Experimental testing of mooring systems for wave energy converters

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Abstract

Given the currently growing concerns for the fossil fuels shortage and the environmental consequences of their excessive consumption, new alternative renewable energy technologies have to be developed and implemented. The sea is a very promising one, yet to be efficiently explored.

The main purpose of the present work was to perform experimental tests on 1:40 scale-models of two distinct mooring configurations (catenary shape and the weight-buoy taut cable), with the purpose of measuring the loads applied to the superior tip of the line (fair-lead). Both mooring configurations were tested under regular oscillating movements on the surge and heave directions in order to validate a finite element code for numerical simulation of mooring systems dynamics for Wave Energy Converters.

The following project contributed to a better understanding of the dynamics of the tested mooring configurations. Considering only the experimental situations, it was concluded that the load amplitude variations are much more intense for horizontal oscillation, with considerable peaks registered for the catenary configuration. On the other hand, mean tension on weight-buoy mooring line is higher, even for milder conditions. From the comparison between numerical and experimental results, it was concluded that the program predicts mean loads with some discrepancies with the experimental results, due to inaccurate determination of extremely influential parameters on moorings behaviour. For the maximum amplitude of the forces, the results from heave simulations showed acceptable concordance with the experimental ones, having however presented accentuated overestimation for the surge direction, needing a wider set of comparing results for solid conclusions.

Keywords: moorings, WEC, wave energy, catenary, weight-buoy, numeric

1. Introduction

In modern times, mankind has been facing the urgent need to ensure viable alternatives to energy consumption from fossil fuels. In this context the sea presents itself as a promising alternative, not only for being renewable and clean but also for its vastness. Of all the energy potential of the oceans, this paper intends to focus on wave energy. The wave energy converters, WECs, can be either fixed or floating and operate alone or coupled with other similar, forming a farm. The conversion of energy is possible due to the movement with respect to the seabed. The devices are attached to the sea-bottom through the use of moorings, whose project is an important and complex step in the design of these systems, given the need to understand the static and dynamic phenomena that govern their behaviour and how it affects the performance of the converters. Similarly to other conventional floating offshore structures, the primary objective of the mooring system is to ensure proper positioning of the device under normal and extreme sea conditions, avoiding drift caused by waves, currents and winds. Good fatigue strengths and wear properties are therefore necessary characteristics of the mooring system, which usually translates to a substantial portion of the total cost of floating WEC systems [1].

1.1. Concepts

Considering the energy extracting mode, the WEC devices can be categorized as follows, for which the primarily extraction energy movement can be diverse [2]:

- Oscillating Water Column, OWC - Examples: Spar Buoy, Mighty Whale, Sloped Buoy, etc.
- Wave Activated Bodies - Examples: IPS Buoy, Wavebob, etc.
- Overtopping devices - Examples: Wave Dragon.

Moorings systems consist basically of two elements: mooring line and anchor, with the possibility of being included other components such as buoys and weights to satisfy certain requirements. These can be characterized about the influence they have on the devices, as follows:

- Passive - These systems just keep the device in the operating position.
- Active - The moorings influence the converters dynamic response and therefore their ability to extract energy.
- Reactive - In this case the moorings are an integral part of the system and may be the link between the float and the energy extraction device.
The main function of a mooring system is to ensure the correct positioning of the devices under normal and extreme sea conditions. However, these systems must meet another set of requirements [1], among which are highlighted some of the most important:

- Limit the movement of the device in order to avoid stresses on electric cables;
- Keep the distance between converters arranged in farm configurations;
- Unless the rigidity of the mooring act as an active element in the energy conversion principle (active mooring systems), the mooring system must be designed to minimize the forces applied to the anchors, mooring lines and device;
- All system components must have appropriate characteristics of resistance, fatigue and durability over the devices operating life;
- Should be avoided any contact between mooring lines.

1.2. Developed work

The development of numerical simulation tools and experimental studies have contributed to a better understanding of the topics regarding mooring systems. Harris et al. [1] present a background on moorings of WECs, addressing their influence on the behaviour of floating systems. In this work, an overview of the existing mooring configurations is performed, as well as of the anchoring systems and complementary elements and its applicability to these devices.

