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Experimental and numerical investigation of a taut-moored wave energy converter—a validation of simulated buoy motions

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Abstract

This study presents an experimental and numerical investigation of a taut-moored wave energy converter (WEC) system with a point absorber type of WEC. The WEC system consists of a buoy, a unique three-leg two-segment mooring system with submerged floaters, and a power take-off system designed for the current experiment as a heave plate. The main objective of the study is to validate a numerical simulation model against experiments carried out in an ocean basin laboratory. Two physical models in model scales 1:20 and 1:36 were built and tested. The detailed experimental testing program encompasses tests of mooring system stiffness, decay tests, and different sea state conditions for ocean current, regular and irregular waves. A numerical model in the model scale 1:20 was developed to simulate coupled hydrodynamic and structural response analyses of the WEC system, primarily using potential flow theory, boundary element method, finite element method, and the Morison equation. Several numerical simulations are presented for each part of the experimental testing program. Results for

the WEC buoy motions under operational conditions from the experiments and the numerical simulations were compared. This study shows that the simulation model can satisfactorily predict the dynamic motion responses of the WEC system at non-resonant conditions while at resonant conditions additional calibration is needed to capture the damping present during the experiment. A discussion on simulation model calibration with regard to linear and non-linear damping highlights the challenge to estimate these damping values if measurement data are not available.

Keywords

Experiment, model validation, point absorber, simulation, taut mooring, wave energy converter.

1 Introduction

Ocean energy as one source of renewable energy has great potential. It is becoming more and more attractive to explore and utilise since the world's energy demand is increasing and we must reduce the use of e.g. fossil fuels in the energy production. A large number of studies has presented novel designs, new numerical models, and field tests of different wave energy converter (WEC) systems which yield a true sustainable source of energy (such as those reviewed in¹⁻³). This far, however, only a few of them are close to reaching the phase of full scale commercialization. Going from concept to commercial product requires building and testing of prototypes of various scales. It is in this area where validated computational models provide profound values to both research and development of WEC technologies. Guidelines for experimental tank testing of WEC systems have been addressed in^{1,4-7}. As addressed in the guidelines, the size, cost, and complexity of full-scale devices motivate cost-effective development and solutions through numerical modelling and scaled testing in ocean basins. The advantage with ocean basin tests is that they enable detailed investigations of the responses of the WEC system, i.e. the buoy with the mooring system, under controlled environment conditions (sea state and ocean current) where they can be varied systematically.

Numerous ocean basin tests of WEC systems have been conducted for various purposes, such as for proof of concept,⁸⁻¹¹ for power performance evaluation,^{12,13} for parametric analysis,¹⁴⁻¹⁶ for hydrodynamic response analysis,^{12,16-18} and for structural and mooring forces assessments.^{13,17-19} The current study aims to validate a previously developed numerical model presented in²⁰ against an extensive testing program on a scale model of a WEC prototype in an ocean basin laboratory. Table 1 shows a summary of studies in the literature where validation of numerical models against experimental tests are presented. In contrast to the studies presented in Table 1, the current investigation contributes to the research area within two areas. First, although the taut mooring system has shown to have a high potential when used in WEC systems,²⁹⁻³⁰ it has only been stand-alone tested in wave basins^{18,31} and needs to be tested and validated in a complete WEC system where a power take-off (PTO) is present. Secondly, the dynamic response

of a WEC system due to wave and ocean current load interaction is yet to be tested. It has been found in³²⁻³⁴ that this load interaction is an important factor in the analysis and assessment of realistic load scenarios acting on WECs.

Table 1. Summary of a literature review of studies which include laboratory tests of WEC systems and validation of numerical models.

Reference	System configuration	Mooring	PTO	Test environment and characteristics	Contribution of the experiment and validated model
21	Sloped WEC	- ^a	Dynamometer	1. Curved wave tank 2. Regular waves	Showed that the linear boundary-element method is capable of computing the general trend of a sloped WEC device.
22	Overtopping WEC	Catenary, while no further detail was provided	- ^a	1. Wave basin 2. Irregular waves	Advanced the model development of wave-to-wire and overtopping analysis specifically for Wave Dragon.
23	Oscillating water column WEC (OWC WEC)	Catenary chain	Orifice	1. Wave tank 2. Regular waves	Established a numerical model to perform hydrodynamic and thermodynamic analyses of OWC WECs.
12	Two-body heave-constrained point absorber	- ^a	Feedback-controlled linear actuator	1. Two-dimension wave tank 2. Regular waves 3. Two bodies of WEC were tested while holding the other body fixed	Enabled the assessment of effect of relative motion constraints and PTO control strategies on power production and heave dynamic motion.
19	A buoy used to represent WEC type such as point absorber	Catenary	- ^a	1. Deep water wave basin 2. Regular and irregular waves 3. Operational condition 4. In comparison with other studies, large model scale of 1:5 was tested	Enabled fully coupled hydrodynamic and structural responses analyses on particular compliant moorings for floating WEC systems.
13,24	Three-float multi-mode line absorber	A light cord with a small buoy, while no further detail was provided	Pneumatic actuator	1. Wave basin 2. Regular and irregular waves 3. Operational condition 4. The design principle of the experiment is that the mooring is negligible and can be excluded from the numerical simulation.	Enabled structural analysis and power capture evaluation of WEC buoys.
17,25,26	Combined semisubmersible wind energy and flap-type WEC	Catenary	Linear mechanical rotary damper	1. Ocean basin 2. A constant wind together with regular or irregular waves 3. Operational and survival conditions	Enabled fully coupled hydrodynamic and structural response analyses on a combined wind and wave concept.
17,27	Combined spar torus wind and wave energy concept	Catenary	Hydraulic quadratic damper	1. Towing tank 2. Constant winds together with regular and irregular waves 3. Operational conditions	Enabled fully coupled hydrodynamic and structural response analyses on a combined wind and wave concept.
28	Array of point absorbers	- ^a	Electric motor with a servo controller	1. Deep water wave basin 2. Irregular waves	Enabled the simulation of WEC array hydrodynamic and power capture.

