Preliminary Assessment Of A Wave Energy Conversion Principle, Using Fully Enclosed Multi-Axis Inertial Reaction Mechanisms

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PRELIMINARY ASSESSMENT OF A WAVE ENERGY CONVERSION PRINCIPLE, USING FULLY ENCLOSED MULTI-AXIS INERTIAL REACTION MECHANISMS

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A novel concept of Wave Energy Converters is considered, composed from a class of fully enclosed, appropriate, internal body mechanisms, which provide inertial reaction against any multi-axis, multi-direction motion of an external vessel. This ensures maximum wave energy capture in comparison to other types of wave energy converters, based on internal reaction mechanisms. The internal bodies are suspended from the external vessel body in such an appropriate geometrical configuration, resulting to a linear trajectory for the center of the mass of the suspended body with respect to the external vessel. Moreover, the suspension geometry ensures a quite simple and robust technological implementation, removing the restrictions of other linear, pendulum or gyroscopic variants of inertial reacting bodies. Furthermore, the mass and the inertia distribution of the internal body is optimized for the maximal conversion and storage of the wave energy. As a result, the dynamic behavior of the internal body assembly, is essentially that of an equivalent vertical physical pendulum. However, the resulting equivalent pendulum length and inertia can far exceed those achieved by an actual technical implementation of other pendulum variants, resulting to a significant reduction of the suspended mass. A preliminary design of such a mechanism is considered and a simple equation is derived to estimate the power conversion potential of such a mechanism, as a function of the main geometrical and inertial design parameters. Then, the behavior of the mechanism is evaluated under a combination of surge and pitch excitations.

1. Introduction

More than a thousand of patents and tens (if not hundreds) of experimental prototypes are being tested in the sea. Some comprehensive recent reviews can be found in [1], [2], and [3]. Today, the main obstacles for efficient wave energy conversion, are mainly related to the requirement for survivability in extreme weather conditions and to the energy efficient and reliable design of the power take-off mechanism (PTO). Towards the last direction, numerous concepts of wave energy converter systems have been conceived, consisting from two-body configurations, in which only one body is in contact with the water and the other body is located above the water or is totally enclosed inside the wetted one.

The earliest example in this direction are perhaps the Frog and PS Frog designs at the University of Lancaster in England [4, 5]. Parallel, an approach for the theoretical modelling and control of such devices has been performed in [6, 7]. An interesting variant of this design, acting essentially in the form of a vertical pendulum has been proposed: the SEAREV [8] concept. The basic disadvantages of these two designs consist in their limited capability for wave capture due to the single axis motion and in the big masses they require for efficient energy capture, thus, demanding complex and unreliable support structures.

The novel concept for the design of a general class of fully enclosed internal body configurations, providing inertial reaction against the motion of an external vessel introduced in this paper, is able to drastically overcome the disadvantages of the abovementioned designs. Acting under the excitation of the waves, the external vessel can perform in general, a six degrees of freedom arbitrary translation and rotation in space.

2. Equations of motion

2.1 Description of the assembly and basic definitions

The considered assembly, is depicted in Fig. 1, consisting from a floating external vessel V, into which an internal, four-bar mechanism ABDE is hoisted. The waves induce to the vessel, a simultaneous surge motion of magnitude u and a pitching motion of an angle θ, with respect to the inertial coordinate system OXY.

The member DE of the internal four-bar mechanism, provides a basis onto which a solid body S is placed. The solid body provides a reaction mass to the motion of the external vessel, rotating with an angle φ about the Z axis of the inertial reference frame OXY. The initial (rest) position R of the centre of mass of this solid body, is located on the axis of symmetry of the vessel V, at distance a, from the origin O. The coordinate reference system RX_bY_b is rigidly attached to the vessel V, following its motion.