The authors conclude that most of the policies used in choosing moorings for floating platforms can be followed to ensure safe placement of WEC systems. However, its influence on the response of the devices should be carefully studied to ensure efficient conversion of energy. The work carried out by Yang et al. [3] presents a numerical approach to mooring analysis for WEC, highlighting the issue of fatigue in these parts. The authors compared the results of simulations performed with an internally developed program, MOODY, with the results of the commercial software DNV DeepC. Four procedures were followed for the simulations (see 1), which shows that two different codes were used to simulate the movement of the float - DeepC-RAO and DeepC-Time - whereas to simulate the dynamic line response were used MOODY and DeepC-Cable codes.

The authors concluded that in spite of the decoupled approach produce less accurate predictions, the results of the fatigue study were satisfactory. Thus, the numerical robustness of DNV DeepC software combined with a reduced need for preparation of the uncoupled model make the TI procedure the most advisable one.

2. Numerical model

The numerical simulation program to be validated in the present work, was created by Luís Gerald Oliveira and performs static and dynamics analysis of mooring systems for isolated WECs or in farm configuration.

It is suitable to evaluate different line shapes and materials, allowing the addition of buoys and weights, performing non-linear analysis accounting for the geometric stiffness of the mooring line and the drag of the components including the cable itself.

2.1. Mathematical approach

The equation of motion for a system of rigid bodies, can be expressed as follows

\[ M\ddot{x} + C\dot{x} + Kx = F, \]

where \( M \) is the mass matrix, \( C \) is the damping matrix and \( K \) the stiffness. The external forces vector is given by \( F \), \( x \), is the displacements vector, \( \dot{x} \), is the velocities vector and \( \ddot{x} \), is the accelerations vector.

From instant \( t \) to \( t + \Delta t \), the previous equation is modified resulting in:

\[ M\ddot{x} + C\dot{x} + K\Delta x = F - I, \]

for which \( I \) is the internal forces vector.

To reduce the simulation run time, for a given \( t + \Delta t \), the matrices \( M \), \( C \), \( K \) and the vector \( I \), are computed at time \( t \), and \( F \), \( \dot{x} \), \( x \) and \( \Delta x \) are computed at time \( t + \Delta t \).

These equations are solved for mooring and buoy coupled system, therefore the presented matrices assemble both moorings and buoy matrices for mass, damping and stiffness. The solution of the differential equations systems is performed using RungeKutta-Nystrom method.

2.2. Catenary element

The code uses a catenary element type (see Figure 1). \( H \) and \( V \) are, respectively, the horizontal and vertical forces applied to the tip node.

![Catenary cable element hanging freely and lying on the seabed.](image)

Figure 1: Catenary cable element hanging freely and lying on the seabed.

The distances \( dx \) and \( dy \), relate to \( V \) and \( H \), for a free hanging element as follows

\[ \Delta x = \frac{H}{w} \left[ \sinh^{-1} \left( \frac{V}{H} \right) - \sinh^{-1} \left( \frac{V - wL}{H} \right) \right] + \frac{HL}{EA}, \]

\[ \Delta y = \frac{H}{w} \left[ \sqrt{1 + \left( \frac{V}{H} \right)^2} - \sqrt{1 + \left( \frac{V - wL}{H} \right)^2} \right] + \frac{1}{EA} \left( VL - \frac{wL^2}{2} \right). \]
and for a partially lying element, as

\[
\Delta x = L - \frac{V}{w} + \frac{H}{w} \left[ \sinh^{-1}\left( \frac{V}{H} \right) \right] + \frac{HL}{EA}, \tag{5}
\]

\[
\Delta y = \frac{H}{w} \left[ \sqrt{1 + \left( \frac{V}{H} \right)^2} - 1 \right] + \frac{V^2}{2EAw}, \tag{6}
\]

where \( L \) is the total length of the cable, \( w \) is the line submerged weight of the line and \( EA \) is its axial stiffness.