Current study	Point absorber	Taut mooring	Heave plate	1. Deep water wave basin 2. A constant current together with regular and irregular waves 3. Operational and survival condition	Provided a fully coupled simulation methodology to study hydrodynamic and structural response analyses of a floating WEC system and its power performance.
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^a Not considered in the experimental model, or, the information is not available.

The development of a WEC system relies heavily on the numerical modelling and simulation because they provide the flexibility to assess a large number of design choices for a WEC at a relatively low cost. Various numerical modelling methods have been reviewed in^{35,36} for WEC systems and in^{37,38} specifically for mooring systems. Because of the coupling effect between the WEC and the mooring system,^{20,39,40} the time-domain coupling method has been suggested best suited to study hydrodynamic and structural responses of mooring lines used in WEC systems. Within the category of time-domain coupled analysis, numerical models using highly non-linear computational fluid dynamics has found to be advantageous to capture the non-linear dynamic response of the mooring lines particularly in the resonance region,⁴¹ yet its high computational expense gives way to linear potential flow theory and finite element method as dominant methods for simulating mooring dynamics, such as examples shown in^{17,19,25-27,42} and in the methodology proposed in the present study. The proposed methodology shows strength in detailed fatigue assessment of mooring lines and power cables used in WEC systems and gives a possibility for a designer to conduct parametric analyses with regard to different design parameters and environmental factors.^{20,43,44}

An important purpose of this study is to contribute to structural reliability analysis of ocean wave energy devices, with emphasis on the structural integrity of the mooring system which is important for its long-term use and survivability. It requires validated numerical models which, for example, can be used to provide confidence in cost-efficient development of WEC systems which is needed to reach the phase of commercialization in large scale. Thus, the current study presents the validation of a numerical methodology together with an extensive test program of a point-absorber WEC system. In comparison to the existing experiments and validation work in the literature, the novelty of this study includes: (1) a taut mooring system and the effect of wave-current interaction were tested together with a complete WEC system in an ocean wave basin (see Sections 2.2 and 2.3), (2) the mooring responses were investigated in detail by measuring the axial force at two places in each leg, and six degree-of-freedom (DOF) motions for one submerged floater (see Section 2.4), (3) the entire WEC system was experimentally modelled and numerically simulated in order to investigate the important coupling interaction between the mooring lines and other components in the WEC system (see Sections 2 and 3).

Section 2 presents the experimental set-up and the design of the ocean basin test program. Section 3 describes the numerical models of the WEC system in the experiments. The results from the validation of the numerical models are presented in Section 4 using results from among others static tests, decay tests, as well as regular and irregular sea state condition tests. Section 5 presents a discussion of model calibration with regard to damping, mooring stiffness, design of experiments and other observations made in the physical experiment which cannot be captured by the current simulation models. The conclusions of the study are presented in Section 6 which are expected to contribute to the methodologies for WEC design and the quantification of the uncertainty associated with numerical predictions of the dynamic behaviour of WEC systems.

2 Experimental set-up

The experiments were performed in the Deepwater Offshore Basin at Shanghai Jiao Tong University in Shanghai, China.⁴⁵ The basin is 50 m long and 40 m wide and the water depth can be adjusted between 0 and 10 m by a movable floor. It is equipped with a segmented wave generator system and ocean current can be applied using an external re-circulated current generating system.

Figure 1 shows the set-up of the WEC system and its orientation in relation to the incoming waves and ocean current. In the following subsections, the detailed experimental testing program and the different parts in the WEC system are presented and motivated. The experimental WEC system is based on a prototype device designed by the Swedish company Waves4Power and it is currently installed in full scale in Runde (Norway).⁴⁶ This study strived for a large-scale test in order to capture hydrodynamic phenomena and force responses which would otherwise be difficult to observe due to the scaling effect. Therefore, a test model of scale 1:20 following the Froude law was chosen for the experiment. However, due to facility limitations of the ocean basin, survival tests were performed using a model of scale 1:36. In the current study, only the results for operational conditions (with the test model of scale 1:20) are reported and discussed, while the survival conditions are a matter of future work.

2.1 WEC buoy and PTO system

Figure 1 consists of two diagrams, (a) and (b), showing the dimensions of a mooring system.

(a) Side view of the mooring system. The diagram shows a vertical structure with a horizontal section at the top. The dimensions are as follows:

- Overall height: 38.162
- Height of the upper section: 34.900
- Height of the lower section: 30.356
- Width of the upper section: 8.000
- Width of the lower section: 3.500
- Horizontal distance from the centerline to the edge of the upper section: 2.250
- Horizontal distance from the centerline to the edge of the lower section: 3.614
- Horizontal distance from the centerline to the edge of the lower section: 1.942
- Horizontal distance from the centerline to the edge of the lower section: 0.200
- The Still Water Level (S.W.L.) is indicated by a dashed line.

(b) Top view of the mooring system. The diagram shows a hexagonal structure with dimensions:

- Horizontal distance from the centerline to the edge: 0.937
- Horizontal distance from the centerline to the edge: 0.226
- Horizontal distance from the centerline to the edge: 1.726
- Horizontal distance from the centerline to the edge: 2.100
- Connection point to the spring Sp1 in mooring segment S1 is indicated by an arrow.
- Connection point to the spring Sp2 in mooring segment S2 is indicated by an arrow.

Figure 2. Profile view and geometrical dimensions of (a) the WEC buoy, and (b) the floater in full scale (unit: metres).

Table 2. Basic properties of the WEC buoy.

	Full-scale prototype	1:20 model
Mass, M_w [kg]	451751	55.092
Draft, D_w [m]	34.9	1.745
Centre of gravity, COG_w [m] ^a	(0, 0, -14.1)	(0, 0, -0.705)
Roll inertia relative to COG_w , I_{xx} [kgm ²]	5.160×10^7	15.732
Pitch inertia relative to COG_w , I_{yy} [kgm ²]	5.160×10^7	15.732
Yaw inertia relative to COG_w , I_{zz} [kgm ²]	2.154×10^5	6.566×10^{-2}

^a The origin of the reference Cartesian coordinate is placed in the plane of water surface at the geometric centre of the WEC buoy, when it is in its unloaded neutral position.

