Hydrodynamics of OWC wave energy converters

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ABSTRACT: This work deals with the numerical studies on hydrodynamics of oscillating water column (OWC) wave energy converters and its damping optimization on maximizing wave energy conversion by the OWC device. As a fundamental step, the hydrodynamic problems have been systematically studied by considering the interactions of the wave-structure and of the wave-internal water surface. Our first attention is on how the hydrodynamic performance can be reliably assessed, especially when it comes to the time-domain analysis, and what the physics behind the considerations is. Further on, a damping optimization for the OWC wave energy converter is also present based on the dynamics of the linear system, and a study on how we can optimize the damping for the given sea states so that the power conversion from irregular waves from irregular waves can be maximized.

1 INTRODUCTION

OWC wave energy converters have been regarded as one of the most promising wave energy converters, and probably the most practical wave energy converters so far. It is reported that the LIMPET OWC plant has generated electricity to the grid for more than 60,000 hours in a period of more than 10 years (Heath [1]), whilst a more recent development is the Mutriku OWC wave energy plant in Spain [2], a multi-OWC wave energy plant with a rated power of 296kW, consisting of 16 sets of "Wells turbines + electrical generator" (18.5 kW each), is estimated a electricity generation of 600 MWh so far[1].

The advantages of the OWC wave energy converters are their unique features in wave energy conversion. In the OWC wave energy converters, the air flow is normally accelerated from the very slow airflow in the chamber (driven by the internal water surface motion) to a quite high speed airflow through the power take-off system by 50-150 times (suppose the power take-off (PTO) air passage area ratio is taken 1:50-1:150 to the water column sectional area). This much accelerated air can effectively drive the air turbine to rotate in a high speed, typically from a few hundreds RPM for the impulse turbines to a few thousands RPM for the Wells turbines (O'Sullivan et al.[3]). The high-rotational speed of the air turbine PTO allows a direct connection to the generator and thus the bulky gearbox may not be necessary. More importantly, for a given power take-off, the high rotational speed mean also a small force or torque acting on the PTO system, which in turn ensures a high reliability in power take-off systems.

To understand and improve wave energy conversion by the OWC devices, numerical methods have been developed, but not yet well validated. Earlier theoretical work on the hydrodynamic performance of OWCs has shown that OWC devices could have a high primary wave energy conversion efficiency if the optimized damping can be attained (Sarmento et al [4], Evans [5] and Evans & Porter [6]). For the more complicated and practical OWC devices analytical solutions may be difficult or even non-existent. However, the boundary element method (and the relevant commercial software, such as WAMIT, ANSYS AQWA et al) can be readily available for any complexity of the geometries, but reliable numerical assessments have not been well established for OWC wave energy converters so far. In this research, the focus is on the hydrodynamic performance of the OWC wave energy converters, which is a prerequisite step in the overall performance assessment for an oscillating water column

[1] EVE, Mutriku OWC Plant, http://www.fp7-marinet.eu/EVE-mutriku-owc-plant.html (accessed on 10/05/2014).

wave energy converter for further applications in coupling with the actual air turbine PTOs. As a fundamental research, the aim is on how the hydrodynamics can be studied and how the optimization can be done though this has been on a simplified case.

2 METHODOLOGY

2.1 Frequency domain analysis

Potential theory has been well developed in the last century and widely used in marine and offshore applications. More recently, the principle has been applied in wave energy conversions, including the oscillating water column devices. In this research, a generalized OWC wave energy converter has been our target, hence the well-developed boundary element method (BEM) is used throughout in the research.

Based on the assumption of the potential flow, the velocity potential of the flow around the floating structure satisfies the Laplace equation:

$$\nabla^2\varphi = 0 \tag{1}$$

where φ is the frequency-domain velocity potential of the flow around the floating structure.

An earth-fixed coordinate system is defined for the potential flow problem. The coordinate is fixed in such a way that the *x*-*y* plane is on the calm water surface and *z*-axis positive up vertically. In the coordinate, the free surface conditions can be expressed in the frequency domain (see Lee et al. [7]), as

$$\frac{\partial\varphi}{\partial z} - \frac{\omega^2}{g}\varphi = \begin{cases} 0, & (on\ S_f) \\ -\dfrac{i\omega}{\rho g}p_0, & (on\ S_i) \end{cases} \tag{2}$$

where ω is the wave frequency, ρ the density of water, g the acceleration of gravity, p_0 the pressure amplitude acting on the internal free surface, S_i the internal free surface in the water column, and S_f the free surface, excluding the internal free surface.

