generally dominated by a few wave systems. In this paper, the Southern part of Isabela Island has fairly constant wave elevations throughout the year. In fact, extreme waves are not much larger than average ones. Swell wave systems have a nearly constant wave peak period of 13 s, which was used as a reference to tune de WEC device. This means that every resonant WEC should be designed and tuned for the local condition where the device is going to operate.

To tune the natural period of a degree of freedom of a vessel, particularly one with parallel side walls, e.g. a barge, the inertia parameter can be easily increased to shift the heave or roll natural periods to match those of the waves. In this paper, we proposed a passive system that make use of the freely available seawater. The device doesn't have underwater moving parts such as articulations and hydraulics that can be dangerous for ocean life. The dual drag force subsea structure is an innovative design that suppresses excessive roll motions and reduces the required submerged weight to avoid slack lines. This will be beneficial for construction and marine operation activities. The leaf springs are passive systems (well-known in the automotive industry) that require little maintenance and can be designed to deflect and survive the loads encountered during their service life.

The mechanical power in the device comes from the dynamic tension difference in the main cables. From figure 13 it was shown that inertia forces are extremely large compared with other hydrodynamic forces. However, researchers in the past have shown that drag forces acting on flopper stoppers can be larger that inertia forces [14]. Thus, whether the tension in the cables is drag- or inertia-dominated, is not relevant in the WEC device introduced in this paper, because these forces will not occur simultaneously in both subsea structures.

The available wave power in the Galapagos Islands can reach an average value of 12 kW/m of wave length. The WEC concept in this paper has a length of 30 m and the annual average power is 47 kW. This shows that only 13 % of the available power is harvested. It is evident that the best degree of freedom to harvest is heave; however, considering the size of the vessel, it is necessary a huge amount of entrapped water in the subsea structures and the resulting forces in the cables can be unacceptable. Thus, it is important to size the main structure components by considering the feasibility of its construction and operation. Moreover, it can be interesting to further investigate the performance of these devices when various degrees of freedom are harvested simultaneously.

The stochastic displacement of the sliding beam in figure 7 is converted into a unidirectional rotation of a mechanical shaft using two sets of racks, gears, and ratchets; however, other alternative mechanisms can be applied. Preliminary sizing of mechanical and structural components is not conducted in this paper.

## V. Conclusions and Recommendations

This paper introduces a novel WEC resonant concept, which is tuned for the Southern part of Isabela Island. The WEC device is composed of a vessel, two subsea structures hanging from the sides of the vessel by means of cables, and a mechanical power take-off system.

The resonant WEC is tuned to the local wave conditions by increasing the roll mass moment of inertia. This is achieved by using freely available seawater as entrapped water of the subsea structures. The natural period of the vessel shifts from 5.5 to 13.4 s, which matches the peak period of two swell systems present in the offshore site.

The subsea structures have innovative dual-drag force hydrodynamic shapes. A large drag force allows for reducing the roll response when the structure moves upwards. In contrast, the drag force reduces when the structure moves downwards, which is necessary to reduce the risk of having slack lines.

For beam seas, the annual average mechanical power is 47 kW, which is about 13 % of the available power at the site.

To increase the efficiency of resonant wave devices, simultaneous harvesting of heave and roll resonant motions can be further investigated.

Model tests to support findings from this paper should be conducted.

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