The heave plate acts as a damper in the entire WEC system. Its target damping effect in full scale is chosen to be equal to the radiation damping of the WEC buoy in heave direction under resonance frequency,⁵⁰ based on an assumption that an ideal linear PTO system is to be used. The frequency dependency of an optimised PTO was therefore disregarded in this approach. The damping effect of the modelled heave plate was estimated by a decay test through a 1 DOF spring-mass-damping system. The dimension of the heave plate was determined when the equivalent linear damping estimated from the 1 DOF decay system satisfied the target linear damping as shown in Table 3.

Table 3. The properties of the PTO system. Values are presented in model scale 1:20 except as otherwise noted.

Dimension of the heave plate, length $\times$ width $\times$ height [m $\times$ m $\times$ m]	$0.21 \times 0.21 \times 0.005$
Dimension of the connection bar, length $\times$ diameter [m $\times$ m]	0.658×0.19
Unloaded length of the spring [m]	0.617
Total length of the spring after pretension [m]	1.168
Mass of the spring [kg/m]	0.811
End stiffness of the spring [N/m]	5.88

Nominal (outermost) diameter of the spring [m]	0.015
Thread diameter of the spring [mm]	1.5
Axial stiffness of the spring [N]	3.626
Pretension force of the spring [N]	3.242
Target linear damping of the PTO system in prototype scale [Ns/m]	3.315×10^4
Target linear damping of the PTO system in model scale [Ns/m]	18.081
Equivalent linear damping (estimated by 1 DOF decay test) of the heave plate [Ns/m]	18.079

The heave plate was rigidly connected to the WEC buoy by a steel rod (indicated as the connection bar in Figure 1(a)), and to a fix point at 30.9 cm above the movable floor of the ocean basin by a spring. This ensures that the movement of the heave plate always follows the motion of the WEC buoy. The heave plate is placed at as large a depth as practically achievable to make the influence of the water motion due to the surface waves small. In order to cover all wave conditions in the experiment, a depth of 50 m (corresponding to full scale) for the heave plate was found to be suitable.⁵¹ No optimisation was made with regard to PTO tuning under various frequencies nor to the position of the heave plate. Whilst the choice of heave plate adopts a linear assumption, its inherent quadratic nature due to viscous force effects was numerically modelled through the Morison equation to ensure a correct validation process, see further discussion in Section 3. Table 3 presents the properties of the PTO system used in the experiments. The damping effect induced by the PTO system is therefore truthfully simulated, hence the dynamic response of the WEC buoy is captured with minimum complexity.

2.2 Mooring system

A taut mooring system was used which consists of three mooring legs (hereafter referred to as L1, L2, and L3), evenly distributed and attached around the outer rim of the WEC buoy; see Figure 1. Each mooring leg consists of two segments (referred to as S1 and S2), with a submerged floater positioned in the connection between S1 and S2. The points at which a mooring line connects to the WEC buoy, and to the floor of the basin, are referred to as the fairlead and the anchor, respectively. The full-scale prototype installed in Runde (Norway) has elastic mooring lines made of polyester. Because a proper scaled material was not available, thin steel wires (referred to as Wi) were used instead in the experiments with additional springs (referred to as Sp) in the connection points that

mimicked the correct mooring segment stiffness in model scale compared to full scale; see Figure 1(a). Table 4 presents the properties of the mooring system used in the experiments, and Figure 2(b) presents the geometrical dimensions of the floater. The top and bottom tips of each floater were the connection points to the springs at each side of the mooring segment; see the illustration in Figure 2(b).

Table 4. Basic properties of the mooring system.

	Full-scale prototype	1:20 model
Depth of the anchor [m]	80	4.0
Radius of the anchor [m]	125.109	6.255
Height of the fairlead [m] ^a	1.125	0.056
Pretension force at the fairlead [N]	2.200×10^4	2.683
Dry mass of each segment [kg/m]	4.900	- ^b
Submerged weight of each segments [N/m]	35.868	0.087
Nominal diameter of each segment [m]	0.08	- ^b
Axial stiffness of each segments [N]	5.754×10^6	701.671 ^c
Length of the segment S1 [m]	100	5.000
Length of the segment S2 [m]	69.745	3.487
End stiffness of the segment S1 [N/m]	5.754×10^4	140.3 ^c
End stiffness of the segment S2 [N/m]	8.250×10^4	201.2 ^c
Mass of the spring [kg/m]	- ^d	0.211
Nominal (outermost) diameter of the spring [m]	- ^d	0.011
Thread diameter of the spring [mm]	- ^d	1.5
Axial stiffness of the spring [N]	- ^d	26.5
Length of the spring Sp1 [m]	- ^d	0.210

Length of the spring Sp2 [m]	^d	0.143
End stiffness of the spring Sp1 [N/m]	^d	126
End stiffness of the spring Sp2 [N/m]	^d	185
Mass of the steel wire [g/m]	^d	13.8
Diameter of the steel wire [mm]	^d	0.83
Axial stiffness of the steel wire [kN]	^d	50.0
Length of the steel wire Wi1 [m]	^d	4.790
Length of the steel wire Wi2 [m]	^d	3.344
Mass of the floater [kg] ^e	2900	0.354
Height of the floater [m]	3.6	0.180

^a Measured from the upper deck.

^b Defined for the spring and steel wire, respectively.

^c The target design value for the combined equivalent stiffness when one spring and one steel wire are connected.

^d Not used in the full-scale prototype.

^e The COG of the floater is located at the geometrical centre of the floater and its moment of inertia was not modelled in the experiment.

2.3 Environmental conditions – test program

Table 5 presents the tested wave and current scenarios and the corresponding values in full scale. The regular waves were chosen so that they cover the ranges between linear (Re1-Re3, and Re6-Re7), intermediate (Re4), and non-linear (Re5) waves. The distinction between different waves is based on the definition in⁵². The irregular waves were chosen from the wave scatter diagram near Runde (Norway) where the full-scale prototype WEC system was installed. The case OP1 was defined as the optimum operation condition for the installed WEC system.⁴⁶ In contrast to OP1, the case OP2 was chosen for being away from the resonance frequency of the WEC buoy. Some cases with ocean current are also included in the test program. Several investigations in the literature have shown that the combination of wave-current environmental loads can have a significant influence on a WEC system's characteristics and how it should be designed with regard to e.g. fatigue of the mooring lines.^{32-34,44} Thus, a current with constant velocity and slab profile over the water depth was tested, where the choice was motivated

for being the design value for the installed full-scale prototype WEC system. In the current investigation, only the results for the operational conditions are reported in the model validation.