It must be noted that the pressure amplitude acting on the internal free surface is an unknown, which must be solved when a power take-off system is applied.

Hydrodynamically, the water surface in an OWC can be regarded as a moonpool, which have been found applications in the operations of offshore platforms and studied theoretically and numerically ([8-11]). The difference between a moonpool and an oscillating water column may be that the application of the power take-off system in the OWC wave energy converters will apply a reciprocating pressure (the alternative positive and negative chamber pressure, and they may be nonlinear if the nonlinear PTO is applied) on the internal water surface, which would make the problem more complicated.

To solve the linear hydrodynamic problems in the OWC wave energy devices, different approaches have been developed and used. The popular approaches include the massless piston model [7, 12] and the pressure distribution model (Evans et al. [5]). In the former approach, the internal free surface is assumed to behave as a massless rigid piston (a zero-thickness structure), and the target solution is the motion of the internal water surface. The internal water surface motion is then coupled with the PTO so that the chamber pressure can be solved. A slightly different version to the massless piston model is a two-body system for the OWCs, in which the first rigid body is the device itself and the second rigid body is an imaginary thin piston at the internal free surface to replace part of the water body in the water column. In the latter approach, the internal free-surface condition is represented in terms of the dynamic air pressure in the chamber (see [13, 14]), and in the numerical simulation, a reciprocity relation must be employed as shown by Falnes [15] so that the conventional BEM method can be still used. However, it must be pointed out that this method may be only applicable for the cases of linear PTOs. However, tank test and field test data have shown the nonlinear chamber pressure contains both wave frequency and high frequency components in regular waves if a nonlinear PTO is used. In contrast, the internal water surface motion remains reasonably linear when a nonlinear PTO is applied for wave energy conversion (see Sheng et al.[16]).

In addition, the physical meaning of the first approach is more obvious, and its implementation in the numerical assessment is quite straightforward. Hence in this research, this approach is applied and studied.

To represent the dynamic system in a general manner, a convention for a two-body system is used: the motion modes of the first body are given by x_i (i=1,2,…,6), corresponding to the first rigid body motions of surge, sway, heave, roll, pitch and yaw, respectively, and the motion modes of the second body are given as x_i (i =7, 8,…, 12), which corresponds to the 6 degrees of freedom motion of the second body, i.e., surge, sway, heave, roll, pitch and yaw.

For a specific problem we are going to deal with in this research, that is, a fixed OWC device, in the hydrodynamic analysis of the two-body system, we will specify the first body as a fixed body, and only the heave motion of the second body, the imaginary piston, is allowed. The extension from the fixed OWC to a floating OWC can be quite straightforward.

For the fixed OWC, the heave motion of the piston in frequency domain can be written as

$$\{-\omega^2[m_{99}+a_{99}]+i\omega b_{99}+c_{99}\}\zeta_9 = f_9 \tag{3}$$

where m_{99} is the masses of the second body; a_{99} the frequency-dependent added masses for the heave motion for the second body; c_{99} the restoring force coefficients; b_{99} the hydrodynamic damping coefficients for heave motion; f_9 the excitations for the second body, and ζ_9 the complex heave motion amplitude of the piston.

2.2 Time domain analysis

For OWC wave energy converters, the whole dynamics may very likely be nonlinear if a linear air turbine PTO take-off system is included, for example, a linear Wells turbine. When a full scale OWC device is considered, the air chamber and the pressure can be large enough, so that the air compressibility in the air chamber can be evident (see Falcao et al [17]), which is essentially nonlinear. If mooring system is included, the nonlinearity can be more obvious when the large motions of the device are considered. For a nonlinear dynamic system, time domain analysis must be employed.

In the time-domain analysis in this research work, the Cummins-Ogilvie hybrid frequency-time domain analysis is used, in which the basic hydrodynamic parameters can be first analysed in frequency domain, then the Cummins time-domain equation is established using the Ogilvie's relation (Cummins [18] and Ogilvie [19]).

