Power-take-off Control in a Scaled Experiment of a Point Absorber Wave Energy Converter

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Abstract—This paper investigated the design of a power-take-off control system for the study of a model scaled point absorber wave energy converter in a wave flume experiment, where the scale of the experiment must be kept low to reduce the blockage effects of the buoy in the wave flume. In keeping the model small, the parasitic loss in the experimental set-up due to friction in the PTO was significant in comparison to the power absorption capability of the model scaled wave energy converter, and consequently, the power absorbed by the wave energy converter was significantly underestimated. To address this major limitation, a simple friction compensation algorithm was tested in the power-take-off control. Experimental results were benchmarked against those obtained from potential wave theory solvers for the understanding of the efficacy of the proposed method. It was evident that the proposed power-take-off control provided high fidelity results under both regular and irregular incident waves as long as the dynamic power-take-off control force was able to overcome the friction force in the experimental set-up.

Keywords—Point absorber, model-scale experiment, wave flume, power-take-off control, parasitic loss compensation

I. INTRODUCTION

A point absorber (PA) wave energy converter (WEC) is a typical type of WEC with the main advantage being its insensitivity to wave direction [1]. Examples of operational PA devices are the Carnegie CETO and Wavebob. Prior to the manufacturing and commissioning of full-scale PA WECs in the open sea, model scaled experiments in wave tanks are required to fully understand the efficiency of the design as well as its survivability [2]. There are some world-class wave tank testing facilities around the world such as Flowave at the University of Edinburgh, Model Test Basin at the Australian Maritime College, and the Ocean Engineering Water Tank at Shanghai Jiao Tong University. A general guide for conducting model scaled experiments on WECs is presented in [3], under the assumption that a dedicated wave tank testing facility is available, which is however not always the case.

This work aims to address the challenges of undertaking model scaled experiments on point absorber WECs [4, 5] in a standard wave flume that many universities have. The main challenge arises from the scale of the experimental set-up in the wave flume. In order to minimise the effects of side wall reflection on the PA, the scale of the buoy must be sufficiently small so that the blockage ratio of the buoy in the flume is lower than 15% [6]. As the scale of the experiment is reduced, the parasitic losses in the experimental set-up (e.g. motor friction) become significant in comparison to the desired power-take-off control force, affecting the total control force applied to the PA and therefore leading to significantly less power absorbed by the PA.

This paper investigated the design of model scaled experiments on a fully submerged point absorber WECs within a standard wave flume facility (32x1.3x1m) located at the University of Adelaide, with an emphasis on the methodology of mitigating the influence of parasitic loss in the model scaled experiments. The method was briefly introduced in [7], however more details and new results are discussed in the current paper. The arrangement of the paper is as follows. Section II presents the potential wave theory used for the benchmark of the experimental results. Section III discusses the experimental set-up in the wave flume, in particular the design of the power-take-off control system for addressing the parasitic loss issue. Results and conclusions are discussed in Section IV and Section V respectively.

II. POTENTIAL WAVE THEORY

The hydrodynamics of the fully submerged point absorber WECs can be solved using potential wave theory, assuming the fluid is incompressible and inviscid, which is reliable as long as the incident waves are small [8]. The computed hydrodynamics of the PA can then be assembled in the frequency-domain. The resulting frequency-domain model can only be used to simulate the behaviour of the PA under regular wave conditions and does not have the capability to model nonlinearities such as drag forces and nonlinear power-take-off (PTO) forces. Alternatively, the hydrodynamics can be substituted into the Cummins equation [9], a deterministic solution originally developed to investigate ship dynamics in the time-domain. The resulting time-domain model is able to include nonlinear forces and can be used for simulation under both regular and irregular wave conditions.
A. Frequency-domain Model

The PA dynamic equation takes the following form in the frequency-domain:

\[
(j \omega (M + A(\omega)) + B(\omega)) \tilde{x}(\omega) = \tilde{F}_e(\omega) + \tilde{F}_b + \tilde{F}_{pto},
\]

where \( M \) is the mass matrix of the buoy; \( A(\omega) \) and \( B(\omega) \) are the hydrodynamic added mass and radiation damping matrices, which vary with wave frequency \( \omega \); \( \tilde{F}_e(\omega) \) is the wave excitation force, which is also frequency dependent; \( \tilde{F}_b \) is the net buoyancy force; \( \tilde{F}_{pto} \) is the PTO control force; \( \tilde{x} \) is the buoy velocities in heave, surge, and pitch under the assumption of plane incident waves. The hydrodynamic terms \( A(\omega), B(\omega) \) and \( \tilde{F}_e(\omega) \) can be solved using boundary element solvers (e.g. WAMIT, ANSYS AQWA, NEMOH) for specific buoy shapes. For generic buoy shapes (e.g. sphere and cylinder), the coefficients can be solved using an analytical model developed by [10] and [11].