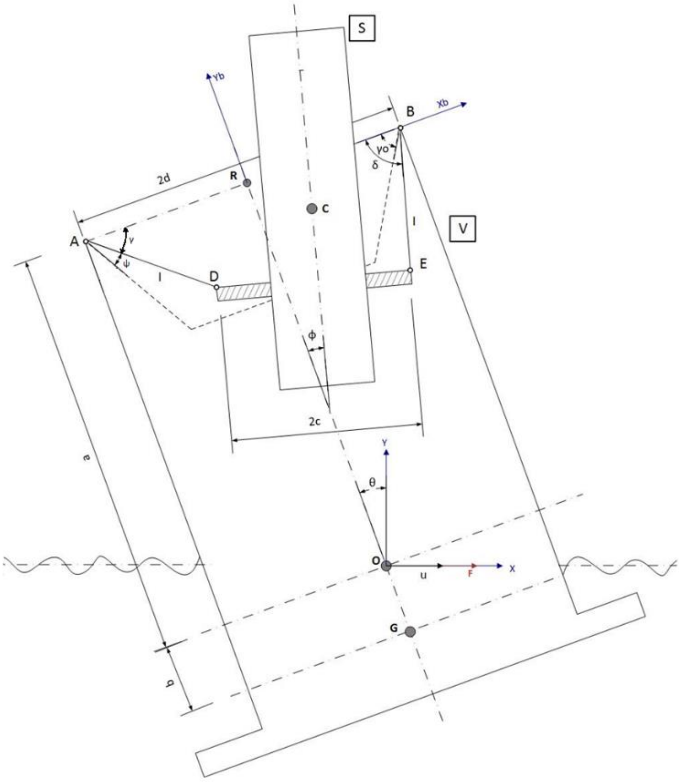

Figure 1: Schematic presentation of the assembly considered. An internally reacting body S is suspended by an appropriate mechanism by an external floating vessel V.

- O: intersection of the level of the sea with the vertical axis of symmetry of the vessel,

- C: centre of mass of the body S,
- R: origin of the body axis system RX_bY_b - initial (rest) position of C,
- G: centre of mass of the vessel V,
- a: distance between R and O,
- b: distance between G and O,
- φ: rotating angle of the body S about the Z axis,
- θ: pitching motion induced by the waves,
- u: surge motion induced by the waves.

2.2 Kinematic analysis

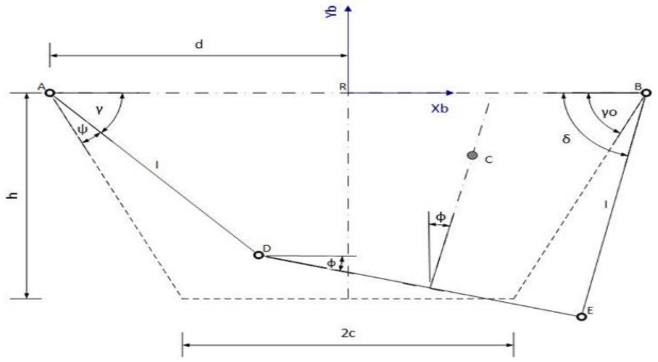

Figure 2: Geometric parameters and kinematic variables of the mechanism.

The basic geometrical configuration of the mechanism is defined by the selection of the three independent lengths d, c, h. The rest of the geometric parameters can be retrieved as follows:

$$l = \sqrt{(d-c)^2 + h^2} \quad , \quad tan(\gamma_o) = \frac{h}{(d-c)} \quad , \quad h = l\, sin\, \gamma_0 \tag{1}$$