To simplify the problem in the oscillating water column wave energy conversion, we assume only the heave motion of the second body is useful for power conversion. The heave motion of the second body can be written in time-domain as

$$[m_{99}+A_{99}(\infty)]\ddot{x}_9(t)+\int_0^t K_{99}(t-\tau)\dot{x}_9(\tau)d\tau+C_{99}x_9(t)=F_9(t) \tag{4}$$

where m_{99} is the mass of the second bodies; $A_{99}(\infty)$ the added masses for the heave motion at the infinite frequency; C_{99} the restoring force coefficients and their interactions; K_{99} the impulse functions for heave motions and their interactions; F_9 the excitations for the second body.

The impulse functions can be obtained if the frequency-domain added mass or damping coefficients have been assessed in frequency domain, and the transform can be done as

$$K_{99}(t)=\frac{2}{\pi}\int_0^\infty b_{99}(\omega)\cos\omega t d\omega \tag{5}$$

where b_{99} is the damping coefficients in frequency domain.

3 PISTON LENGTH

3.1 Natural period of the piston motion

As shown by Evans et al [6], the interior free surface has a natural period, T_0, if the length of the cylinder is much larger than its diameter as,

$$T_0 = 2\pi\sqrt{\frac{D}{g}} \tag{6}$$

where D is the draft of the water depth or the length of the water column and g the gravity acceleration. This formula corresponds to the natural period of a cylinder of a draft D in water without a correction from the added mass.

If the imaginary piston is considered as an isolated cylinder, its added mass for the heave motion has been given according to McCormick ([20]) when the draft D is far larger than its diameter as follows,

$$T_0 = 2\pi\sqrt{\frac{D+0.848R}{g}} \tag{7}$$

where R is the radius of the cylinder.

For a large water column, its natural period of the water surface motion has been studied by Veer et al.[11], and they gave a formula for the calculation of the natural period of the internal water surface motion of a moonpool as,

$$T_0 = 2\pi\sqrt{\frac{D+0.41*S_0^{1/2}}{g}} \tag{8}$$

where S_0 is the sectional area of the moonpool/water column.

A more appropriate approach in obtaining the internal water surface motion and its relevant natural period is to employ the conventional boundary element method (BEM, in this case, WAMIT). In the BEM code, the interaction between the water body and the floating structure is fully accounted. Hence the natural period of the internal water surface would be more accurately calculated by the BEM method.

Table 1Natural periods of the internal water surface

Method	Natural period of internal water surface, T_0
Evans et al.	0.777s
McCormick	0.998s
Veer et al.	0.970s
WAMIT	0.935s

All the natural periods using different formulas, including the one obtained from WAMIT simulation are listed in

Table 1 for a generic cylinder OWC device, with a water column radius of 0.115m, and draft of 0.15m. One can see that those semi-empirical and numerical methods give quite similar estimations to the natural period of the internal water surface motion if an appropriate added mass can be included.

3.2 Piston lengths

As it is well known that, in many cases in studying an OWC wave energy converter, the water column of an OWC device has been represented by a thin piston or a zero thickness structure ([21, 22]). The thin or zero thickness structure is used to replace the internal free surface (see Figure 1). It has been shown theoretically by Falcao et al [21] (also Evan et al. [12]) that the added mass for the thin rigid-body is the entire entrained-water by the water column plus some additional added-mass. This interesting result can be interpreted that the mass of the thin piston plus the entrained water (i.e., the major part of the corresponding added-mass) may be possibly equivalent to that of a full piston.

An extreme case of the thin piston is the zero thickness structure (i.e., a massless piston), which has been also studied by Lee et al. [7, 23] via a method called 'generalised modes', and the generalised modes for the internal water surface motion can be simply specified as the additional motion modes in the boundary element codes, so that a significant modification to the code is not necessary. As can be seen in many practical cases, the thin/massless pistons or the full pistons are both popular in studying the performance of an OWC device. Therefore, there may be a question, what will happen if a certain length of the piston is considered, as shown in Figure 2.

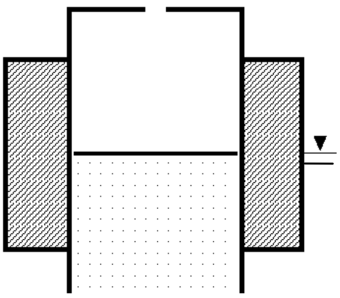


Figure 1 A very thin piston on the internal free surface (L is small or zero)

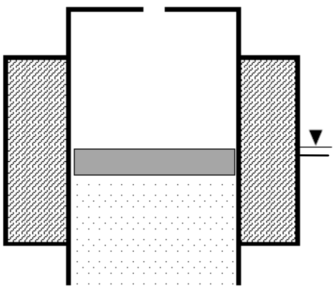


Figure 2 An illustration of a thick piston for representing the internal free surface