B. Time-domain Model

The Cummins model takes the following form in the time-domain:

\[
(M + A(\infty)) \ddot{x}(t) + \int_0^t B(t - \tau) \dot{x}(\tau) d\tau = F_e(t) + F_b(t) + F_{pto}(t) + F_d(t),
\]

where \( A(\infty) \) is the infinite-frequency added mass matrix \( A(\omega) \) for \( \omega = \infty \); \( \int_0^t B(t - \tau) \dot{x}(\tau) d\tau \) represents the memory effect of the radiation force, which can be approximated as a transfer function. The wave excitation force time series \( F_e(t) \) can be calculated for both regular waves at a single frequency \( \omega \) and irregular waves based on wave spectra models. Typical wave spectra models are JONSWAP, Bretschneider and Pierson-Moskowitz; \( F_d(t) \) is the quadratic nonlinear drag force that takes the form \( \frac{1}{2} \rho C_d A |\dot{x}| \dot{x} \) on each dimension, where \( \rho \) is the density of seawater, \( C_d \) is the drag coefficient usually determined from experiments for particular buoy shapes, and \( A \) is the cross-sectional area of the buoy perpendicular to the motion direction.

III. EXPERIMENTS

Fig. 1 shows the set-up of the scaled experiments in a standard wave flume facility within the University of Adelaide, which is 32m long, 1.3m wide and 1m deep. A piston-type hydraulically-driven wave paddle is placed at the upstream end of the flume. The up and down motion of the paddle generates propagating waves. A perforated anechoic beach sits at the downstream end of the flume, which is used to prevent waves from reflecting back. A fully submerged PA is placed 5m downstream of the wave paddle. A small pulley is placed at the bottom of the flume forming a mooring point. The PA is anchored via a high strength non-compliant fishing line, which passes through the pulley and winds onto the rope spool within a custom-built PTO system. Six custom-built wave probes are placed around the PA to monitor surrounding wave conditions, two upstream, two downstream, and two at the sides. A DS1104 rapid prototyping dSPACE was coded via Matlab to control the wave conditions generated by the paddle based on the wave probe readings, as well as control the PTO system to apply desired PTO control force on the PA. It also collects data from wave probes and PTO unit for post-processing.

A. Design of PA Buoy Assembly

In this study, a spherical fully submerged buoy was used due to its simple hydrodynamics. Fig. 2 shows the buoy assembly of the PA, whose shell was 3D printed from VisiJet M3. The inner space of the buoy shell is used to place an IMU (LORD MicroStrain 3DM-GX4-25) that measures buoy movements and additional weights that allow the change of the net buoyancy and the centre of gravity of the buoy. The shell of the buoy is fixed via screws and sealed by double O-rings. The IMU cable exits the buoy shell via a cable gland fixed at the top of the shell. A U channel profile is printed at the bottom of the buoy shell, forming an anchoring point for the tether. The dimension of the 3-D printed buoy (0.136m diameter) is 1/73.5 of the full-scale buoy (10m diameter) so that the blockage ratio of the buoy to the flume width is 11%. A higher blockage ratio has the risk of modifying the scattering force on the buoy. The ratio (1/73.5) was also chosen considering the similarity of drag coefficients between small/full scales.

B. Design of Power-take-off Control System

Fig. 3 shows the PTO assembly design that is used to actively generate any desired PTO control force on the PA and therefore to emulate any PTO behaviour. A Maxon RE50 motor drives a rope spool via a shaft coupling and consequently applies a PTO force to the PA via pulley-tether coupling. The shafts of the rope spool and motor are held in position by three rolling-
element bearings fixed to the base. The housing of the motor is attached to the base via an aluminium shell and a Lorenz Messtechnik GmbH D-2209 torque sensor. The torque sensor measures the torque applied by the motor and therefore measures the total PTO control force. A motor encoder measures the motor/tether displacement, and therefore measures the displacement of the emulated PTO.

