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Optimization of nonlinear wave energy converters

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ABSTRACT

This paper presents an optimization approach for the nonlinear control of wave energy converters (WECs). The proposed optimization method also presents the option of optimizing the system nonlinearities, such as those due to the buoy shape, such that the harvested energy is maximized. For the sake of control design, the control force and the system optimizable nonlinear force, each is expressed as a truncated power series function of the system states. The power series coefficients in both the control and system forces are optimized. A hidden genes genetic algorithm is used for optimization. The optimized system's nonlinear force is assumed to drive the design of the WEC. The numerical test cases presented in this paper show that it is possible to attain multiple fold higher harvested energy when using nonlinear control optimization. The advantage of being able to optimize the WEC design simultaneously with the control is the potential of harvesting this multiple fold higher energy without causing large WEC motion and with less dependence on reactive power. While this paper focuses on the optimization part of the problem, the implementation of the obtained control in realtime is discussed at the end of the paper.

1. Introduction

One of the challenges in wave energy harvesting is the buoy motion control. There has been significant developments on different control methods for wave energy converters (WECs) (Falnes, 2007). Most studies on the control of one-degree-of-freedom heaving WECs adopt a linear dynamic model (the Cummins' equation (Cummins)) which can be written as:

$$(m+\tilde{a}_\infty)\ddot{z} = f_{ex} + u - B_v\dot{z} - kz + f_r \tag{1}$$

where z is the heave displacement, m is the buoy mass, k is the hydrostatic stiffness due to buoyancy, $\tilde{a}_\infty$ is the added mass at infinite, f_{ex} is the excitation force, u is the control force, B_v is a viscous damping coefficient, and f_r is the radiation damping force. The buoyancy stiffness term is called the hydrostatic force. This is linear model is usually implemented using boundary element methods (BEM) that assume small motions around the mean position.

There are, however, multiple sources of possible nonlinearities in the WEC dynamic model (Wolgamot and Fitzgerald, 2015). For example, if the buoy shape is not a vertical cylinder near the water surface then the hydrostatic force will be nonlinear. The coupling between the heave and pitch modes in a point absorber is nonlinear (Villegas and van der Schaaf, 2011; Zou et al., 2017). The hydrodynamic forces can also be nonlinear in the case of large motion (Giorgi et al., 2016). Control strategies that aim at maximizing the harvested energy usually increase the motion amplitude and hence the impact of these nonlinearities increases. Reference (Giorgi et al., 2016) presents a numerical analysis for the nonlinear hydrodynamic forces at different levels from a full nonlinear model using computational fluid dynamics (CFD) tools, to linear models corrected by the nonlinear Froude-Krylov force as well as nonlinear viscous and hydrostatic forces. The power take off (PTO) unit may have nonlinearities as well (Bacelli et al., 2015). Reference (Retes et al., 2015a) points out that different WEC systems should choose only the relevant nonlinear effects to avoid unnecessary computational costs. For example, in the case of heaving point absorbers the nonlinear Froude-Krylov force is essential while the nonlinear diffraction and radiation can be neglected; the nonlinear viscous effects are weak as well for point absorbers (Retes et al., 2015a). The nonlinear PTO and mooring effects seem to be significant. In fact references (Merigaud et al; Retes et al., 2015b) focus on the nonlinear FroudeKrylov forces and show that they are the dominant nonlinearities in the case of a heaving point absorber with nonuniform cross sectional area.

Reference (Giorgi, 2017) discusses representative linear models that provide an average model over the full operational space; these models are more accurate than the linear models in the cases of large motions. Yet, due to the average nature of these representative models, they may not be very useful in controlling a WEC in large motion. Linearizing the WEC motion about an operation point is also not always feasible due to the fact that ocean waves change characteristic continuously, and hence

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there is no single operation point. Reference (Penalba et al., 2017) points out that for WEC control in the case of large motions, nonlinear models are inevitable.

Reference (Giorgi and Ringwood, 2016) presents a computationally efficient way of computing the static and dynamic nonlinear Froude-Krylov forces. Two methods are presented in (Giorgi and Ringwood, 2016). The first method assumes that the wave length is considerably longer than the characteristic length of the device and hence ignores the dependance of the pressure (on the buoy surface) on the surge coordinate. The second method uses a McLaurin expansion to simplify the force integral calculation. The latter method was demonstrated to be more accurate for various sea states.

Most of the studies described above consider the modeling of different nonlinearities of a given WEC, and use the relevant ones in modeling the system. Reference (Giorgi and Ringwood, 2016) found that the harvested energy of the nonlinear WEC is less than that of the linear system, when using a latching control (Babarit et al., 2004; Durand et al; Clément and Babarit, 1959). Nonlinear systems, however, possess some characteristics that can be exploited for higher harvested energy as pointed out in (Nayfeh and Mook, 2008; Robinett and Wilson, 2011). This paper addresses the nonlinear one-degree-of-freedom heaving WEC from a different prospective. The goal here is not to model the nonlinearities in the system. Rather, the goal is to increase the harvested energy of the nonlinear WEC compared to the linear one. One way to increase the harvested energy of a nonlinear WEC is to design a controller for the nonlinear WEC; in other words the control optimization process should take into account the nonlinearities in the WEC, allowing the control force to be nonlinear function of the system states. The energy can be further increased if, in addition to optimizing the control, we optimize the system nonlinearities simultaneously with the control. The nonlinear FroudeKrylov force, for instance, is dictated by the buoy shape; hence the buoy shape can be optimized along with the control to maximize the energy. The FroudeKrylov force is one source of the system nonlinearities. In this paper, the hydrodynamic and hydrostatic (hydro) forces along with all other optimizable nonlinear forces are referred to as the system nonlinearities. The system nonlinearities and the control (also nonlinear) are here optimized simultaneously. For the sake of control design, it is convenient to express the optimizable system nonlinearities as a series function as follows:

