A study of reactively controlled floating point absorber in wave tank experiments

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1 Introduction

Floating point absorbers (FPA) normally float on the free-surface and operate primarily in the heave (vertical) direction to generate electrical energy which is extracted by means of a power-take-off (PTO). The PTO also provides the control force that tunes the FPA into optimal operational condition (e.g. resonance) so that the power absorption efficiency is significantly increased. The design of PTO devices for FPAs is a complex task and an ongoing research topic. Previous studies have used a variety of simplified systems to simulate PTO devices. Yeung[1] constructed a permanent magnet linear generator to obtain a maximum energy extraction efficiency of 96.34% from an asymmetric heaving floater in waves. More researchers preferred to use PTO simulators based on passive elements. Yu and Li[2] applied a miniature hydraulic cylinder as a damper to absorb wave power within their FPA model. Flocard and Finnigan[3] reported using three PTO damping settings from a PTO simulator based on a rotational viscous dashpot. Other simulators, such as Oil-filled dashpots and pneumatic dampers were also employed commonly in many references. Although the simplicity and low cost of passive element PTO simulators are attractive, these systems provide only low resolution adjustments of the PTO force and do not allow investigations of alternative PTO control schemes. Meanwhile, the PTO devices mentioned above can only obtain the maximum possible energy conversion efficiency at natural frequencies and are unable to tune floaters to resonance in specific wave conditions. Away from resonance, however, they generally have poor performance of wave energy conversion.

In this study, we regarded the PTO system as a linear spring-damper. An electric, feedback controlled motor which could provide an arbitrary defined force with high resolution was used to apply reaction that represented the forces applied to a FPA WEC by a PTO device. An inviscid hydrodynamics modelling resulting from wave-body interactions[4] helped to define the active PTO control parameters to tune the FPA model to resonance and to obtain the maximum wave energy conversion. A regular wave test of the FPA model including the axisymmetric floater and the present PTO simulator was conducted in the Muti-function towing tank of Shanghai Jiaotong University. A numerical model using a boundary element method (BEM)[5] was used to predict the power performance of the FPA model that operates predominantly in heave based on linearized potential flow theory. The additional viscous damping is modelled as a quadratic drag force. By comparison, we observed that the numerical simulation results agreed well with the experimental data in most wave conditions even though the linear hydrodynamics modelling has its limitation. The methods developed in this work could be applied to study FPAs in model-scale experiments and even other wave absorbers such as breakwaters.

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2 Numerical modelling

In this study, a heaving FPA is considered and the viscous damping force considered as a the quadratic drag force. The FPA dynamic equation takes the following form in the frequency domain:

$$\left(-\omega^2[m + A(\omega)] - \mathrm{i}\omega[B(\omega) + C_{pto}] + K + K_{pto}\right) z = f_e(\omega) + f_v, \tag{1}$$

where m is the mass of the floater; $A(\omega)$ and $B(\omega)$ are the hydrodynamic added mass and radiation damping, which vary with wave frequency; K is the hydrostatic restoring stiffness in heave; $f_e(\omega$ is the wave excitation force, which is also frequency dependent; f_v is the viscous damping force; C_{pto} and K_{pto} are the spring stiffness and power absorption damping of PTO; z is the complex amplitude of floater's heave displacement. In this study, the inviscid added mass, radiation damping and the wave excitation force were calculated through the use of a frequency domain BEM solver SJTU-WEB[5] for specific buoy shapes. To represent the viscous damping on floaters, a nonlinear drag force is assumed, which is proportional to the square of the floater velocity[6]:

$$F_{\mathrm{v}} = -\frac{1}{2}\rho A_{\mathrm{s}} C_{\mathrm{d}} \dot{Z}|\dot{Z}|, \tag{2}$$

where A_{s} is the characteristic area of bodies perpendicular to the direction of motion; ρ is the density of water; C_{d} is the viscous drag coefficient usually determined from experiments for particular buoy shapes. In the frequency domain analysis, the nonlinear drag force can be substituted by a linear term using the Lorentz linearization

$$f_{\mathrm{v}} = \frac{4\mathrm{i}\omega^2}{3} A_{\mathrm{s}} C_{\mathrm{d}} \|z\| z, \tag{3}$$

Since this damping force depends on the heave motion response z, the solution to Eq.(1) is weakly nonlinear and must be found iteratively at each frequency and incident wave height.