3.3 Piston length and responses

For the forementioned OWC device, a longest piston length could be the full length of the water column of 0.15m (i.e., *L*=0.15m, where *L* is the actual length of the piston), and the shortest piston length is zero in the massless piston. In between, the length of the piston could be set as any length between 0.0m and 0.15m. In Figure 3, a comparison of the internal water surface responses for different piston lengths is given (the legends '*L*=0.001m', '*L*=0.10m' et al. indicating the lengths of the pistons). It can be seen that the IWS responses are very close for all four different piston lengths (Figure 3), with the full length of piston giving a slight difference.

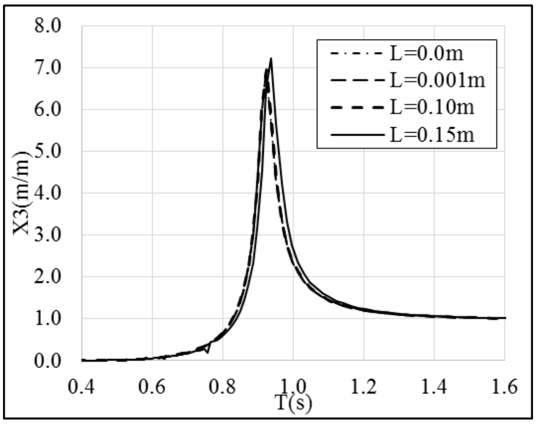


Figure 3 IWS response predictions of piston motion (no PTO damping)

3.4 Piston length and added mass

As given in the time domain equation of the floating structure, an important parameter would be the added mass at infinite frequency, which is an essential part of the overall mass in the dynamic system, and hence does very much decide the resonance period/frequency of the dynamic system.

Table 2 shows the masses and the added-masses at infinite frequency for the pistons and the device from the simulations using WAMIT. For the massless piston (its length *D*=0.0m), the 'generalised modes' has been used, while for the cases of certain lengths of pistons, two-body system is used in WAMIT simulations.

Table 2 Piston mass and its added mass (in kg)

Piston length	0.0m	0.001m	0.01m	0.05m	0.1m	0.15m
M_{99}	0.0	0.04	0.42	2.08	4.15	6.23
A_{99}	-70.1	4.35	6.30	6.43	4.66	2.93
$M_{99}+A_{99}$	-70.1	4.39	6.72	8.51	8.81	9.16

From the table, the added masses for the device heave motion at the infinite frequency are similar except the one in the case of the massless piston, which is obviously ‘wrong’ (a very large negative added mass!). For the cases of certain lengths of the pistons, the overall mass for the piston can be different, and their correctness will be examined later in this research. However, one can see that when the piston length is larger than 0.05m, the overall mass (given by the mass and added mass together) is very similar, though the piston mass itself can be very different (2.08kg for the piston length 0.05m and 6.23kg for the piston length of 0.15m), see Figure 4.

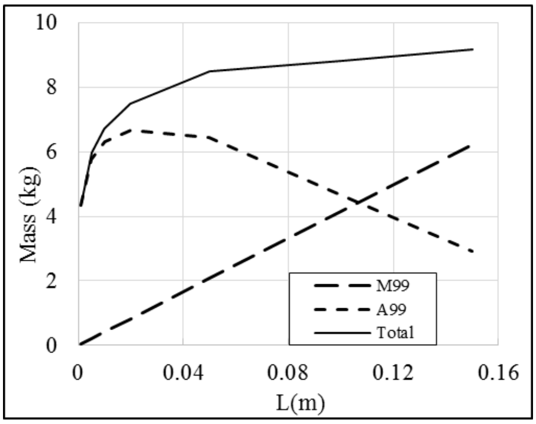


Figure 4 Masses and the added masses of the pistons for different lengths of pistons

4 DAMPING OPTIMISATION

4.1 Dynamic equation for fixed OWCs with a PTO

Following the convention used by Falnes ([15]), the frequency domain equation for internal water surface motions of the OWC device with a linear PTO under wave excitation can be expressed as

$$\left[i\omega(m_{99} + m_{99}) + b_{99} + \frac{c_{99}}{i\omega} \right] v_9 = f_9 - A_0 p \tag{9}$$

where v_9 is the complex velocity amplitude of the internal water surface; A_0 the sectional area of the water column and p is the complex amplitude of the gauge pressure in the air chamber.