$$\tilde{f}_s = \sum_{i=1}^{Ns} \alpha_{s_i} z^i + \sum_{j=1}^{Ms} \beta_{s_j} \left| \dot{z}^j \right| sign(\dot{z}) \tag{2}$$

where $\tilde{f}_s$ is the nonlinear force, α_{s_i} and β_{s_i} are constant coefficients, $\forall\ i$; N_s and M_s are the number of nonlinear terms that determine the order of the nonlinear forces. Eq. (2) is written intuitively; consider for example the Proportional-Derivative (PD) controls which are widely used in linear systems. In a PD control, the proportional part is constructed as linear term in the state, and the derivative term is constructed as a linear term in the state derivative. The proportional term is a stiffness term since it has spring-like effect, which means this part of the force does not add/remove energy on average. The derivative term, however, is a damper-like term, and it continuously adds/removes power. One might think of nonlinear stiffness or damping terms, as discussed in details in several references such as (Nayfeh and Mook, 2008). The first term in $\tilde{f}_s$ represents a nonlinear stiffness force, and the second term contains a nonlinear damping force. Note that all β_{s_j} are always negative coefficients, and hence the second term is always a damping term (energy flow is always from the water to the device). Optimizing the system nonlinearities means in this case finding the optimal coefficients α_{s_i} and β_{s_i}. Once the control and $\tilde{f}_s$ are optimized, the WEC system (e.g. the buoy shape) is designed so that the WEC nonlinear force matches the optimized nonlinear force $\tilde{f}_s$. This last step of designing a WEC system to generate a prescribed nonlinear force is not addressed in this paper; the focus of this paper is on the optimization of $\tilde{f}_s$ and the control. Section 4.3, however, presents a numerical case study for demonstration of optimizing both the control and the buoy shape simultaneously. The cases when the optimized $\tilde{f}_s$ cannot be realized will also be discussed in Section 6. Also the case when there are nonlinear forces in the system that are not optimizable is addressed in Section 6.

2. Dynamic model of the nonlinear WEC system

This section presents the dynamic model that will be used to approximate a nonlinear WEC, for the purpose of control design. The nonlinearities in the dynamic model could be because of the buoy shape, the large buoy motions, and/or the PTO. To model a nonlinear WEC, we start with a linear approximation that has a buoy of height h. This is here referred to as the baseline model. To simplify the presentation we start by assuming a regular wave, then the work is extended to irregular waves in Section 5. For the case of a linear WEC in a regular wave, the radiation force reduces to a linear damping and an added mass term. The equation of motion in Eq. (1) then becomes:

$$(m + \tilde{a})\ddot{z} + (c + B_v)\dot{z} + kz = f_{ex} + u \tag{3}$$

where c is the radiation damping coefficient, and $\tilde{a}$ is the added mass at the excitation frequency. The excitation force in this case can be written as:

$$f_{ex} = \hat{f}\ cos(\Omega t + \phi) \tag{4}$$

where Ω is the excitation force frequency, $\hat{f}$ is the amplitude of excitation force, and ϕ is the phase of excitation force.

Now consider the case of a nonlinear WEC in which the additional nonlinear force, compared to the baseline model in Eq. (3), is $\tilde{f}_s$. The nonlinear force $\tilde{f}_s$ is expressed as in Eq. (2). The control force is expressed as a summation of two quantities $u = u_l + \tilde{u}_c$, where u_l is the linear part of the control, and $\tilde{u}_c$ is the nonlinear control part which is assumed in the form:

$$\tilde{u}_c = \sum_{i=1}^{Nc} \alpha_{c_i} z^i + \sum_{j=2}^{Mc} \beta_{c_j} \dot{z}^j \tag{5}$$

where α_{c_i} , β_{c_i} are constant coefficients, $\forall\ i$; N_c and M_c are the number of nonlinear terms that determine the order of control forces. The equation of motion of the system then is:

$$(m + \tilde{a}_\infty)\ddot{z} + B_v\dot{z} + kz = f_{ex} + f_r + u_l + \tilde{u}_c + \tilde{f}_s \tag{6}$$

The equation of motion, Eq. (6), is derived assuming that the buoy does not leave the water nor gets fully submerged in the water. In the case of nonlinear WECs presented in this paper, the motion of the buoy may grow large and these two cases should not be excluded. Hence the model in Eq. (6) is modified as follows. Consider the coordinates defined in Fig. 1, a range $|z| < z_s$ is defined in which the model in Eq. (6) is considered valid. The limit z_s is selected based on the buoy dimensions and the wave height. When $|z| > z_s$, there are two possible cases. The first case is when ($z > 0$), that is the buoy is (or very close to being) fully submerged under water. The second case is when ($z < 0$), that is the buoy is (or very close to being) totally out of the water. In these two cases, the dynamic model in Eq. (6) is not valid, and an approximate dynamic model is defined as follows:

<u>Case 1:</u> ($z > 0$) The linear stiffness term becomes a constant $kh/2$.

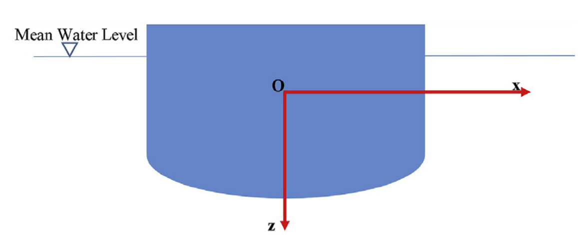

Fig. 1. Buoy coordinate system.