Using the present dynamic modelling, for a heaving single-body FPA, the PTO spring stiffness required to tune floaters to resonance can be evaluated by[9]

$$K_{\mathrm{PTO}} = \omega^2[m + A(\omega)] - K, \tag{4}$$

It can be seen from Eq.(4) that for a single-body FPA, the desired value of the PTO spring stiffness could be negative due to the presence of the hydrostatic restoring stiffness[7], which is difficult to implement in practice. In the present experiment, we use an electric, feedback controller motor as a PTO simulator to provide desired spring stiffness to make the FPA model resonant in regular waves.

3 Experimental design and setup

Fig. 1 shows a model-scale heaving FPA under reactive control set in the SJTU multi-functional towing tank. The buoy shape applied was a vertical cylinder with hemispheric bottom to minimize vortex shedding. The buoy was mounted on the locked towing carriage, with only its heaving direction free of motion whilst other degrees of freedom were fully constrained. This was achieved by a linear guide, mounted in parallel with an electrical linear motor, both of which were aligned with the heaving axis of the buoy and were connected between the buoy fixation flange and the towing carriage mount. When the buoy was excited by incident waves, two ball bearings on the linear guide afforded the horizontal forces and bending moments arising from buoy-wave interaction, and thus constrained the buoy motion in pure heave. In the meanwhile,

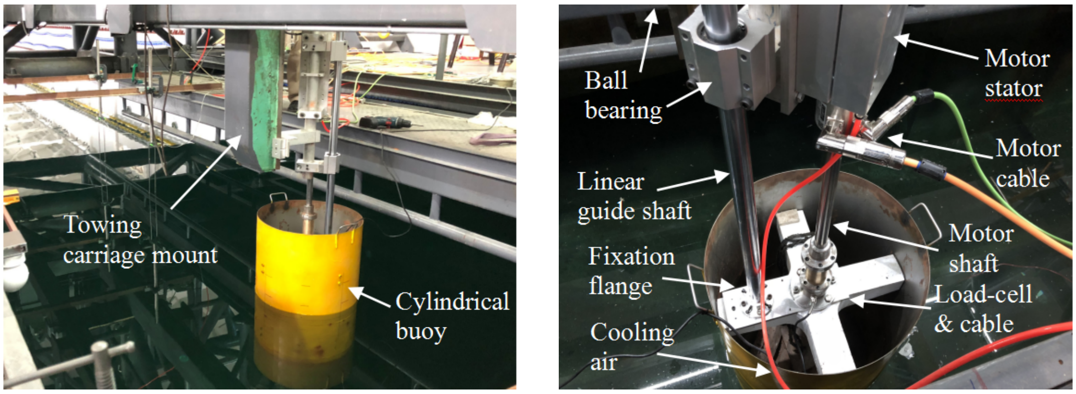


Figure 1: Left: experimental setup in the SJTU multi-functional towing tank; Right: rig components.