### 2.3. Matrix rigidez

For each element the mass matrix is given by

\[
K_F = \frac{1}{\Delta K} \begin{bmatrix}
\frac{\partial \Delta y}{\partial V} & \frac{\partial \Delta y}{\partial H} \\
\frac{\partial \Delta x}{\partial H} & \frac{\partial \Delta x}{\partial V}
\end{bmatrix}, \tag{7}
\]

where

\[
\Delta K = \frac{\partial \Delta y}{\partial V} \frac{\partial \Delta x}{\partial H} - \frac{\partial \Delta y}{\partial H} \frac{\partial \Delta x}{\partial V}, \tag{8}
\]

is the determinant of matrix \( K_F \). Thus, for a two node element with two degrees of freedom, horizontal and vertical displacement, the stiffness matrix is

\[
K_T = \begin{bmatrix}
K_F & -K_F \\
-K_F & K_F
\end{bmatrix}. \tag{9}
\]

Adding a new degree of freedom, about an axis perpendicular to a plane containing the mooring, \( K_T \) is equal to:

\[
K_T = \begin{bmatrix}
K_{F_{11}} & -K_{F_{12}} & 0 & -K_{F_{11}} & K_{F_{12}} & 0 \\
-K_{F_{12}} & K_{F_{11}} & 0 & -K_{F_{12}} & K_{F_{11}} & 0 \\
0 & 0 & 0 & 0 & 0 & 0 \\
K_{F_{21}} & -K_{F_{21}} & 0 & K_{F_{21}} & -K_{F_{21}} & 0 \\
-K_{F_{22}} & K_{F_{22}} & 0 & -K_{F_{22}} & K_{F_{22}} & 0
\end{bmatrix}. \tag{10}
\]

### 2.4. Mass matrix

In this program the elements mass is considered to be lumped in nodes. For every element the individual mass matrix has the form:

\[
m_c = \begin{bmatrix}
\frac{wL}{2} & 0 & 0 \\
0 & \frac{wL}{2} & 0 \\
0 & 0 & \frac{wL^3}{24}
\end{bmatrix}. \tag{11}
\]

### 2.5. Cable drag forces

The forces on the cables are determined using Morison’s equation, which has the following form

\[
F_C = \rho C_M \frac{\pi d^4}{4} L (\dot{x}_W - \ddot{x}) + \frac{1}{2} \rho L \pi d C_D (\dot{x}_W - \ddot{x}) [\dot{x}_W - \ddot{x}], \tag{12}
\]

where \( C_M \) is the inertia coefficient of a cylinder, \( C_D \) is the drag coefficient of a chain with diameter, \( d \), and a length \( L \). The terms \( \dot{x}_W \) and \( \ddot{x}_W \) represent the velocity and the acceleration of water particles (in the absence of the chain). \( \dot{x} \) and \( \ddot{x} \) are the velocity and acceleration of a chain point.

The drag forces for buoys and weights can be determined using Equation 12, as well. The added mass term of the Equation 12 is:

\[
m_{ac} = C_M \begin{bmatrix}
\frac{\rho \pi LD^2}{8} & 0 & 0 \\
0 & \frac{\rho \pi LD^2}{8} & 0 \\
0 & 0 & \frac{\rho \pi L^3 d^2}{96}
\end{bmatrix}. \tag{13}
\]

### 3. Experimental tests

#### 3.1. Description

As mentioned before, the experimental tests were conducted separately for surge and heave directions, for 1:40 scaling-models, in a channel with 1.0 m width, 1.5 m deep and a length much longer than the axial length of the moorings. In order to impose the oscillating movement to the fair-lead, an electromagnetic linear motor was used and two different structures created to provide the steady fixing of this device, both for horizontal and vertical movements, see Figure 2. The motor consists of a magnet forced to move due to electrical field variations produced by a stator. The constituent materials of the structures were steel, for its mechanical properties and availability and aluminum when magnetic forces needed to be avoided.
The experimental installation included a set of electronic devices to control the motor within safe conditions for user and equipment. Figure 3 shows schematically the experimental system.

In order to measure the loads applied to the line, a bi-axial transducer was attached to the fairlead. Due to the reduced output signal registered, it was necessary to perform an amplifying step before the acquisition of the final signal by a data acquisition card running with resource to real time software XpcTarget.