4.2 Linear air turbine PTO

For the purpose of analysis, we consider a linear air turbine PTO, for instance, the Wells turbine, due to its roughly linear relation between the flow rate and the pressure across the turbine. For a linear turbine,

$$p = k_1 q_{pto} \tag{10}$$

where k_1 is the damping coefficient of the turbine, and q_{pto} the complex amplitude of the volume flow rate through the air turbine PTO.

We further simplify the analysis by ignoring the air compressibility. The more complicated cases, for example, for floating OWC and the case with air compressibility are possible, but beyond the scope of the research. Hence the flow rate driven by the internal water surface is fully through the air turbine PTO as

$$q_{pto} = A_0 v_9 \tag{11}$$

Combining (10) and (11) yields

$$p = k_1 A_0 v_9 \tag{12}$$

4.3 Solution of the piston motion

Substituting (11) and (12) into (9), we have the dynamic equation of the piston motion in frequency domain as

$$\left\{ i\omega(m_{99} + a_{99}) + \left(b_{99} + k_1 A_0^2\right) + \frac{c_{99}}{i\omega} \right\} v_9 = f_9 \tag{13}$$

and the solution is

$$v_9 = \frac{f_9}{i\omega(m_{99} + a_{99}) + \left(b_{99} + k_1 A_0^2\right) + \frac{c_{99}}{i\omega}} \tag{14}$$

The averaged power converted by the OWC can be written as

$$\overline{P} = \frac{1}{2} p \times q_{pto}^* = \frac{1}{2} k_1 A_0^2 v_9 \times v_9^* \tag{15}$$

where the star indicates the conjugate of the complex variable, so

$$\overline{P} = \frac{1}{2} k_1 A_0^2 \frac{|f_9|^2}{\left(b_{99} + k_1 A_0^2\right)^2 + \omega^2 \left[(m_{99} + a_{99}) - \frac{c_{99}}{\omega^2} \right]^2} \tag{16}$$

4.4 Damping optimisation

The optimised damping of the linear air turbine for maximising the averaged power can be achieve if the following condition is satisfied as

$$\frac{d\overline{P}}{dk_1} = 0 \tag{17}$$

which leads to an optimsed damping coefficient for maximizing wave energy conversion by the OWC device, as

$$k_1 = \frac{\sqrt{b_{99}^2 + \omega^2 \left[(m_{99} + a_{99}) - \frac{c_{99}}{\omega^2} \right]^2}}{A_0^2} \tag{18}$$

It can be observed easily that the optimized damping coefficient k_1 is very much frequency-dependent (see Figure 5).

The maximized captured power is given eq. (19), a same formula for the point absorber (see Falnes [15]) and the corresponding captured power in wave of unit amplitude is seen in Figure 6.

$$\overline{P}_{\max} = \frac{1}{4}\frac{\left|f_9\right|^2}{b_{99}+\sqrt{b_{99}^2+\omega^2\left[\left(m_{99}+a_{99}\right)-\dfrac{c_{99}}{\omega^2}\right]^2}} \qquad (19)$$