Table 5. Summary of tested wave and current scenarios, presented in full-scale and model-scale values. All cases are operational conditions tested at model scale 1:20 except for the cases Curr₃₆, SURV, and SURVc.

Case name	Regular or irregular waves ^a	Regular wave period (T) or peak wave period (T_p)		Regular wave height (H) or significant wave height (H_s)		Current velocity		Wave and current direction ^b
		Full-scale	Model-scale	Full-scale	Model-scale	Full-scale	Model-scale	Full- and model-scale
		[s]	[s]	[m]	[m]	[m/s]	[m/s]	[deg]
Re1	Regular	3.185	0.712	0.238	0.012	-	-	0
Re2	Regular	5.370	1.201	0.675	0.034	-	-	0
Re3	Regular	6.370	1.424	0.950	0.048	-	-	0
Re4	Regular	6.370	1.424	1.900	0.095	-	-	0
Re5	Regular	6.370	1.424	3.801	0.190	-	-	0
Re6	Regular	7.370	1.648	1.272	0.064	-	-	0
Re7	Regular	12.740	2.849	3.679	0.184	-	-	0
OP1	Irregular	6.5	1.453	2.5	0.125	-	-	0
OP1d	Irregular	6.5	1.453	2.5	0.125	-	-	180
OP1c	Irregular	6.5	1.453	2.5	0.125	0.514	0.115	0
OP1n ^c	Irregular	6.5	1.453	2.5	0.125	-	-	0

OP2	Irregular	9.6	2.147	4.5	0.225	-	-	0
Curr	-	-	-	-	-	0.514	0.115	0
Curr ₃₆	-	-	-	-	-	0.514	0.086	0
SURV	Irregular	13.8	2.3	9	0.25	-	-	0
SURVc	Irregular	13.8	2.3	9	0.25	0.514	0.086	0

^a A regular wave is defined by the wave period (T) and the wave height (H), while an irregular wave is defined by the peak wave period (T_p) and the significant wave height (H_s). All irregular waves follow the JONSWAP spectrum with the spectral peak parameter (γ) is set as 2.4.

^b The definition of loading direction in relation to the orientation of the WEC system is shown in Figure 1(b).

^c The test condition of OP1n is the same as for OP1 except that the PTO system is removed in OP1n.

2.4 Measured quantities and instruments

Six quantities of the system were measured during the experiments for the operational conditions:

1. Motion of the WEC buoy in 6 DOFs.
2. 6 DOFs motion of the floater on mooring leg L2.
3. Accelerations at the three fairleads in 3 DOFs.
4. Forces on the upper and lower sides of the heave plate.
5. Forces at the upper ends of both segments for all three mooring legs.
6. Water surface elevations.

All the measurements were sampled at a frequency of 25 Hz which ensured reliable and sufficient data collection from the entire measuring system used in the experiments. To ensure that the installation of the instruments does not affect the motion responses of the WEC system, the contribution of the weight from all instruments were considered during the mass calibration of the WEC buoy.

The motions were measured by the Qualysis Oqus non-contact optical motion tracking system. For the WEC buoy, the motion optical markers were attached at its upper deck; see Figure 3(a). Because the floaters were submerged 0.6 m (in model scale 1:20) below the water surface and the motion measurement can only be taken above the water surface, a long, light rod was attached to the floater on mooring leg L2 and its measurements were taken from the upper end of the rod; see Figure 3(b). A sensitivity study was performed prior to the main test scheme and it showed that the WEC system's dynamics was not affected by the instrumentation (i.e. the floater with the long rod); see the discussion in Section 5.4. Three KYOWA AS-10TG accelerometers (Figure 3(a)) were also installed in each fairlead point and measured the accelerations in the three translational DOFs. They served as redundant measurement sources and were also used to derive the rotational acceleration of the WEC buoy and motion at the three fairleads.

Figure 3. Instrument installation in the WEC system: (a) optical motion makers at the upper deck of the WEC buoy, and accelerometers at the fairlead of the three moorings; (b) optical motion markers at the upper end of the rod which attaches to the floater on mooring leg L2; (c) force transducers at the upper and lower side of the heave plate used in the PTO system; (d) force transducers at the fairlead of each mooring leg (i.e. upper end of mooring segment S1); and (e) force transducers at the upper end of the mooring segment S2.

All forces were measured by KYOWA LUX-B-100N-ID force transducers. The total effect of the PTO system was quantified by measuring the force at the upper and lower sides of the heave plate; see Figure 3(c). Moreover, two force transducers were used for each mooring leg, one at the fairlead (Figure 3(d)) and the one at the upper end of the mooring segment S2 (Figure 3(e)). Three RBR WG-50 wave gauges were used

during the wave calibration test for checking the water surface elevation in the basin before immersing the tested WEC and mooring system. The first wave gauge (WG(C)) was in the target position of the WEC buoy and the other two were installed 3 m away at the north (WG(N)) and west (WG(W)) sides of the central one. During the main test program, WG(C) was replaced by the WEC buoy and it was tested together with wave gauges WG(N) and WG(W) retained in the basin; see Figure 1(b).

3 Numerical models and simulations of the experiments

The simulation model to be validated is shown in Figure 4. The entire simulation model consists of six sub-models, each one with a specific purpose. These are a panel model of the buoy, a Morison model of the buoy, a Morison model of the PTO system, a point model of the buoy, a finite element (FE) model of the mooring system (including floaters) and of the helical spring used in the PTO system, and an environment load model. The details about each model are summarized in Table 6. The theoretical background and the numerical simulation procedure that connects the different sub-models are explained in detail in^{20,43}.

Figure 4. (a) Initial configuration of the complete numerical model, and (b) zoom-in view of the geometry of the Morison model for the buoy and the PTO system.

Table 6. Presentation of sub-models that form the full simulation model of the experimental set-up.