3.2. Mooring configurations

The configurations tested comprised two distinct types of moorings. One, generally referred as catenary chain, consist of a simple steel chain hanging from the fair-lead to the anchor with a considerable portion lying on the bottom of the channel, see Figure 4. The other, being a taut steel cable with compliance of a weight and a buoy without any contact with the ground, see Figure 5. These two configuration have different characteristic, given that the restoring force for the catenary is mainly due to its weight and friction force with the ground (for this configuration two different chains were tested). In turn, the weight-buoy mooring relies on the impulsion force of the buoy and the gravitic force of the weight, as restoring force. The characteristics of the tested chains are presented in Table 2.

**Table 3:** Tested cable characteristics.

<table>
<thead>
<tr>
<th>Type</th>
<th>L [m]</th>
<th>d [mm]</th>
<th>EA [kN]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cable 2-3</td>
<td>1.243</td>
<td>1.0</td>
<td>45</td>
</tr>
<tr>
<td>Cable 3-4</td>
<td>1.143</td>
<td>1.0</td>
<td>45</td>
</tr>
<tr>
<td>Cable 4-5</td>
<td>4.523</td>
<td>1.0</td>
<td>45</td>
</tr>
</tbody>
</table>

$L$ and $d$ values were directly measured and $EA$ was determined using the following equation: $EA = 45000d^2$.

The mooring fixation to the ground was accomplished by means of a steel block with a mass of 16.5 kg.

3.3. Tests performed

The control of the linear motor was done using Matlab Simulink, through which the user inserted the characteristics of a sinusoidal signal of the form

$$y = A \sin(\omega t),$$

where $A$ is the amplitude of the wave to be recreated, $y$ is the position of the fair-lead and $\omega$ is the angular frequency, related to the period $T$, by:

$$\omega = \frac{2\pi}{T}.$$
### Table 2: Tested chains characteristics.

<table>
<thead>
<tr>
<th></th>
<th>Chain 1</th>
<th>Chain 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Link diameter - (d) [mm]</td>
<td>3.0</td>
<td>2.0</td>
</tr>
<tr>
<td>Axial length - (L) [m]</td>
<td>7.00</td>
<td>7.00</td>
</tr>
<tr>
<td>Submerged chain weight per unit length - (w) [N/m]</td>
<td>1.35</td>
<td>0.53</td>
</tr>
<tr>
<td>Axial stiffness - (EA) [kN]</td>
<td>850</td>
<td>360</td>
</tr>
</tbody>
</table>

It is necessary to reveal that these values had to be recalculated for the model scale, see Table 5. Concerning the amplitudes, because of some technical limitations the maximum value that could be achieved corresponded to a real value of 2.44 m, see Table 6. The tests had the duration of 120 s and the sampling time of the analog signal was set at 1 ms.

### Table 5: Tested periods.

<table>
<thead>
<tr>
<th>Period [s]</th>
<th>Model scale</th>
<th>Real Scale</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.60</td>
<td>3.79</td>
<td></td>
</tr>
<tr>
<td>0.90</td>
<td>5.69</td>
<td></td>
</tr>
<tr>
<td>1.20</td>
<td>7.59</td>
<td></td>
</tr>
<tr>
<td>1.50</td>
<td>9.49</td>
<td></td>
</tr>
<tr>
<td>1.80</td>
<td>11.38</td>
<td></td>
</tr>
<tr>
<td>2.30</td>
<td>14.55</td>
<td></td>
</tr>
<tr>
<td>2.80</td>
<td>17.71</td>
<td></td>
</tr>
</tbody>
</table>

### Table 6: Tested amplitudes.

<table>
<thead>
<tr>
<th>Amplitude [cm]</th>
<th>Model scale [cm]</th>
<th>Real scale [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.20</td>
<td>0.480</td>
<td>0.048</td>
</tr>
<tr>
<td>2.40</td>
<td>0.960</td>
<td>0.096</td>
</tr>
<tr>
<td>4.50</td>
<td>1.800</td>
<td>0.180</td>
</tr>
<tr>
<td>5.60</td>
<td>2.240</td>
<td>0.224</td>
</tr>
<tr>
<td>6.10</td>
<td>2.440</td>
<td>0.244</td>
</tr>
</tbody>
</table>