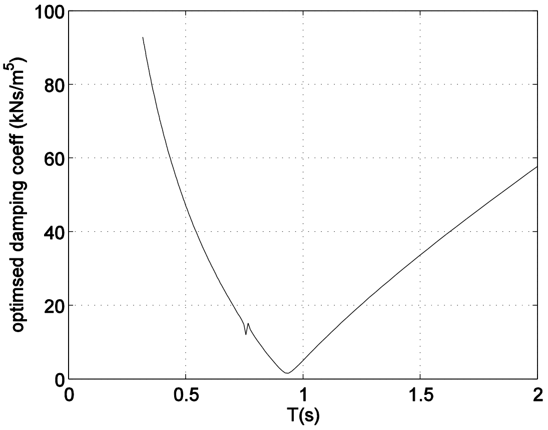


Figure 5 Optimised damping coefficient for the linear air turbine

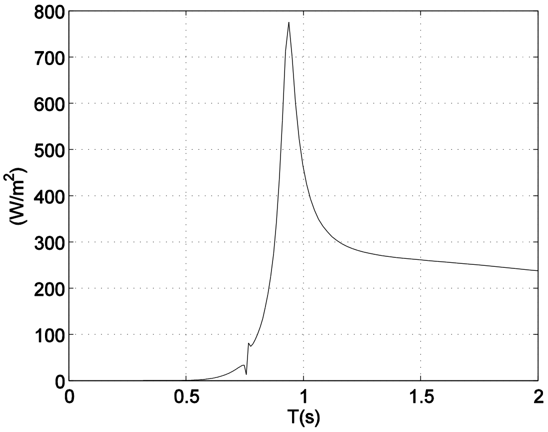


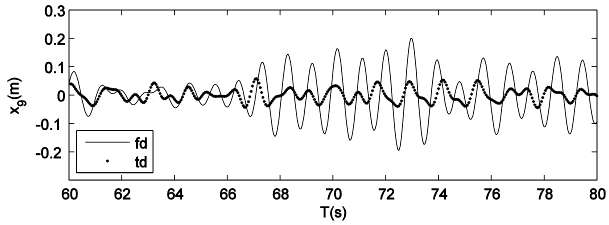
Figure 6 Maximized captured power in wave in unit amplitude

5 RESULTS AND ANALYSIS

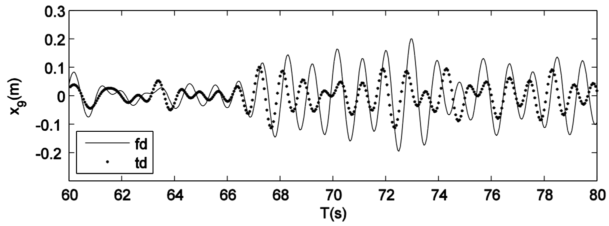
5.1 Time domain analysis

For the time domain analysis, irregular waves of a significant wave height H_s=0.1m, and a mean period of T_{01}=1.0s are chosen due to its closeness to the resonance periods of the piston motion (T_0=0.935s).

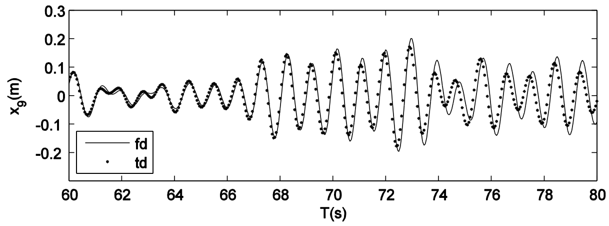
From the comparisons of the time domain analysis, it can be seen that in this particular irregular wave, the effects of the infinite frequency added mass to the motions can be significant. Figure 7a to Figure 7e give the piston motion analysis in time domain for different lengths of the piston. For comparison, the targeted time series are those obtained by combining the frequency-domain responses and the wave elevations (same waves used in the time domain analysis), which can be regarded as the correct motion of the piston. For Figure 7a, the piston length is very small, and its added mass at infinite frequency is also small. Hence an overall mass for the dynamic system is well underpredicted. As a result of this, the corresponding resonance period would be longer than it should be. That's why the piston heave motion prediction is very small when compared to the target time series. Similar situation can be seen for the piston length L=0.01m, which is still small than necessary. but the motion is much better than that of shorter length of piston (see Figure 7b). Further increase the length of the piston, better results are obtained and expected (see Figure 7c to Figure 7e).



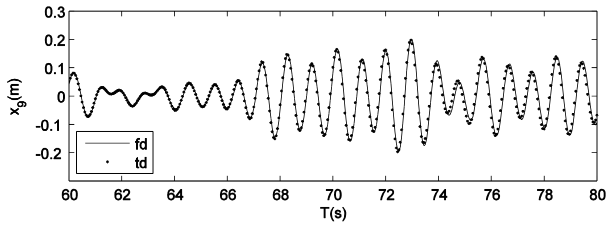
(a) piston length L=0.001m



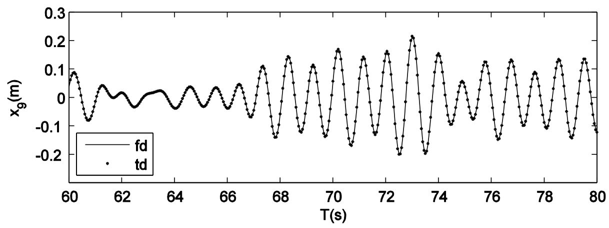
(b) piston length L=0.01m



(c) piston length L=0.05m



(d) piston length L=0.10m



(e) piston length L=0.15m

Figure 7 Heave motions of the pistons in time domain analyses

5.2 Damping optimization for irregular waves

The forementioned optimized damping for maximizing wave power conversion is very frequency dependent, which can not be directly used in irregular waves since in irregular waves, the wave period/frequency and amplitude are changing from wave to wave. Hence a fully optimised damping does not exist. For a practical purpose, a simple but a more practical case for optimising wave energy power conversion is the optimization of the damping for the actual sea state, so that the wave energy conversion may be maximized from the irregular waves. Since the damping level is only set and kept constant for the sea state, the linear theory is still suitable.