Sub-model	Simulation purpose	Software (core solver)	Applied theory
Panel model of WEC buoy	<ol style="list-style-type: none"> 1. Hydrostatic data and inertia properties of WEC buoy 2. Global responses of WEC buoy, including first and second order wave exciting forces and moments, hydrodynamic added mass and damping, and steady drift forces and moments 	HydroD ⁵³ (WADAM)	<ol style="list-style-type: none"> 1. Boundary element method 2. First and second order three-dimensional potential theory
Morison model of WEC buoy	<ol style="list-style-type: none"> 1. Linearized viscous drag of the WEC buoy 	HydroD (WADAM); DeepC ⁵⁴ (SIMO)	<ol style="list-style-type: none"> 1. Morison equation 2. Two-node beam element model
Morison model of PTO system	<ol style="list-style-type: none"> 1. Linearized viscous drag effect of the PTO system (i.e. heave plate) 2. Restoring contribution of the PTO system (i.e. helical spring) 	HydroD (WADAM); DeepC (SIMO)	<ol style="list-style-type: none"> 1. Morison equation 2. Two-node beam element model
Point model of WEC buoy	<ol style="list-style-type: none"> 1. Motion response of WEC buoy in time domain 	DeepC (SIMO)	<ol style="list-style-type: none"> 1. Rigid body motion 2. Retardation function 3. Newmark β time stepping scheme
FE model of mooring system and helical spring used in PTO system	<ol style="list-style-type: none"> 1. Time-domain motion and force response of the mooring system (including floaters, wires, and springs), and helical spring used in PTO system 	DeepC (RIFLEX)	<ol style="list-style-type: none"> 1. Morison equation for hydrodynamic loads 2. Continuum mechanics theory for structural response 3. Newmark β time stepping scheme

Environment load model	1. External environmental loads acting on the WEC system	HydroD (WADAM); DeepC (SIMO and RIFLEX)	1. Airy wave theory 2. Unidirectional current (time- and depth-invariant)
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The commercial software package DNV GL SESAM^{53,54} was used to perform the numerical simulations of the physical experiments presented in Section 2. Given the intention of validation in the current study, the simulation procedure and the modelling principle are fully adopted from^{20,43}. However, the detailed set-up is redefined so it is consistent with the present experimental set-up. This includes intrinsic differences in the WEC system between the cited references and the current study, namely, dimensions of the system, configuration and constitution of the mooring system, and realisation of the PTO system. The main characterises of the simulation model in this study can be summarized as follows:

1. The dimension of the numerical model is defined directly according to the scaled-down experiment model. Moreover, the comparison of the validation result is performed exclusively in model scale, i.e. there is no transformation of the values to full-scale values in the comparison between the simulated and experimental results.
2. The main global response of the WEC buoy is evaluated numerically by the hydrodynamic diffraction/radiation software. However, a Morison model is included additionally to account for the viscous drag damping effect due to its long cylindrical and hemispheric geometry of the WEC buoy. Figure 4(b) presents the geometry of the Morison model. The effects from the long cylinder and the hemisphere are simulated by Morison models with a geometry as straight line and circle, respectively, indicated as a cylinder and a hemisphere in Figure 4(b).
3. The spring and the wire used in each segment of a mooring leg (see Section 2.2) are defined separately. They are both represented by first-order bar elements with homogeneous cross-sectional properties, and their motions as well as force responses are solved through element discretisation with the FE method.
4. The floaters are modelled as point objects, which are defined exclusively by the volume, mass, and hydrodynamic loads (namely added mass and drag coefficient). It is assumed that each of the two mooring segments in one mooring leg is attached in the COG of the floaters (see Table 4). To compensate the length discrepancy of the mooring system due to the point-modelling of floaters, the height of the floater (Table 4) is equally divided and added to the length of the wires at the upper and lower segments of each mooring leg. The model of the floater is directly attached to a nodal point of the FE model (described in the previous point) and has no motion DOFs by itself.
5. According to Section 2.1, the PTO system consists of a connection bar, a heave plate, and a spring that is connected to the movable floor. The spring of the PTO system is modelled in the same way as the mooring system. The connection bar and the heave plate are modelled as rigid bodies and they are considered as small

- structures according to the definition in⁵². Therefore, they are represented by the Morison model, as indicated by “connection bar” and “heave plate” respectively in Figure 4(b).
6. The wave zone kinematic, and hence, the wave force is computed to the instantaneous water level according to the formulation in⁵⁵. For the FE and Morison models listed in Table 6, the wave-induced excitation forces are calculated along the whole instantaneously submerged length together with the instantaneous wave kinematics, whereas the viscous loads are computed using the drag term in the Morison equation.
 7. The numerical simulation of the dynamic behaviour of the WEC system is performed by a coupled response analysis as recommended in^{20,43-44}.

4 Results

The main purpose of the validation is to assess to which degree the computational and simulation model is an accurate representation of the real world from the perspectives of the intended use of the model. Validation should always be preceded by a verification of the numerical model where it is assured that the model accurately represents the underlying mathematical model and its solution. This is important so that the errors discovered through validation can be isolated from those discovered through verification. Another critical issue is that validation should be performed at different levels in the system hierarchy of the model to assure that the model both describes the detailed physics in components and the non-linear interactions between components. In the present study, this is accomplished by both comparing motions on the component level and stiffness and decay behaviour of the whole WEC system. The following steps have been used during the validation.

An abstraction was made to a conceptual model where the domains of interest of the model are identified. This includes important physical processes and assumptions, and system-response quantities. The conceptual model then forms the basis for two parallel tracks, the computational model and the validation experiment. The computational track starts with a verification process where first algorithmic and programming errors are identified. In the present case, this verification of code is made based on existing certificated commercial software produced by DNV GL.⁵⁶ This also forms the fundamental verification of numerical accuracy during computation. Next, the numerical modelling accuracy is verified. For the current model this has been performed in a previous study, see²⁰.