The reference (rest) position of the mechanism, indicated by dashed lines, is defined by the relations: φ = ψ = 0 and γ = δ = γ_0. The origin R of the coordinate system RX_bY_b, is selected in the middle of the stationary link (ground) AB of the mechanism, with the position of the axes indicated as in Fig. 1. Through kinematic analysis of the mechanism, the main variables can be fully retrieved as a function of a chosen, single degree of freedom, in this case the angle φ, which results to:

$$\gamma = 2\, tan^{-1}\left(\frac{-a_{1\varphi} + \Delta}{a_{2\varphi}}\right) \quad , \quad \psi = \gamma_0 - \gamma \quad , \quad \delta = sin^{-1}\left(sin\, \gamma + \frac{2c}{l} sin\, \varphi\right) \tag{2}$$

$$a_{2\varphi} = r_\varphi + x_\varphi l \quad , \quad a_{1\varphi} = y_\varphi l \quad , \quad a_{0\varphi} = r_\varphi - x_\varphi l \tag{3}$$

$$r_\varphi = x_\varphi^2 + y_\varphi^2 \quad , \quad \Delta = \sqrt{{a_{1\varphi}}^2 - a_{2\varphi} a_{0\varphi}} \tag{4}$$

$$x_\varphi = d - c\, cos\, \varphi \quad , \quad y_\varphi = \quad c\, sin\, \varphi \tag{5}$$

The time derivatives of the compatibility relations of the closed kinematic chain from Eqs (2), lead to the following equations for the angular velocities:

$$\dot{\gamma} = -\mu\dot{\varphi} \quad , \quad \dot{\psi} = -\dot{\gamma} = \mu\dot{\varphi} \tag{6}$$

$$\mu = \frac{2c}{l\sigma} \quad , \quad \sigma = \frac{sin(\gamma + \delta)}{sin(\varphi + \delta)} \tag{7}$$

The initial position of the centre of mass C of the body S attached to the mechanism, is assumed to coincide with the origin R of the coordinate system RX_bY_b at the rest position of the mechanism. Therefore, the coordinates of point C, x_B and y_B with respect to this system at an arbitrary position of the mechanism, can be derived as follows:

$$x_B = -d + l\, cos\, \gamma + c\, cos\, \varphi + h\, sin\, \varphi \quad , \quad y_B = l\, sin\, \gamma + c\, sin\, \varphi - h\, cos\, \varphi \tag{8}$$

It is easy to derive that, the vertical position y_B of the centre of mass, is equal to zero in the predicted range of angle φ, when the relation between the main lengths of the mechanism is l=2d=4c, which is the special case of the Roberts linkage (AD=BE=AB=2*DE). Therefore, the point C moves approximately in a straight line over the segment AB. The velocities of point C can be derived as follows:

$$\dot{x}_B = r_x\dot{\varphi} \quad , \quad \dot{y}_B = r_y\dot{\varphi} \tag{9}$$

$$r_x = l\mu \sin\gamma - c\sin\varphi + h\cos\varphi \quad , \quad r_y = -l\mu\cos\gamma + c\cos\varphi + h\sin\varphi \tag{10}$$

The translation of the reaction mass according to the system OXY is as follows:

$$x_M = u + x_B\cos\theta - (\mathrm{a} + y_B)\sin\theta \quad , \quad y_M = x_B\sin\theta + (\mathrm{a} + y_B)\cos\theta \tag{11}$$

The expression of the corresponding velocities $\dot{x}_M$ and $\dot{y}_M$ result as follows:

$$\dot{x}_M = \dot{u} - l_{xM}\dot{\theta} + r_{xM}\dot{\varphi} \quad , \quad \dot{y}_M = l_{yM}\dot{\theta} + r_{yM}\dot{\varphi} \tag{12}$$

$$l_{xM} = (\mathrm{a} + y_B)\cos\theta + x_B\sin\theta \quad , \quad l_{yM} = x_B\cos\theta - (\mathrm{a} + y_B)\sin\theta \tag{13}$$

$$r_{xM} = r_x\cos\theta - r_y\sin\theta \quad , \quad r_{yM} = r_x\sin\theta + r_y\cos\theta \tag{14}$$

2.3 Dynamic equations of motion

The kinetic and potential energy captured from the bodies can be written as:

$$T = \frac{1}{2}m_V(\dot{x}_G^2 + \dot{y}_G^2) + \frac{1}{2}I_V\dot{\theta}^2 + \frac{1}{2}m_s(\dot{x}_M^2 + \dot{y}_M^2) + \frac{1}{2}I_s(\dot{\theta} - \dot{\varphi})^2 \tag{15}$$

$$U = m_s g y_M + \frac{1}{2}K_V\theta^2 + m_V g y_G \tag{16}$$

- m_V: the mass of the vessel V including the added mass of the water
- m_s: the mass of the body S
- I_V: the moment of inertia of the vessel about O
- I_s: the moment of inertia of the reaction mass about C.
- K_V is the hydrostatic stiffness in pitch (and/or roll) for the vessel about O.