\[ f_{pto} = -(\rho V - m)g + k_{pto} \cdot \theta + c_{pto} \cdot \dot{\theta} \cdot r^2. \]

(3)

where \( V \) is the displaced water when the buoy is fully submerged; \( m \) is the buoy mass; \( g \) is the gravitational acceleration; \( k_{pto} \) is the model scaled PTO spring stiffness; \( c_{pto} \) is the model scaled PTO damping coefficient; and \( l_d \) is the displacement of the tether. This PTO control force along the tether can be mapped to a torque generated by the motor in the PTO assembly:

\[ T_m = -(\rho V - m)g \cdot r + k_{pto} \cdot \dot{\theta} \cdot r^2 + c_{pto} \cdot \dot{\theta} \cdot r^2. \]

(4)

where \( r \) is the radius of the rope spool; \( \dot{\theta} \) is the instantaneous angular displacement of the motor that can be measured by the motor encoder. This motor control torque is computed in the dSPACE controller and is converted to a motor current command in real-time. The motor current command is input into a Maxon ESCON 70/10 controller via an analog signal. The Maxon ESCON controller operates a current control loop that continuously applies the desired control torque to the motor.

As the scale of the experiment is small, the Coulomb friction within the PTO assembly is significant in comparison to the desired PTO control force, and therefore must be compensated in the motor torque control:

\[ T_m = -(\rho V - m)g \cdot r + \text{sign}(\dot{\theta}) \cdot c_c + k_{pto} \cdot \dot{\theta} \cdot r^2 + c_{pto} \cdot \dot{\theta} \cdot r^2, \]

(5)

where \( c_c \) is the Coulomb friction coefficient of the PTO assembly. Since there is noise within \( \dot{\theta} \) arising from differentiating the encoder reading, \( \text{sign}(\dot{\theta}) \cdot c_c \) is replaced by a relay function and a low pass filter is applied to remove noise within \( \dot{\theta} \) for the PTO damping control. The cut-off frequency of the low-pass filter must be sufficiently high (>50Hz) so that the phase of the PTO damping force is not largely affected, which is important from the power absorption point of view.

By observing Eqs (3-5), the following statements can be made for the design of the PTO assembly:

1) The motor needs to be selected to generate sufficient PTO control force including the static force that overcomes the net buoyancy force of the buoy as well as the dynamic force arising from the desired characteristics of the PTO system.

2) The selection of the rope spool radius \( r \) compromises the maximum PTO control force that can be generated by the motor and the magnitude of the Coulomb friction within the total motor control torque. An optimal value of \( r \) exists depending on the characteristics of the motor and the required PTO control force.

The instantaneous power absorbed by the linear spring-damper PTO system can be estimated by

\[ P_{\text{inst}} = c_{pto} \cdot \dot{\theta}^2 \cdot r^2. \]

(6)

The mean absorbed power can then be obtained by averaging the instantaneous power over a certain period.

IV. RESULTS

Table I illustrates the configuration of the experiments for the assessment of the proposed PTO control system. The parameters of the full scale system are typical for a fully submerged point absorber WEC. Rope spool radius and inertia are 15mm and 0.273kg (along the tether) respectively. The total Coulomb friction coefficient is 0.012Nm, the identification of which is described in Section IV-A. The relay function of the friction compensation term \( \text{sign}(\dot{\theta}) \cdot c_c \) is defined to have a threshold of \( eps \) in Matlab. The cut-off frequency of the low-pass filter on the motor encoder velocity output was set to be 50Hz.

<table>
<thead>
<tr>
<th>TABLE I</th>
<th>EXPERIMENTAL CONFIGURATION</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>PA property</strong></td>
<td><strong>Values in both full-scale and (small-scale)</strong></td>
</tr>
<tr>
<td>Water depth</td>
<td>48m (0.65m)</td>
</tr>
<tr>
<td>Buoys radius</td>
<td>5m (0.068m)</td>
</tr>
<tr>
<td>Submergence depth (buoy centre to water surface)</td>
<td>8m (0.109m)</td>
</tr>
<tr>
<td>Buoy mass</td>
<td>264048kg (0.665kg)</td>
</tr>
<tr>
<td>Buoy mass to buoyance ratio</td>
<td>0.5</td>
</tr>
</tbody>
</table>
A. Identification of Friction in the PTO assembly