### 4. Experiment results

#### 4.1. Signal filtering

Despite all the precautions taken in order to reduce the noise present in the transducer’s output signal, post-processing filtering was necessary. The method used was the well known Savitsky-Golay in which the data is smoothed fitting a polynomial to a set of \(n_p\) \((x_i, y_i)\) points within a certain window, using the method of the minimum least squares. The used polynomial has the following form [6]

\[
p(x) = \frac{C_6 x^6 + C_4 x^4 + C_2 x^2 + C_0}{D},
\]

where the coefficients \(C_6, C_4, C_2, C_0\) and \(D\) are functions of the number of points within the window, \(n_p\), given by:

\[
C_6 = -15015, \quad (19)
\]

\[
C_4 = \frac{24255n_p^2 - 347655}{4}, \quad (20)
\]

\[
C_2 = \frac{-11025n_p^4 + 330750n_p^2 - 150745}{16}, \quad (21)
\]

\[
C_0 = \frac{1225n_p^6 - 57575n_p^4 + 605395n_p^2 - 952245}{64}, \quad (22)
\]

\[
D = 4n_p(n_p^6 - 56n_p^4 + 784n_p^2 - 2304). \quad (23)
\]

Given that a single filter is unable to eliminate all the unwanted frequencies present within the signal, there were conducted sequential filtering steps to the same signal, varying \(n_p\). The set of number of points used for experimental tests with conditions of \(T = 2.3\) s was \(n_p \in \{295, 281, 267, 253, 239, 227\}\).

The recorded signal and the final result after filtering, for a test with \(A = 5.6\) cm and \(T = 2.3\) cm, are shown in the Figure 6.

#### 4.2. Results and discussion

As observed, the surge and heave tests were not performed within the same conditions. This is easily verified considering that the mean tension for both cases is not equal. However, the installation and mooring parameters were invariable for the same type of test (horizontal or vertical) and therefore within the same direction, the results could be compared. The analysis focused on the mean tension applied to the line, \(F_m\) and on the maximum amplitude of the tension, \(F_{max}\) given by:

\[
F_{max} = \max(F - F_m), \quad (24)
\]
The deviations were calculated as follows

\[ \text{Deviation} = \left| \frac{x_{\text{ref}} - x}{x_{\text{ref}}} \right| \times 100\%, \]  

(25)

where \( x_{\text{ref}} \) represents the value for the standard case and \( x \) represents the value for the comparative case. \( F_m \) and \( F_{\text{max}} \) were determined in the interval of 95 s to 115 s of the test, in order to avoid the proximity to the initial transients.

The results analysis was divided into three categories:

- sensitive analysis based on experimental results only;
- direct comparison between mooring configurations;
- direct comparison between numerical and experimental results.

The first approach consisted of varying individually quantities as the amplitude of the signal, \( A \), the period of the signal, \( T \), the distance between the fair-lead and the anchor, \( D_h \), the total line length, \( L \) and the submerged chain weight, \( w \), for a specific case, in order to understand how these parameters influenced the values of the forces.

For that, a standard case was chosen, with the following conditions:

- Mooring type - catenary chain
- Amplitude, \( A = 5.6 \) cm.
- Period, \( T = 2.3 \) s.
- Total line length, \( L = 7.00 \) m.
- Horizontal distance from fair-lead to anchor, \( D_h = 6.60 \) m.
- Vertical distance from fair-lead to anchor, \( h = 1.30 \) m.
- Submerged weight per unit length, \( w = 0.138 \) kgf/m.
- Axial stiffness, \( EA = 850 \) kN.
- Chain diameter, \( d = 3.0 \) mm.

One of the most important findings was related with \( D_h \) and \( L \). For the catenary configuration under surge conditions, was verified that a reduction of 1.6% of \( L \), can reflect an augmentation of 222.2% on the maximum amplitude of the load, \( F_{\text{max}} \), and an increase of 25% on the mean load value, \( F_m \), see Figure 7. Because of an experimental problem it was not possible to run the vertical tests for this case.