The system presents three degrees of freedom: $\mathbf{r} = \{u, \theta, \varphi\}$. The equations of motion of the system can be derived by the application of the Lagrange principle, which results to:

$$\frac{d}{dt}P_u + R_u\dot{u} = F_w \tag{17}$$

$$\frac{d}{dt}P_\theta + K_v\theta + T_{v\theta} + T_{g\theta} = 0 \tag{18}$$

$$\frac{d}{dt}P_\varphi + T_{g\varphi} = T_p \tag{19}$$

A state space representation for the system of equations is possible under the following form:

$$\begin{Bmatrix} P_u \\ P_\theta \\ P_\varphi \end{Bmatrix} = \begin{bmatrix} M_{uu} & M_{u\theta} & M_{u\varphi} \\ M_{u\theta} & M_{\theta\theta} & M_{\theta\varphi} \\ M_{u\varphi} & M_{\theta\varphi} & M_{\varphi\varphi} \end{bmatrix} \begin{Bmatrix} \dot{u} \\ \dot{\theta} \\ \dot{\varphi} \end{Bmatrix} \quad \equiv \quad \boldsymbol{z_2} = \boldsymbol{M}\,\dot{\boldsymbol{z}}_{\boldsymbol{1}} \;\Rightarrow\; \dot{\boldsymbol{z}}_{\boldsymbol{1}} = \boldsymbol{M}^{-1}\boldsymbol{z_2} \tag{20}$$

$$\dot{\boldsymbol{z}}_{\boldsymbol{2}} = \boldsymbol{f}_R = \begin{bmatrix} F_w - R_u\dot{u} \\ -K_{v\theta} - T_{v\theta} - T_{g\theta} \\ -T_{g\varphi} + T_p \end{bmatrix} \tag{21}$$

$$\begin{aligned} M_{uu} &= m_v + m_s & M_{\theta\theta} &= I_v + I_s + m_v b^2 + m_s\left(l_{xM}^2 + l_{yM}^2\right) \\ M_{u\theta} &= m_v b\cos\theta - m_s l_{xM} & M_{\theta\varphi} &= -\left[I_s + m_s\left(r_{xM}l_{xM} - r_{yM}l_{yM}\right)\right] \\ M_{u\varphi} &= m_s r_{xM} & M_{\varphi\varphi} &= I_s + m_s\left(r_{xM}^2 + r_{yM}^2\right) \end{aligned} \tag{22}$$

The resulting moments due to the gravity are:

$$T_{v\theta} = m_v g b\sin\theta \quad , \quad T_{g\theta} = m_s g l_{ym} \quad , \quad T_{g\varphi} = m_s g r_{ym} \tag{23}$$

The rest of the terms are:

- R_u: an added damping coefficient for the surge motion induced by the waves,
- F_w: the force due to the incident and diffracted waves,
- T_p: the reaction force of the PTO mechanism.

2.4 Equation of motion of the internal inertial reacting body

Under the assumption that the surge and pitch motion of the external vessel are known in the time domain, the equations of motion can be further simplified, retaining only the set of equations which refer to the mechanism itself:

$$\frac{d}{dt}\left(M_{\varphi\varphi}\dot{\varphi}\right) = -\frac{d}{dt}\left(M_{u\varphi}\dot{u} + M_{\theta\varphi}\dot{\theta}\right) - T_{g\varphi} + T_p \tag{24}$$