The validation experiments have been performed with the purpose to provide the information needed to assess the validity of the simulation model. This experimental part also includes quantification of various sources of uncertainty of the experimental data, for instance measurement errors, evaluation errors, or uncertainties in the applied loading. The experimental results and the simulation results, including uncertainties, were finally compared. It is important to note that the experimental data could be used for either the validation purpose to assess the predictive capability of the model, or, for calibration,

which only determines the model's fitting ability, not its predictive capability. A model calibrated to experimental data may not yield accurate predictions over the range of its intended use and data used for model calibration must therefore remain independent of data used to assess the predictive capability of the model. The main focus of the current study is the model validation while an example of possible calibration is elaborated in Section 5.1.

This section presents only a selection of results from the validation study which are representative for all studied cases. It begins with a presentation of results from an analysis of the horizontal stiffness of the mooring system in Section 4.1 followed by a comparison of results from decay tests (heave and surge) in Section 4.2. Results from regular wave cases, and from operation conditions with irregular wave cases are presented in the Sections 4.3, and 4.4, respectively. All results presented in Sections 4 and 5 are for the test model of scale 1:20.

4.1 Horizontal stiffness of the mooring system

Horizontal stiffness tests of the mooring system were performed in calm water. The PTO system was removed from the WEC, and tests were made in surge and sway directions, to obtain the horizontal stiffness curves of the whole mooring system. With the horizontal stiffness curves, the restoring capacity of the mooring system can be compared between the simulation model and the experiment. Figure 5 presents results from the simulation model and the experiments in surge and sway directions. For the surge direction, the simulated mooring stiffness is smaller in the high load region (i.e. forces above 5 N) but larger in the low load region, while the stiffness of the mooring in sway direction is always estimated to be larger from the simulation. Regardless of the direction, a larger discrepancy between the experiment and simulation is always observed in the high force region. The relative error in the horizontal offset of the WEC buoy does not exceed 15% in either direction.

Figure 5. Horizontal stiffness (surge and sway directions) of the mooring system.

4.2 Decay tests

The purpose of the decay tests was to investigate the resonant period and damping coefficient of the WEC system in its horizontal (surge) and vertical (heave) directions of motion. A series of tests were carried out in calm water: with/without the presence of mooring lines, with PTO (i.e. with/without heave plate and/or spring) and without PTO. The purpose of conducting multiple decay tests was to compare the damping effects for different system configurations, with emphasis on what the simulation model can predict and what was measured.

The length of the testing time and assessment was according to common practice: at least two periods must be included, and that a decay tendency must be clearly indicated. Figure 6(a) shows examples of the heave decay tests from the experiment and the numerical simulation, respectively. Time zero (0 s) is defined as the moment when the WEC system is released from the forced displacement. No value was shown at the time before zero and after ten seconds for the experimental result. The lack of data was due to an enforced cut-off in the time signals from the laboratory signal analysis instrument.

Figure 6. (a) Heave decay test results for the complete WEC system (buoy, mooring, PTO-1, and PTO-2 were all installed) from an experiment and its corresponding numerical simulation. (b) Illustration of data evaluation from a decay test result. The peak and trough points are marked and they are used in the calculation of the resonant period and the damping coefficient.

To calculate the resonant period and the damping coefficient from the decay test, peaks and trough points need to be identified; see an illustration in Figure 6(b). Note that the first decay oscillation was ignored in the calculation, because the WEC buoy may experience additional damping due to the removal of the experimental instruments which were used to enforce an offset of the WEC system. With these local maxima and minima, the resonant period (T_r) and the damping coefficient (ζ) were calculated as follows, for each decay test case using the results from the simulation model and the experiments:⁵⁷

$$T_r = (t_5 - t_1)/2 \quad (1)$$

$$\zeta = \frac{\ln(u_1/u_5)/2}{\sqrt{4\pi^2 + [\ln(u_1/u_5)/2]^2}} \quad (2)$$

where t_i is the time at which the i th peak or trough occurs, and u_i is the amplitude of the i th peak or trough in the free vibration, as shown in Figure 6(b).

It was assumed that the decay response follows the logarithmic decrement, and only 1 DOF motion is present in each decay test. Table 7 presents the summary of results from all the conducted decay tests. The results from the most complete set-up (buoy, mooring lines, PTO-1, and PTO-2) show very good agreement for the resonant period, with a relative error between simulation and experiment no larger than 5%. The agreement for the damping coefficient is acceptable when the PTO system is absent from the WEC system (maximum relative error of 18%) but less satisfactory when any of the PTO systems (PTO-1 or PTO2) were included (maximum relative error of 120%). This result suggests that the use of the Morison model may not be able to capture the total damping effect from the PTO system.

Table 7. Resonant period (T_r) and damping coefficient (ζ) from decay tests (PTO-1: heave plate (no spring); PTO-2: heave plate and spring).

System components in decay test				Motion direction	Experiment		Simulation	
Buoy	Mooring	PTO-1	PTO-2		T_r [s]	ζ [-]	T_r [s]	ζ [-]
x	x			Surge	32.080	0.055	29.980	0.057
x	x	x			30.000	0.045	31.480	0.063
x	x	x	x		23.760	0.057	23.180	0.053
x				Heave	1.420	0.045	1.430	0.046
x	x				1.440	0.039	1.440	0.046
x	x	x			1.520	0.048	1.440	0.046
x	x	x	x		1.480	0.028	1.440	0.062

4.3 Regular wave cases

The design of the current WEC system enables large buoy motions in heave motion, but the mooring system configuration also allows for large motions in surge. Moreover, point absorbers are generally found to exhibit large heave and pitch motions.^{58,59} Hence, these motion directions were compared between the simulation model and the experiments.

The results under the regular wave cases are presented in the form of response amplitude operators (RAOs); see Figure 7. There is good agreement between the simulations and the experiments in both heave and surge motions when the incoming wave frequency is far off from the resonance frequency of the system (see the cases Re1, Re2, Re6, and Re7), with only one exception found in surge for the case Re2. For the cases where good agreements were obtained, the ratios of the estimated response amplitude between the simulations and experiments are in average of 0.97 and 1.05 in surge and heave, respectively.

Figure 7. Results from experiments and numerical simulations of (a) surge, (b) heave, and (c) pitch RAOs of the WEC buoy. For each set of the results, the data points from short to long wave periods, and from low to high RAOs are corresponding to the regular wave cases Re1 to Re7.