For \( D_h \), given a 15.2% reduction, \( F_{\text{max}} \) is increased by 375.5%, while the value of \( F_m \) rises 49.4%, see Figure 8. Although \( F_{\text{max}} \) variations were not so considerable to vertical displacement (85.4%), \( F_m \) was increased by 60.9%, see Figure 8.

Knowing this, the accurate determination of these parameters assumes greater importance, since very reduced variations can produce considerable change in the measured loads.

The weight-buoy configuration showed less sensitive to \( D_h \) in contrast with the catenary. Nevertheless, for the same conditions as the previous, \( F_m \) values are in general higher and \( F_{\text{max}} \) values are considerably lower as can be
seen in Figure 10.

Respecting the comparison between numerical and experimental results, some deviations were observed, see Figure 11 and Figure 12. Clearly, there is acceptable concordance for results under heave conditions. The deviations registered are likely related to imprecisions on determination of \( D_h \). However, for surge direction, the difference observed for \( F_{max} \) is clearly out of the pattern. One plausible explanation for this value might be due to an inappropriate response from the structure to attach the motor, to higher loads. Under certain conditions some parts may slightly bend leading to erroneous measurements.

\[ D_h = 6.635 \text{ m}, \] represented a reduction to 2.9\% in \( T_m \) for horizontal motion.

The most important results were presented, however other parameters were studied and determined their effects on \( F_m \) and \( F_{max} \), which are qualitatively presented in Table 8.

5. Conclusions

The experimental tests provided a considerable set of results from which some important conclusions were taken. Respectively to the experimental results only, for the weight-buoy configuration, it was concluded that the mean tension is, for the majority of cases, higher than for the catenary mooring, even though the latter is subjected to more intense forces during the excursion. For the finite element code, some simulations showed acceptable concordance with the experimental results, mainly for vertical motion. However, some discrepancies were observed for the surge direction, which would require more comparative cases in order to be able to draw solid conclusions about the program accuracy for this type of movement. Some of the deviations were due to inaccurate determination of some parameters with strong influence on the forces applied to the mooring line. Future work could be focused on recreating a wider set of experimental tests comprising more variate conditions (e.g. irregular oscillations, oblique motion, orbital motion) for numerical comparison.

6. References

Table 7: Comparison of $F_m$ and $F_{max}$ for numerical and experimental simulations.

<table>
<thead>
<tr>
<th></th>
<th>Surge</th>
<th>Heave</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$F_m$ [kgf]</td>
<td>$F_{max}$ [kgf]</td>
</tr>
<tr>
<td>Experimental</td>
<td>0.659</td>
<td>0.054</td>
</tr>
<tr>
<td>Numeric</td>
<td>0.559</td>
<td>0.109</td>
</tr>
<tr>
<td>Deviation [%]</td>
<td>15.1</td>
<td>101.9</td>
</tr>
</tbody>
</table>

Table 8: Resume of the qualitative influence of the studied parameters on mooring forces (assuming an increase in the parameters values). Upwards and downwards arrows represent an increase or a decrease, respectively. Horizontal arrows show a negligible variation of the forces. Horizontal dashes stand for non tested conditions.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Catenary</th>
<th>Weight-buoy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amplitude, $A$</td>
<td>-</td>
<td>↑</td>
</tr>
<tr>
<td>Period - $T$</td>
<td>-</td>
<td>↓</td>
</tr>
<tr>
<td>Horizontal distance - $D_h$</td>
<td>↑↑</td>
<td>↑↑↑↑↑↑</td>
</tr>
<tr>
<td>Total line length - $L$</td>
<td>↓↓</td>
<td>↓↓↓↓</td>
</tr>
<tr>
<td>Submerged line weight - $w$</td>
<td>↑</td>
<td>↑</td>
</tr>
<tr>
<td>Axial stiffness - $EA$</td>
<td>→</td>
<td>-</td>
</tr>
</tbody>
</table>