Free rotation tests were implemented on the PTO assembly for the identification of the friction components in the PTO assembly. The motor was run at various speeds, with the equivalent motor torques measured by the torque sensor. At each motor speed, when the motor reached steady state and therefore the torque balance was achieved in the PTO assembly, the motor torque was equal to the total friction torque in the PTO assembly. Fig. 4 shows the test results. The PTO assembly has a stiction of 11mNm, a Coulomb friction of 6mNm, and a viscous friction coefficient of 0.018Nm.s/rad. The viscous friction is negligible for our experiments because the motor velocity is within 6rad/s. It is difficult to compensate for the stiction given its complex dynamic behaviour. In addition, the system is not going to experience stiction as long as it is moving. Therefore, only the Coulomb friction force is compensated in the PTO control algorithm as shown in Eq. (5), where \( c_r \) is set to be 12mNm consisting of the Coulomb friction within the PTO assembly (6mNm) and the Coulomb friction within the pulley placed at the bottom of the wave flume that the tether goes through (6mNm).

B. Dry Tests for the Validation of PTO Control

Before conducting testing in the wave flume, tests were conducted in air where the buoy was attached to the PTO system via the fishing line without interacting with water in order to validate the PTO control algorithm as shown in Eq. (5). The virtual spring \( k_{pto} \) was set to the optimal values for various regular wave conditions, which were determined from time-domain simulations. The virtual damper \( c_{pto} \) was set to be zero so that if the friction compensation is successful, the system will undergo an undamped oscillation. Free oscillation tests were undertaken by releasing the buoy 30mm away from its equilibrium position under spring-damper and friction compensation control. Results are shown in Table II, where the analytical natural frequency was calculated from \( \sqrt{k_{pto}/m} \) and both the measured natural frequency and damper were measured from the free oscillation histories of the buoy. It was evident that under the proposed PTO control, the buoy oscillated around its analytical natural frequency and the damping residue within the system after friction compensation was negligible in comparison to the optimal damping to be applied in the wave flume experiments (usually higher than 3 N.s/m).

<table>
<thead>
<tr>
<th>Spring value N/m</th>
<th>Damper value N.s/m</th>
<th>Analytical natural frequency Hz</th>
<th>Measured natural frequency Hz</th>
<th>Measured damping residue N.s/m</th>
</tr>
</thead>
<tbody>
<tr>
<td>90</td>
<td>0</td>
<td>1.559</td>
<td>1.561</td>
<td>0.0743</td>
</tr>
<tr>
<td>60</td>
<td>0</td>
<td>1.273</td>
<td>1.279</td>
<td>0.1227</td>
</tr>
<tr>
<td>30</td>
<td>0</td>
<td>0.900</td>
<td>0.901</td>
<td>0.0924</td>
</tr>
</tbody>
</table>

C. Tests in the Wave Flume

The optimal PTO settings for six typical regular and irregular wave conditions were determined from time-domain simulations subjected to a buoy heave motion constraint of 3m. The 3m constraint was set to prevent the buoy from moving out of the water and therefore avoid the occurrence of higher order nonlinear forces in the experiments. Table III shows the six wave conditions (e.g. three regular wave conditions and three irregular wave conditions) used in simulations and experiments, as well as the corresponding optimal PTO settings applied.

<table>
<thead>
<tr>
<th>Wave amplitude or (height) (m)</th>
<th>Wave period (s)</th>
<th>*energy period for irregular wave</th>
<th>Regular waves ( k_{pto} ) N/m</th>
<th>Regular waves ( c_{pto} ) N.s/m</th>
<th>Irregular waves ( k_{pto} ) N/m</th>
<th>Irregular waves ( c_{pto} ) N.s/m</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5 (3)</td>
<td>7</td>
<td>89.6</td>
<td>4.1</td>
<td>71.1</td>
<td>3.1</td>
<td></td>
</tr>
<tr>
<td>1.5 (3)</td>
<td>9</td>
<td>54</td>
<td>4.8</td>
<td>53.7</td>
<td>2.9</td>
<td></td>
</tr>
<tr>
<td>1.5 (3)</td>
<td>12</td>
<td>29.7</td>
<td>4.3</td>
<td>35.7</td>
<td>4.3</td>
<td></td>
</tr>
</tbody>
</table>