In an equivalent state space representation:

$$\dot{\boldsymbol{z}} = \begin{bmatrix} \dot{\varphi} \\ \dot{P_{\varphi}} \end{bmatrix} = \begin{bmatrix} \left(P_{\varphi} - M_{u\varphi}\dot{u} - M_{\varphi\varphi}\dot{\varphi}\right)/M_{\varphi\varphi} \\ -T_{g\varphi} + T_p \end{bmatrix} \tag{25}$$

3. Maximum Power Conversion Capability

3.1 Linearization of the equations of motion

Under the assumption of small perturbations around the rest position of the mechanism for the angles φ, θ and ψ of the assembly, the Eqs (8) of motion can thus be simplified as follows:

$$x_B \approx l_p\varphi \quad , \quad y_B \approx 0 \tag{26}$$

$$l_p = (\mu + 1)h \quad , \quad \mu \approx \frac{1}{d/c - 1} \quad , \quad \sigma \approx 2\cos\gamma_o \tag{27}$$

Equation (26) implies that the physical motion of the centre of the mass of the body is linear, exactly in the same way as the traditional designs of linear sliding mass WECs, as for e.g. in the form of PS Frog. Similar simplified relations hold for the factors r_x, r_y, l_{x_M}, l_{y_M}, r_{x_M} and r_{y_M}, as well as for the components of the matrix M:

$$\boldsymbol{M} = \begin{bmatrix} m_v & m_v b & m_s l_p \\ m_v b & I_v + I_s + m_v b^2 & -I_\theta \\ m_s l_p & -I_\theta & I_\varphi \end{bmatrix} \tag{28}$$

$$I_\theta = I_s + m_s l_p \mathrm{a} \quad , \quad I_\varphi = I_s + m_s l_p^2 \tag{29}$$

and the moments due to the gravity:

$$T_{v\theta} \approx 0 \quad , \quad T_{g\theta} \approx m_s g l_p \varphi \quad , \quad T_{g\varphi} \approx m_s g l_p \theta \tag{30}$$

3.2 Proposed form for the Power Take Off force and Feedback Law

In view of the non-linear equation of motion, Eq (24), the mechanism is inherent to an unstable behaviour. For this reason, a feedback law is incorporated in the power take off force, being of the following form:

$$T_p = -K_p\varphi - R_p\dot{\varphi} + T_N \tag{31}$$

where K_p and R_p are constant linear feedback gains to be properly selected and T_N denotes an appropriate compensator for the non-linearity of the system in the form:

$$T_N = \frac{d}{dt}\left(P_\varphi - m_s l_p \dot{u} + I_\theta\dot{\theta} - I_\varphi\dot{\varphi}\right) + \left(T_{g\varphi} - m_s g l_p \theta\right) \tag{32}$$

which results to the following equation for motion of the internal body:

$$I_\varphi\ddot{\varphi} + R_p\dot{\varphi} + K_p\varphi = -m_s l_p \ddot{u} + I_\theta\ddot{\theta} - m_s g l_p\theta + T_N \tag{33}$$

Obviously T_N is equal to zero for a linearized system.

Equation (33) implies that the motion of the internal body is fully equivalent dynamically to that of a damped physical pendulum, with a mass m_s and inertia I_s about its centre of mass, which is suspended at a distance l_p from its centre of mass.

However, it should be stretched, that in view of Eqs (27), the equivalent length l_p of this pendulum can be many orders of magnitude higher than that expected by any other vertical pendulum, realized in the traditional natural technological way, as for e.g. in the form of SEAREV.

This pendulum can simultaneously convert three different forms of wave energy:

- The kinetic energy resulting from the surge motion.
- The kinetic energy resulting from the pitching motion.
- The potential energy resulting from the pitching motion.

In view of Eq. (33), the selection of the feedback gains can be performed appropriately to ensure stability of the system, optimal tuning of the natural periods of the system to the periods of the excitation, as well as maximum power conversion capability.