The results from the simulations and experiments deviate both in surge and heave for the cases with a wave period close to the resonance period (namely cases Re3, Re4, and Re5). The simulation model slightly underestimates (by a factor of 0.76) the surge motion in low wave height condition (Re3) while it overestimates the surge motion (by a factor of 2.11) in the high wave height condition (Re5). This result aligns with the trend observed from the horizontal stiffness test presented in Section 4.1. For the heave

direction, larger responses were always predicted by the simulations, where the ratio of the estimated response amplitude between the simulations and experiments is in average of 2.7 (Re3, Re4, and Re5).

The comparison between the experiments and simulations in the pitch direction is less satisfactory, where the ratio between the simulations and experiments ranges from 0.35 to 1.75. The cause of this discrepancy is probably attributed to the uncertainty from a complex PTO installation. This result also suggests that a more detailed modelling of the PTO system may be needed in future work. A general conclusion made from the regular wave test cases is that the simulation model can capture the characteristics at the resonance frequency, but the current modelling approach particularly for the damping is not sufficient to capture inherent non-linear nature of the damping presented in the WEC system and this insufficiency is particularly evident under the resonant wave conditions.

4.4 Irregular wave cases

Irregular waves were included in the test program, see Table 5, to investigate the integrated hydrodynamic characteristics of the full WEC system. Comparison between the experimental and simulation results for significant motions (namely, highest third of the motion response) and standard deviation in the surge, heave, and pitch directions of the WEC's motions from all the irregular waves cases are presented in Figure 8. Additionally, Figure 9 presents the time and frequency domain results of heave and surge motion responses for OP1 and OP1c, i.e. without and with ocean current, from simulations and experiments.

Figure 8. Significant motion and standard deviation (std) of the surge, heave, and pitch motions of the WEC under irregular wave cases.

Figure 9. Experimental ((a) and (b)) and numerically simulated ((c) and (d)) time domain results for surge and heave motion responses of the WEC buoy for the irregular sea state conditions OP1 and OP1c, where the response spectra for the case OP1 is further presented in (e) and (f).

A comparison in the surge motion under the case Curr confirmed that the use of current load coefficient together with the mean drift approximation (see Table 6) is capable of modelling the current load effect on the WEC buoy. The ratio of the estimated significant motion in surge between the simulation and experiment is 0.7; see Figure 8(a). Ideally, the current load coefficient should be determined by a dedicated tank test with the WEC buoy, while this study uses an empirical formula as suggested in⁶⁰. Hence, a discrepancy observed in the estimated surge motion due to the current load between the experiment and simulation results was judged to be acceptable. The comparison between the cases OP1 and OP1c shows that the influence of the current load on the buoy's motions is correctly modelled by the simulation model, where the presence of the current

load will lead to an increased mean drift of the WEC buoy in surge direction, decreased pitch motion, while the effect on the heave motion is insignificant.

In the previous Sections 4.2 and 4.3, results have shown that the damping in the simulation model is not fully satisfactory because of the use of radiation damping together viscous drag damping may not be sufficient to capture the non-linear nature of the damping particularly under the resonant condition. This further affects the results from the irregular wave case simulations. For all the test cases with the incoming wave as OP1, the results again show more damping (smaller motion responses) in the experiments compared to the simulation model results (Figure 8), especially for the larger motions where non-linearity becomes more pronounced. Such phenomena can be further elaborated by Figure 9. From the time domain result, one can clearly see that the motions of the WEC buoy estimated from the simulation model have higher probability to experience large peak motions. In addition, by examining the frequency domain results, the difference between the results from the experiments and simulations are increased when closer to the resonance frequency. For all the OP1 related cases, the average ratios of the estimated significant motion between the simulation and experiment are 1.64, 1.85, 1.38 in surge, heave, and pitch direction, suggesting that an additional damping needs to be implemented to fully capture all the damping effect present in the WEC system.

A comparison between the experiment and numerical simulation shows reasonable agreement for case OP2, with a relative difference in the observed motion response of 8%, 35%, and 5% for surge, heave, and pitch directions, respectively. Together with the results presented in Section 4.3, this study demonstrates that our numerical model approach is useful to predict the dynamic motion responses of the WEC buoy under off-resonant wave conditions.

5 Discussion

The objective and goals with this validation study of the simulation model are considered fulfilled even though there have been some discrepancies compared with the experimental results. Sources of uncertainties and deficiencies have been identified that show e.g. how the damping model in the software can be employed on the current WEC system. The following subsections present additional and valuable findings from the validation study.

5.1 Damping model and model calibration

In the current study, the damping of the WEC system is considered in two forms, namely radiation damping and linearized constant viscous drag damping. Although the use of viscous drag damping considers the non-linear nature of the damping, the linearization is nonetheless unable to account for damping characteristics in terms of its frequency dependency. The results in Sections 4.3 and 4.4 showed good agreement for the wave

cases that are outside the resonance frequency of the WEC system. However, for all cases which are on or close to the resonance frequency of the system, such as the cases Re3-Re5 and OP1, larger discrepancy was observed in the estimated buoy motions between the numerical simulations and experiments. This result suggests that the use of one single value of viscous drag damping is still insufficient to capture the WEC's dynamics under all wave conditions. The situation can in a specific case be handled if a model calibration and adjustment of the damping model are carried out, for example, by introducing additional damping. Figure 10 shows an example after such corrections of the damping. For this specific calibration example, the damping of the system is calibrated by additionally introducing a linear damping of 50 Ns/m and a critical damping coefficient (defined as the fraction of critical damping) of 0.15 both in heave direction, which resulted in good agreement in results between the simulation model and the experiments.

Figure 10. Surge, heave, and pitch motion responses of the WEC buoy under the regular wave case Re4: experiment, simulations before and after corrections including calibration.

5.2 Mooring stiffness

The taut-moored system used in the study is very complex and included many parts and connection points in the physical model. It was a challenge to represent it in the simulation model as it introduced large model uncertainty. For instance, different small steel rings were used to link the floaters, wires, and springs in the experimental model (see a photo of the steel rings in Figure 11) but they were represented as nodes in the numerical model.