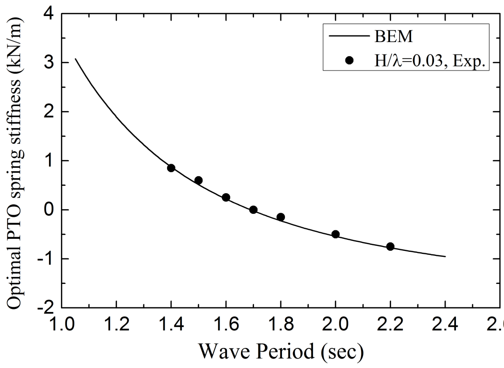
the electrical linear motor (LinMot PS10-70x240U-BL-QJ) applied a high resolution controlled force to the buoy to simulate the desired PTO behaviour (e.g. in this study, a spring and damper PTO force for generality). A cylindrical load-cell was mounted in series with the linear motor shaft to measure the applied PTO control force for both post-analysis and feedback force control. However, the linear motor radiated significant electric noise which contaminated the load-cell reading and made feedback force control impossible. As a result, by the time of abstract submission, only feedforward force control was tested. The algorithm applied was similar to [8], where the total control force was given by:

$$F_c = -K_{pto}Z - C_{pto}\dot{Z} + \text{sign}(\dot{Z})\epsilon \quad (5)$$

where the first term is the spring controlled force, the second term is the damper controlled force, and the last term is a feedforward compensation term that compensates for the dry friction loss of the PTO mechanism. The dry friction coefficient ϵ was determined experimentally using free decay tests. Eq.(5) was solved in the NI MyRIO real-time controller at a loop rate of 100 Hz, where the buoy heave displacement Z, was sampled from the motor encoder. In parallel, the sampled F_c was fed into the motor servo drive (LinMot E1450-EC-QN-0S) and the servo drive ran a 1kHz current control loop to achieve the instantaneous target control force on the buoy.

4 Results and Conclusion

Here we present some preliminary results from wave tank tests and numerical simulations. We first adjusted the PTO spring stiffness to tune the floater to resonance at different wave periods. Figs.2 and 3 show the optimal PTO spring stiffness and heave RAOs when the floater was resonant. It was evident that the present PTO simulator was able to accurately tune the floater to resonance, and the heave response derived from the numerical simulation agreed fairly well with the experimental data. It is worthwhile to note that the optimal PTO stiffness became negative at large wave periods. Fig.4 plots the maximum PTO averaged power absorbed when the floater was resonant, where it was shown that the present numerical model led to substantial over-prediction of wave power output. Considering that viscous damping was included in the numerical model, this disagreement may be attributed to unexpected mechanical friction from the system when the floater was heaving. How to eliminate the friction during highly dynamic force tracking is to be studied in the future. A potential solution is to use feedback force control in combination with feedforward friction compensation.



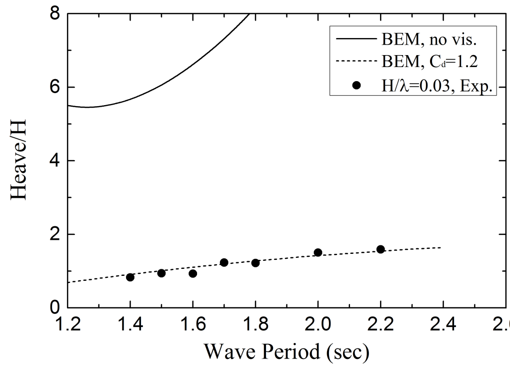


Figure 2: Optimal PTO spring stiffness required to tune the floater to resonance ($C_{pto} = 0$).

Figure 3: Heave RAO of the FPA model in resonance ($C_{pto} = 0$).

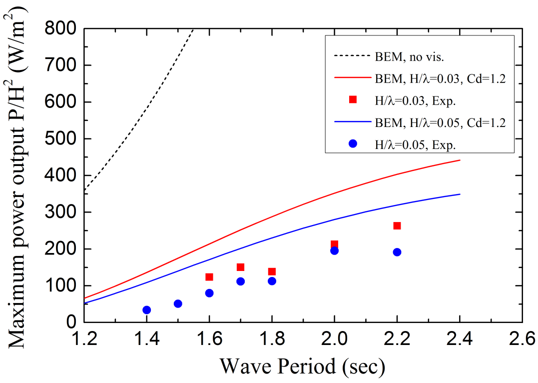


Figure 4: Maximum averaged power outputs of the FPA model under regular waves.

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