3.3 Calculation of maximum power conversion capacity

The analysis of the power conversion capability can be performed independently for the surge and pitch motion of the converter. However, the design of the external vessel and the coupled form of equations [9] imply that dependence exists in fact between them. Detailed analysis of such dependence is performed in [10]. Following the outline of such an analysis, the vessel will be assumed to be subjected to a pitching motion of amplitude Θ_C and of frequency ω:

$$\theta(t) = \Theta_C \cos \omega t \tag{34}$$

The surge motion depends on the pitch motion as follows:

$$u(t) = -b\theta(t) = -b\Theta_C \cos \omega t \tag{35}$$

As a result, the equation of motion (33) now becomes:

$$I_\varphi \ddot{\varphi} + R_p \dot{\varphi} + K_p \varphi = -M_e \Theta_C \cos \omega t \tag{36}$$

$$T_p = -K_p \varphi - R_p \dot{\varphi} \quad , \quad M_e = \omega^2 I_P + m_s g l_p \quad , \quad I_P = I_S + m_s l_p (\mathrm{a} + b) \tag{37}$$

The steady state response of the system is a harmonic function with a frequency equal to ω and with a phase difference of π/2 with the excitation force, in order to maximize power capture from the excitation force:

$$\varphi(t) = -\Phi_S \sin \omega t \tag{38}$$

The minus sign is used to denote that for positive θ angle, a negative φ angle should result, in order to ensure the stability of the vessel. The mean power absorbed by the Power Take Off is defined as follows:

$$P_{out} = \frac{1}{T_w} \int_0^{T_w} T_p \dot{\varphi} dt = -\frac{1}{2} \omega M_e \Theta_c \Phi_s = -\frac{1}{2} \omega \Theta_c m_s X_M a_e - P_{in} \tag{39}$$

where $X_M = l_p \Phi_s$ is the amplitude of the linear motion of the oscillating body's CoM and:

$$a_e = g + \omega^2 (\mathrm{a} + b + l_I) \quad , \quad l_I = I_S / (m_s l_p) \tag{40}$$

Substitution of Eqs (37) into Eq (40) leads to the following results for the PTO:

$$R_p = \frac{M_e \Theta_c}{\omega \Phi_s} \quad , \quad K_p = \omega^2 I_\varphi \tag{41}$$

4. Indicative Implementation - Standalone 0.25-2 MW rated power WEC

Typical values for "good" offshore locations, range between 20 and 70 kW/m as annual average and occur mostly in moderate to high latitudes. A design approach for a peak energy level of 40 kW/m can be reasonably used as a target value to be reached by the subsequent WECs to be designed.

The actual power that can be absorbed by a pitching and surging WEC is expressed by the value of the Capture Width, which is around L=λ/π, for pitching and surging WECs [2]. Typical values for the wavelength of $\lambda \approx 20 \div 160$ m are considered. Calculating L and multiplying this value with the wave power per unit length [MW/m], results to a total value of a power around 0.25-2.00 MW [8], which can be absorbed by a unique WEC. Moreover, assuming the buoy to be of a hemispherical shape, estimates for the optimum radius of the vessel $\mathrm{D_V}/2 = \mathrm{R_V} = 0.262\ \mathrm{T_w^2}$ can be derived [2], although this value has been proved for heaving motion only. Taking into account wave periods of

6-10 seconds, this leads to a buoy with a radius of at least 10-26m, able to capture the amount of power calculated previously. The above estimates are in line with the results of [11].

Therefore, an indicative mechanism for a standalone WEC is presented. A body consisting of two unequal spheres and a beam that links them together will be used as an inertial mass. This body is suspended with three links inside a sealed vessel. This basic configuration has the form presented in Fig. 3. It should be noted, that hydraulic power take off systems can offer a reliable and efficient approach for wave energy conversion [12]. As Fig. 3 indicates, the vessel is a fully sealed hull with a plate at the bottom for maximizing reaction, increasing the added mass and lowering the centre of mass of the external vessel. The hydraulic system power pack can be placed at the bottom, while the rams operate in the same plane with the centre of mass of the oscillating body. For the detailed design of the hull, efficient procedures for its design can be applied [13], [14].