Unlike in regular waves where the period and amplitude are well defined, the irregular waves are characterised by the significant wave height and a characteristic wave period if the spectrum is given, and both are statistical values. For instance, the characteristic period can be specified by any of the following periods: spectrum mean period, up-zero crossing mean period, modal period or energy period. If we are trying to optimize the damping for the sea state, which period is the reference period? This is the problem we are discussing here.

As it is well known that in regular waves, the maximizing power capture can be optimized if the device is in resonance, and the maximized capture power can be very close to the theoretical limit. However, this optimised damping coefficient may not be the optimized damping coefficient for producing a maximal power in a sea state. Hence it is often necessary to tune the damping coefficient for the given sea state so that the maximal power can be converted.

Table 3 and Table 4 give the optimised damping coefficients for a linear air turbine based on different characteristic wave periods (note: this is for same waves), namely, spectrum mean period, T_{01}, modal period, T_p and energy period, T_e, respectively, and the corresponding power conversions from two different sea states. It can be seen that the damping coefficients optimised based on the energy period give maximal power conversions.

Table 3 Optimsed damping based on different periods for same sea state (Tp=1.269s, Hs=0.1m)

Period	k1_opt ($kN*s/m^5$)	Pave(W)	Eff (%)
T_{01}=1.00s	4.975	0.224	13.30
T_P=1.296s	22.79	0.247	14.67
T_e=1.110s	11.98	0.260	15.41

Table 4 Optimsed damping based on different periods for same sea state (Tp=1.944s, Hs=0.1m)

Period	k1_opt ($kN*s/m^5$)	Pave(W)	Eff (%)
T_{01}=1.50s	33.61	0.263	10.38
T_P=1.944s	55.07	0.256	10.13
T_e=1.665s	41.85	0.265	10.58

Further evidence can be seen in Figure 8. For the case of T_{01}, the power conversions is increased with the increase of the damping coefficient added to the 'optimised' damping. Obviously, for this case, the system is underdamped; and for the case of T_p, the power conversion is decreased with the increased damping coefficient, meaning for this case, the system is overdamped. For the case of T_e, the power conversion decreases when the damping coefficient is increased and decreased, meaning that this is the correctly optimised damping coefficient for the sea state.

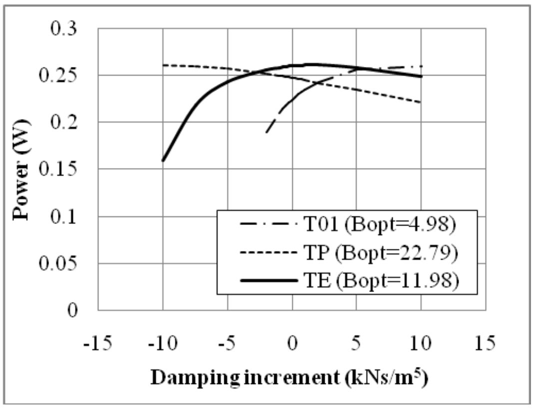


Figure 8 power conversion due to damping changes (around the optimised damping for the corresponding periods, see Table 3)

6 CONCLUSIONS

In hydrodynamic study of OWC wave energy converters, some issues on how to conduct the analysis of the OWC performance have been examined and discussed. From the study, the following conclusions can be drawn:

- The length of the imaginary piston for the water body in the water column has little influence on the responses of the motions for the frequency range of interest.
- The length of the piston can have significant effect on the time domain analysis, because a selection of the length of the piston may cause very different added mass at infinite frequency.
- The damping optimised method has been formulated for a simple case: a fixed OWC device, and in the analysis, the air compressibility has been ignored. as a result of the simplification, the optimised damping for the OWC device is quite similar to a point absorber.
- For a given sea state, the optimised damping coefficient based on the energy period gives best result.

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