The results in Section 4.2 showed reasonable agreement for the mooring stiffness in both surge and sway directions (with a measure of maximum 15% difference in the horizontal offset of the WEC buoy in either direction), while the agreement can be considered less satisfactory in the high force regions. When performing the stiffness test by numerical simulation, the applied force strictly applies in the x - or y -direction and at the geometric centre of the WEC buoy in the plane of the water surface. In the experiment, however, uncertainty can be expected due to the fact that the WEC buoy was pushed manually in the x - and y -directions, respectively with some possible angular deviations from these directions. Thus, one may anticipate a larger discrepancy in the estimated mooring stiffness between the experiments and simulations in the high force regions.

Figure 11. Different steel rings which were used to connect different components in a mooring leg. Similar steel rings were also used in the PTO system, see Figure 3(c).

5.3 Design of the PTO in the experiment

The PTO system in the physical test model used in the experiments was not designed to mimic the PTO in the full-scale prototype WEC installed in Runde (Norway). It is a PTO which was designed and fit for purpose for validation of a simulation model against a physical model used in ocean basin experiments. The test program was designed to carry out experiments with and without the PTO (namely OP1 and OP1n) in order to be able to validate the simulation model with or without it included in the model. The validation for the PTO analysis is a matter of future work. Nonetheless, by comparing the experimental results for the cases OP1 and OP1n as presented in Figure 8, the presence of the PTO system was indeed able to keep the buoy moving in the vertical direction (by reducing the pitch motion) and to produce a damping effect (as can be observed from a reduced heave motion of the WEC buoy). These results suggest that the designed experimental PTO system was suited for the purpose of the study.

5.4 Sensitivity of the installed rod for floater measurement

As presented in Section 2.4, a rod was attached to one floater in order to measure its motions. Prior to the experiments, a mass calibration was performed to ensure that the mass and COG of the rod-attached floater was the same as for the other two floaters. During the main test program, the cases Re7 and OP1 were repeated to investigate whether the presence of the rod influenced the dynamic responses of the system. Table 8 presents the statistical results of the motion of the WEC buoy from the two repeated test cases. The results confirm that the WEC's motions were insensitive to the presence of the rod on the floater.

Table 8. Comparison of the WEC's motions for a floater with and without a rod. The case OP1 is compared in terms of the significant motion and standard deviation while the case Re7 is compared in terms of mean response amplitude and standard deviation.

		OP1		Re7	
		No rod	With rod	No rod	With rod
Surge [m]	Significant motion / Mean response amplitude	0.372	0.369	0.080	0.075
	Standard deviation	0.093	0.092	0.052	0.051
Heave [m]	Significant motion / Mean response amplitude	0.162	0.165	0.121	0.123
	Standard deviation	0.040	0.041	0.073	0.073
Pitch [deg]	Significant motion / Mean response amplitude	9.510	9.031	3.444	3.365
	Standard deviation	2.378	2.258	2.302	2.299

5.5 Other observations in the experiments

During the experiments, overtopping of the WEC (i.e. green water) was visually observed in all the irregular wave cases and regular waves cases Re3 – Re5; see examples in Figure 12. It is known that overtopping can contribute to additional damping of WECs. The used simulation model and software can neither simulate the phenomenon, nor consider its influence on the WEC's responses. This can explain why the motion responses simulated by the model in the wave conditions where overtopping was observed were larger than what was measured.

Figure 12. Example of observed overtopping of the WEC buoy, where the photo was taken from the test case OP1: (left) the WEC buoy is completely submerged, and (right) the WEC buoy leaves the submerged condition.

Vortex-induced motions were also visually observed during some of the experiments but it is a motion phenomenon which cannot be simulated by the DeepC software. This motion was captured in the recorded response signals, and identified as small, superimposed fluctuations on the larger motion responses, see an example of such a response shown in Figure 13.

Figure 13. Example of the observed vortex induced motion of the WEC buoy under the case Curr.

6 Conclusions

The study presented a validation of a numerical simulation model against a physical test model used in experiments in an ocean basin. The physical model was made in scale 1:20 of a taut-moored, point absorber type of WEC. The WEC system consisted of a buoy, a unique three-leg two-segment mooring system with submerged floaters, and a PTO system designed for the current experiment as a heave plate. The validation was made primarily with regard to motion responses of the WEC buoy in surge, heave, and pitch, but also horizontal mooring stiffness and decay tests.

A simulation of the horizontal stiffness test in surge and sway directions showed satisfactory agreement with the experiments with a relative error not exceeding 15%. The decay tests were performed in surge and heave directions, and very good agreement was achieved to estimate the resonant period with an error percentage less than 5% for all tested system conditions. For all the tested regular and irregular wave cases, the simulation results showed good agreement with the experiments when the wave

frequency is far off the resonance frequency of the WEC system, namely cases OP2, Re1-Re2, and Re6-Re7. The ratios of the estimated motion responses between simulation and experiment were in average of 1.15 and 1.11 in surge and heave, respectively. The comparison in the pitch motion is less satisfactory (i.e. a ratio of 0.72). For resonant wave conditions (such as cases OP1, and Re3-Re5), larger discrepancy was observed between the experimental and simulation results. The average estimated surge, heave, and pitch motion responses from the simulations were 1.49, 1.22, and 1.21 times those of the experiments. These results suggest that the damping model used in the current study was not sufficient. The non-linear dynamic characteristics were particularly evident under resonance conditions.

The proposed numerical methodology demonstrates a good capability to simulate dynamic motions of the WEC system in most of the cases but also shows some limitations when the wave conditions are close to the WEC's resonance frequency. The major reason for this was identified as the lack of possibility to include damping correctly in the software without carrying out a calibration. An example was presented which showed that after a correction and calibration of the damping in the simulation model was made, much better resemblance in surge, heave, and pitch motion responses was achieved close to the resonance frequency of the WEC system. The differences between experimental and simulation results give useful information about the predictive capability of the model. In addition, it should be noted that for sensitivity investigations, which are of high interest during the early design phase and where the influence of parameter variations are studied, the prediction uncertainty in the corresponding output variations are probably significantly smaller.

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Declaration of Conflicting Interests

The authors declare that there is no conflict of interest.

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