Figure 3: A fully enclosed multi axis combined surge and pitch WEC.

In order to compute and present the main parameters of the structure for various values of the desired output power, using the linearized equations, the following parameters are considered constant throughout this examination:

$$d = 10\ [\mathrm{m}] \quad c = 8\ [\mathrm{m}] \quad a = 5\ [\mathrm{m}] \quad b = 2.5\ [\mathrm{m}] \quad T_w = 8\ [\mathrm{s}] \quad \Theta_c = 25\ [°] \quad \Phi_s = 5\ [°]$$

Table 1: Design parameters of a standalone 0.25 – 2.00 MW WEC for both pitch and surge excitation.

Four-bar mechanism					
P_{out} [MW]	h [m]	γ_0 [°]	l [m]	μ	l_p [m]
0.25	11.00	79.70	11.18	4.00	55.00
0.50	14.00	81.87	14.14	4.00	70.00
0.75	15.50	82.65	15.63	4.00	77.50
1.00	17.00	83.29	17.12	4.00	85.00
1.50	20.00	84.29	20.10	4.00	100.00
2.00	21.50	84.69	21.59	4.00	107.50
Body & Vessel					
P_{out} [MW]	m_1 [tn]	m_2 [tn]	r_1 [m]	r_2 [m]	I_s [kgm²]
0.25	15.00	5.00	4.00	12.00	1.46E+06
0.50	20.00	10.00	8.50	17.00	5.09E+06
0.75	25.00	15.00	10.80	18.00	8.78E+06
1.00	35.00	15.00	8.14	19.00	8.99E+06
1.50	50.00	15.00	6.60	22.00	1.11E+07
2.00	60.00	20.00	7.67	23.00	1.61E+07
Response & PTO Parameters					
P_{out} [MW]	M_e [kgm²/s²]	X_M [m]	R_p	K_p	
0.25	1.68E+07	4.80	1.07E+08	3.82E+07	
0.50	3.35E+07	6.11	2.13E+08	9.38E+07	
0.75	5.02E+07	6.76	3.19E+08	1.54E+08	
1.00	6.69E+07	7.42	4.26E+08	2.28E+08	
1.50	1.01E+08	8.73	6.41E+08	4.08E+08	
2.00	1.34E+08	9.38	8.54E+08	5.80E+08	

For this task, a preliminary set of geometrical restrictions has been set, by examining the configuration of the system: primarily $d > c$ and $h < r_2$. Assuming that the weight of the beam and its supporting brackets are evenly distributed along its total length, the values of m_1, r_1 and m_2, r_2 can be calculated, where m_1 and m_2 indicate the masses of the two spheres and r_1, r_2 their distance measured from the centre of mass C of the body ("pendulum") S. Considering that $m_s=m_1+m_2$ and $m_1/m_2=r_2/r_1$, the moment of inertia of the body S:

$$I_{S_C} = m_s r_1 r_2 \quad \Rightarrow \quad I_S \equiv I_{S_O} = m_s(r_1 r_2 + a^2) \tag{42}$$

It should be clarified, that the above values refer just to an indicative implementation of a mechanism for a WEC and they are by no means optimized. Such an approach is obviously necessary in full association to the design of an optimized external vessel.

5. Conclusion

As it results, the linear motion of the centre of mass of the suspended body and the suspension geometry enables the introduction of a quite simple form of a PTO mechanism, rendering this design far more reliable and easily implementable than all other known types of internally reacting masses. The combined values of power and suspended mass in Table 1 compare more than favourably to those necessary for other types of internally reacting WECs, such as PSFrog [9] or SEAREV [8], [12] or other technologies [15]. The concept is flexible and parametrically designed, enabling its implementation in any form of floating vessels. An alternative direction consists in properly embedding it in floating offshore platforms, supporting wind turbines. Such a design can drastically enhance the performance, the efficiency and the potential of floating offshore energy applications.

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