Fig. 5 shows the power output of the PA resulting from the frequency-domain simulation (Eq. (1)), the time-domain simulation (Eq. (2)), and the scaled experiments with friction compensation (Eq. (5)) and without friction compensation (Eq. (4)) under three regular wave conditions. For experiments, at each wave period, the mean power and the standard deviation was calculated from eight experimental runs in the wave flume, where the mean power in each run was averaged over 60 wave periods. As evident in Fig. 5, the frequency-domain simulation overestimated the power absorption efficiency of the PA because the nonlinear drag force was not considered. The experiments without friction compensation significantly underestimated the power absorption capability of the PA because the Coulomb friction within the PTO assembly absorbs the majority of the power arising from the waves. The experiments with friction compensation provided results similar to the ones obtained from time-domain simulation for...
wave periods of 7s and 9s, with errors within 10%. Even with friction compensation, the experiments at 12s wave period significantly underestimated the power. In addition, there was a large deviation between runs.

To understand the cause of this issue, the time histories of the control torque components on the motor under regular waves of 7s and 12s periods (Fig. 6 and Fig. 7) are analysed. For the 7s wave period case, the magnitude of the dynamic (e.g. spring + damper) control torque was much higher than the magnitude of the stiction torque in the PTO assembly (e.g. 11mNm) and friction compensation worked properly when the buoy changed its motion direction (e.g. the compensation torque always had the sign that was opposite to the Coulomb friction). However, for the 12s wave period case, the magnitude of the dynamic control torque was about the same as the stiction torque in the PTO assembly. Furthermore, the dynamic PTO control torque reached zero during the direction change of the buoy motion (e.g. at 51.3s, 52.7s and 54.1s), in which circumstance the motion of the buoy had a high risk to be caught by the motor stiction (e.g. 54.1s onwards). When the buoy was caught by stiction, no power was generated in the PTO until the buoy was forced to move again. Therefore, power was lost at partial cycles and the number of cycles caught by stiction were random due to the complex dynamics of stiction.

To further validate the conclusion drawn from the previous analysis, additional experimental tests were undertaken for the 12s wave period case, however, with a higher wave amplitude (2m) that resulted in higher wave excitation. Fig. 8 shows the power outputs of the PA under regular waves of 12s period and 1.5m and 2m amplitudes respectively. It was evident that at wave amplitude of 2m, the experiments provided similar results to the ones obtained from time-domain simulations. The time histories of the control torque components on the motor at regular wave of 12s period and 2m amplitude are shown in Fig. 9. It was evident that the dynamic PTO control torque under 2m amplitude wave was no longer dominated by motor stiction. In addition, the dynamic PTO control torque is non-zero during the direction change of the buoy motion as highlighted in Fig. 9.
Experimental validation was also implemented for the three irregular wave conditions as shown in Table III, with results shown in Fig. 10. At each energy wave period, the mean power and the standard deviation was calculated from eight experimental runs in the wave tank, where the mean power in each run was averaged over 300 wave periods. It was evident that at all the wave periods, the mean power output of the PA obtained from experiments were slightly lower than those obtained from the time-domain simulations. In addition, the standard deviation in the power output between experimental runs were significant. This was due to the low amplitude components at the bounds of the spectrum (Pierson-Moskowitz) of the irregular wave, resulting in some small waves in the whole wave series that were not able to generate excitation force higher than the motor stiction. Therefore, the power output was sensitive to the stiction dynamics for these small waves.

V. CONCLUSIONS

A PTO control system has been designed for the model scaled experimental study on point absorber WECs. A simple friction compensation algorithm has been tested in both regular and irregular wave experiments, with results compared to those obtained from numerical solvers. In general, the PTO control system worked well and provided good estimation to the full-scale system. However, in particular cases, where the dynamic PTO control torque is close to the magnitude of the stiction torque in the PTO assembly, the experiments have the risk to significantly underestimate the power absorption capability of the PA. This issue can be addressed by slightly increasing the scale of the model in the experiments. The authors of this paper are in the process of determining the optimal model scaled size for the wave flume experiments on point absorber WEC, considering both the effects of the parasitic loss in the PTO assembly and the effects of buoy to flume blockage ratio. On the other hand, a better PTO assembly design is also under investigation, with a physical torsional spring in parallel to the motor to support the static PTO force required to overcome the net buoyancy force and also provide the majority of the PTO spring force so that a smaller motor with less stiction and parasitic loss can be chosen. Follow-up findings will be presented at 2017 EWTEC.

REFERENCES


